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Energy Resources Sustainability Moisture Management in Buildings Measurement and Instruments Abbreviations and Symbols Units and Conversions Codes and Standards Additions and Corrections to the 2014, 2015, and 2016 volumes Index Back . . . This file is licensed to John Murray (). Publication Date: 6/1/2017 CONTRIBUTORS In addition to the Technical Committees, the following individuals contributed significantly to this volume. The appropriate chapter numbers follow each contributor's name. Kashif Nawaz (1, 4, 5) Oak Ridge National Laboratory Don Gatley (1) Gatley & Associates, Inc. Sebastian Herrmann (1) Hochschule Zittau/Görlitz, University of Applied Sciences Reinhard Radermacher (2) University of Maryland Hoseong Lee (2) University of Maryland Licensed for single user. © 2017 ASHRAE, Inc. Rick Couvillion (3, 4, 6) University of Arkansas Michael Ohadi (4, 6) University of Maryland Mirza Shah (5) Amir H. 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Brundage (34) Southern Company Services Stephen Idem (37) Tennessee Technological University ASHRAE HANDBOOK COMMITTEE Forrest S. Yount, Chair 2017 Fundamentals Volume Subcommittee: David P. Yuill, Chair Bass Abushakra Larry W. Akers Van D. Baxter Sheila J. Hayter Ramesh P. Paranjpey ASHRAE HANDBOOK STAFF W. Stephen Comstock, Publisher Director of Publications and Education Mark S. Owen, Editor Heather E. Kennedy, Managing Editor Nancy F. Thysell, Typographer/Page Designer David Soltis, Group Manager, and Jayne E. Jackson, Publication Traffic Administrator, Publishing Services This file is licensed to John Murray (). Publication Date: 6/1/2017 ASHRAE Research: Improving the Quality of Life ASHRAE is the world's foremost technical society in the fields of heating, ventilation, air conditioning, and refrigeration. Its members worldwide are individuals who share ideas, identify needs, support research, and write the industry's standards for testing and practice. The result is that engineers are better able to keep indoor environments safe and productive while protecting and preserving the outdoors for generations to come. One of the ways that ASHRAE supports its members' and industry's need for information is through ASHRAE Research. Thousands of individuals and companies support ASHRAE Research annually, enabling ASHRAE to report new data about material properties and building physics and to promote the application of innovative technologies. Chapters in the ASHRAE Handbook are updated through the experience of members of ASHRAE Technical Committees and through results of ASHRAE Research reported at ASHRAE conferences and published in ASHRAE special publications, ASHRAE Transactions, and ASHRAE's journal of archival research, Science and Technology for the Built Environment. For information about ASHRAE Research or to become a member, contact ASHRAE, 1791 Tullie Circle N.E., Atlanta, GA 30329; telephone: 404-636-8400; www.ashrae.org. Licensed for single user. © 2017 ASHRAE, Inc. Preface The 2017 ASHRAE Handbook—Fundamentals covers basic principles and data used in the HVAC&R industry. The ASHRAE Technical Committees that prepare these chapters provide new information, clarify existing content, delete obsolete materials, and reorganize chapters to make the Handbook more understandable and easier to use. An accompanying CD-ROM contains all the volume's chapters in both I-P and SI units. This edition includes a new chapter: • Chapter 36, Moisture Management in Buildings, presents data on indoor vapor release and measured indoor/outdoor vapor pressure/concentration differences, and discusses moisture sources and sinks that can reduce materials' durability, as well as the negative effects of insufficient or excessive indoor relative humidity. Other selected highlights include the following: • Chapter 7, Fundamentals of Control, has new content on thermostatic valve actuators, placement of sensors, auxiliary control devices, and network architecture. • Chapter 9, Thermal Comfort, has new content from ASHRAE research project RP-1504 on nonwestern clothing; combined chilled-ceiling, displacement ventilation, and vertical radiant temperature asymmetry effects on sedentary office work; and updates to align with ASHRAE Standard 55-2013. • Chapter 10, Indoor Environmental Health, has updates on bioaerosols, plus new content on electronic cigarettes and on climate change. • Chapter 11, Air Contaminants, has new content on particle sizes and settling times, particulate contaminant effects, polymerase chain reaction (PCR) measurement, volatility, mercury, e-cigarettes, and 3D printers. • Chapter 14, Climatic Design Information, includes new data for 8118 locations worldwide—an increase of 1675 locations from the 2013 edition of the chapter—as a result of ASHRAE research project RP-1699. • Chapter 15, Fenestration, has updated discussion on U-factor, solar-optical glazing properties, complex glazings and window coverings, tubular daylighting devices (TDDs), and spectrally selective glazing. • Chapter 16, Ventilation and Infiltration, has been updated and revised for clarity throughout, including recent research results on envelope air leakage. • Chapter 17, Residential Cooling and Heating Load Calculations, has updates for 2017 climate data and current standards. • Chapter 18, Nonresidential Cooling and Heating Load Calculations, has new design data for lighting power densities, motors, kitchen equipment, LED lighting, walls and roofs, and an updated example calculation. Copyright © 2017, ASHRAE • Chapter 19, Energy Estimating and Modeling Methods, extensively revised, has new sections on method development history, using models, uncertainty, thermal loads and model inputs, envelope components, HVAC components, terminal components, low-energy systems, natural and hybrid ventilation, daylighting, passive heating, hybrid inverse method, and model calibration. • Chapter 20, Space Air Diffusion, has new content on outlet types and characteristics, air curtains, thermal plumes, and air movement in occupied zones. • Chapter 21, Duct Design, was reorganized for ease of use, and updated for data from the latest version of the ASHRAE Duct Fitting Database. • Chapter 22, Pipe Design, has a new title and now incorporates the content of its sister chapter, Pipes, Tubes, and Fittings, from HVAC Systems and Equipment. Also added are content on PEX pipe, plus expanded applications. • Chapter 24, Airflow Around Buildings, has new content on flow patterns around building groups and isolated buildings, environmental impacts, pollutant dispersion and exhaust reentrainment, pedestrian wind comfort and safety, and wind-driven rain. • Chapter 30, Thermophysical Properties of Refrigerants, has new or revised data for R-1233zd(E), R-245fa, R-1234ze(E), and R1234yf. • Chapter 34, Energy Resources, has extensive updates for new statistics on worldwide energy use and resources. • Chapter 35, Sustainability, has new content on the water/energy nexus, embodied energy, and climate change. This volume is published, as a bound print volume and in electronic format on CD-ROM and online, in two editions: one using inch-pound (I-P) units of measurement, the other using the International System of Units (SI). Corrections to the 2014, 2015, and 2016 Handbook volumes can be found on the ASHRAE website at www.ashrae.org and in the Additions and Corrections section of this volume. Corrections for this volume will be listed in subsequent volumes and on the ASHRAE website. Reader comments are enthusiastically invited. To suggest improvements for a chapter, please comment using the form on the ASHRAE website or, using the cutout page(s) at the end of this volume's index, write to Handbook Editor, ASHRAE, 1791 Tullie Circle, Atlanta, GA 30329, or fax 678-539-2187, or e-mail Mark S. Owen Editor This file is licensed to John Murray (). Publication Date: 6/1/2017 Related Commercial Resources CHAPTER 1 PSYCHROMETRICS Composition of Dry and Moist Air .... U.S. Standard Atmosphere.. Thermodynamic Properties of Moist Air .... Thermodynamic Properties of Water at Saturation.................. Humidity Parameters .... Perfect Gas Relationships for Dry and Moist Air..................... 1.1 1.1 1.2 1.4 1.8 1.8 Thermodynamic Wet-Bulb and Dew-Point Temperature........... 1.9 Numerical Calculation of Moist Air Properties...................... 1.10 Psychrometric Charts.. 1.10 Typical Air-Conditioning Processes .... 1.12 Transport Properties of Moist Air.. 1.15 Symbols .... 1.15 SYCHROMETRICS uses thermodynamic properties to analyze conditions and processes involving moist air. This chapter discusses perfect gas relations and their use in common heating, cooling, and humidity control problems. Formulas developed by Herrmann et al. (2009) may be used where greater precision is required. Herrmann et al. (2009), Hyland and Wexler (1983a, 1983b), and Nelson and Sauer (2002) developed formulas for thermodynamic properties of moist air and water modeled as real gases. However, perfect gas relations can be substituted in most air-conditioning problems. Kuehn et al. (1998) showed that errors are less than 0.7% in calculating humidity ratio, enthalpy, and specific volume of saturated air at standard atmospheric pressure for a temperature range of –50 to 50°C. Furthermore, these errors decrease with decreasing pressure. Saturation conditions change when the interface radius is very small (e.g., with ultrafine water droplets). The relative molecular mass of water is 18.015 268 on the carbon-12 scale. The gas constant for water vapor is Licensed for single user. © 2017 ASHRAE, Inc. P 1. COMPOSITION OF DRY AND MOIST AIR Atmospheric air contains many gaseous components as well as water vapor and miscellaneous contaminants (e.g., smoke, pollen, and gaseous pollutants not normally present in free air far from pollution sources). Dry air is atmospheric air with all water vapor and contaminants removed. Its composition is relatively constant, but small variations in the amounts of individual components occur with time, geographic location, and altitude. Harrison (1965) lists the approximate percentage composition of dry air by volume as: nitrogen, 78.084; oxygen, 20.9476; argon, 0.934; neon, 0.001818; helium, 0.000524; methane, 0.00015; sulfur dioxide, 0 to 0.0001; hydrogen, 0.00005; and minor components such as krypton, xenon, and ozone, 0.0002. Harrison (1965) and Hyland and Wexler (1983a) used a value 0.0314 (circa 1955) for carbon dioxide. Carbon dioxide reached 0.0379 in 2005, is currently increasing by 0.00019 percent per year and is projected to reach 0.0438 in 2036 (Gatley et al. 2008; Keeling and Whorf 2005a, 2005b). Increases in carbon dioxide are offset by decreases in oxygen; consequently, the oxygen percentage in 2036 is projected to be 20.9352. Using the projected changes, the relative molecular mass for dry air for at least the first half of the 21st century is 28.966, based on the carbon-12 scale. The gas constant for dry air using the current Mohr and Taylor (2005) value for the universal gas constant is Rda = 8314.472/28.966 = 287.042 J/(kgda ·K) (1) Moist air is a binary (two-component) mixture of dry air and water vapor. The amount of water vapor varies from zero (dry air) to a maximum that depends on temperature and pressure. Saturation is a state of neutral equilibrium between moist air and the condensed water phase (liquid or solid); unless otherwise stated, it assumes a flat interface surface between moist air and the condensed phase. The preparation of this chapter is assigned to TC 1.1, Thermodynamics and Psychrometrics. 1.1 Copyright © 2017, ASHRAE Rw = 8314.472/18.015 268 = 461.524 J/(kgw ·K) 2. (2) U.S. STANDARD ATMOSPHERE The temperature and barometric pressure of atmospheric air vary considerably with altitude as well as with local geographic and weather conditions. The standard atmosphere gives a standard of reference for estimating properties at various altitudes. At sea level, standard temperature is 15°C; standard barometric pressure is 101.325 kPa. Temperature is assumed to decrease linearly with increasing altitude throughout the troposphere (lower atmosphere), and to be constant in the lower reaches of the stratosphere. The lower atmosphere is assumed to consist of dry air that behaves as a perfect gas. Gravity is also assumed constant at the standard value, 9.806 65 m/s2. Table 1 summarizes property data for altitudes to 10 000 m. Pressure values in Table 1 may be calculated from p = 101.325(1 – 2.25577 10 –5Z) 5.2559 (3) The equation for temperature as a function of altitude is t =15 – 0.0065Z (4) where Z = altitude, m p = barometric pressure, kPa t = temperature, °C Table 1 Standard Atmospheric Data for Altitudes to 10 000 m Altitude, m –500 0 500 1000 1500 2000 2500 3000 4000 5000 6000 7000 8000 9000 10 000 Temperature, °C 18.2 15.0 11.8 8.5 5.2 2.0 –1.2 –4.5 –11.0 –17.5 –24.0 –30.5 –37.0 –43.5 –50 Source: Adapted from NASA (1976). Pressure, kPa 107.478 101.325 95.461 89.875 84.556 79.495 74.682 70.108 61.640 54.020 47.181 41.061 35.600 30.742 26.436 This file is licensed to John Murray (). Publication Date: 6/1/2017 1.2 2017 ASHRAE Handbook—Fundamentals (SI) Equations (3) and (4) are accurate from –5000 m to 11 000 m. For higher altitudes, comprehensive tables of barometric pressure and other physical properties of the standard atmosphere, in both SI and I-P units, can be found in NASA (1976). 3. THERMODYNAMIC PROPERTIES OF MOIST AIR Table 2, developed from formulas by Herrmann et al. (2009), shows values of thermodynamic properties of moist air based on the International Temperature Scale of 1990 (ITS-90). This ideal scale differs slightly from practical temperature scales used for physical measurements. For example, the standard boiling point for water (at 101.325 kPa) occurs at 99.97°C on this scale rather than at the traditional 100°C. Most measurements are currently based on ITS-90 (Preston-Thomas 1990). The following properties are shown in Table 2: t = Celsius temperature, based on the ITS-90 and expressed relative to absolute temperature T in kelvins (K) by the following relation: T = t + 273.15 Ws = humidity ratio at saturation; gaseous phase (moist air) exists in equilibrium with condensed phase (liquid or solid) at given temperature and pressure (standard atmospheric pressure). At given values of temperature and pressure, humidity ratio W can have any value from zero to Ws. vda = specific volume of dry air, m3/kgda. vas = vs – vda, difference between specific volume of moist air at saturation and that of dry air, m3/kgda, at same pressure and temperature. vs = specific volume of moist air at saturation, m3/kgda. hda = specific enthalpy of dry air, kJ/kgda. In Table 2, hda is assigned a value of 0 at 0°C and standard atmospheric pressure. has = hs – hda, difference between specific enthalpy of moist air at saturation and that of dry air, kJ/kgda, at same pressure and temperature. hs = specific enthalpy of moist air at saturation, kJ/kgda. sda = specific entropy of dry air, kJ/(kgda ·K). In Table 2, sda is assigned a value of 0 at 0°C and standard atmospheric pressure. ss = specific entropy of moist air at saturation kJ/(kgda ·K). Licensed for single user. © 2017 ASHRAE, Inc. Table 2 Thermodynamic Properties of Moist Air at Standard Atmospheric Pressure, 101.325 kPa Temp., °C Humidity Ratio t Ws, kgw/kgda –60 –59 –58 –57 –56 –55 –54 –53 –52 –51 –50 –49 –48 –47 –46 –45 –44 –43 –42 –41 –40 –39 –38 –37 –36 –35 –34 –33 –32 –31 –30 –29 –28 –27 –26 –25 –24 –23 –22 –21 –20 0.0000067 0.0000076 0.0000087 0.0000100 0.0000114 0.0000129 0.0000147 0.0000167 0.0000190 0.0000215 0.0000243 0.0000275 0.0000311 0.0000350 0.0000395 0.0000445 0.0000500 0.0000562 0.0000631 0.0000708 0.0000793 0.0000887 0.0000992 0.0001108 0.0001237 0.0001379 0.0001536 0.0001710 0.0001902 0.0002113 0.0002345 0.0002602 0.0002883 0.0003193 0.0003532 0.0003905 0.0004314 0.0004761 0.0005251 0.0005787 0.0006373 Specific Volume, m3/kgda Specific Enthalpy, kJ/kgda vda vas vs hda 0.6027 0.6055 0.6084 0.6112 0.6141 0.6169 0.6198 0.6226 0.6255 0.6283 0.6312 0.6340 0.6369 0.6397 0.6425 0.6454 0.6482 0.6511 0.6539 0.6568 0.6596 0.6625 0.6653 0.6682 0.6710 0.6738 0.6767 0.6795 0.6824 0.6852 0.6881 0.6909 0.6938 0.6966 0.6994 0.7023 0.7051 0.7080 0.7108 0.7137 0.7165 0.0000 0.0000 0.0000 0.0000 0.0000 0.0000 0.0000 0.0000 0.0000 0.0000 0.0000 0.0000 0.0000 0.0000 0.0000 0.0001 0.0001 0.0001 0.0001 0.0001 0.0001 0.0001 0.0001 0.0001 0.0001 0.0002 0.0002 0.0002 0.0002 0.0003 0.0003 0.0003 0.0004 0.0004 0.0004 0.0005 0.0005 0.0006 0.0007 0.0007 0.6027 0.6055 0.6084 0.6112 0.6141 0.6169 0.6198 0.6226 0.6255 0.6283 0.6312 0.6340 0.6369 0.6397 0.6426 0.6454 0.6483 0.6511 0.6540 0.6568 0.6597 0.6626 0.6654 0.6683 0.6711 0.6740 0.6769 0.6797 0.6826 0.6855 0.6883 0.6912 0.6941 0.6970 0.6998 0.7027 0.7056 0.7085 0.7114 0.7143 0.7172 –60.341 –59.335 –58.329 –57.323 –56.317 –55.311 –54.305 –53.299 –52.293 –51.287 –50.281 –49.275 –48.269 –47.263 –46.257 –45.252 –44.246 –43.240 –42.234 –41.229 –40.223 –39.217 –38.211 –37.206 –36.200 –35.195 –34.189 –33.183 –32.178 –31.172 –30.167 –29.161 –28.156 –27.150 –26.144 –25.139 –24.133 –23.128 –22.122 –21.117 –20.111 has 0.016 0.018 0.021 0.024 0.027 0.031 0.035 0.040 0.046 0.052 0.059 0.066 0.075 0.085 0.095 0.107 0.121 0.136 0.153 0.172 0.192 0.215 0.241 0.269 0.301 0.336 0.374 0.417 0.464 0.516 0.573 0.636 0.706 0.782 0.866 0.958 1.059 1.170 1.291 1.424 1.570 Specific Entropy, kJ/(kgda·K) hs sda ss Temp., °C t –60.325 –59.317 –58.308 –57.299 –56.289 –55.280 –54.269 –53.258 –52.247 –51.235 –50.222 –49.209 –48.194 –47.179 –46.162 –45.144 –44.125 –43.104 –42.081 –41.057 –40.031 –39.002 –37.970 –36.936 –35.899 –34.859 –33.815 –32.766 –31.714 –30.656 –29.593 –28.525 –27.450 –26.368 –25.278 –24.181 –23.074 –21.958 –20.831 –19.693 –18.542 –0.2494 –0.2447 –0.2400 –0.2354 –0.2307 –0.2261 –0.2215 –0.2169 –0.2124 –0.2078 –0.2033 –0.1988 –0.1943 –0.1899 –0.1854 –0.1810 –0.1766 –0.1722 –0.1679 –0.1635 –0.1592 –0.1549 –0.1506 –0.1464 –0.1421 –0.1379 –0.1337 –0.1295 –0.1253 –0.1211 –0.1170 –0.1129 –0.1088 –0.1047 –0.1006 –0.0965 –0.0925 –0.0884 –0.0844 –0.0804 –0.0765 –0.2494 –0.2446 –0.2399 –0.2353 –0.2306 –0.2260 –0.2213 –0.2167 –0.2121 –0.2076 –0.2030 –0.1985 –0.1940 –0.1895 –0.1850 –0.1805 –0.1761 –0.1716 –0.1672 –0.1628 –0.1583 –0.1539 –0.1495 –0.1451 –0.1408 –0.1364 –0.1320 –0.1276 –0.1232 –0.1189 –0.1145 –0.1101 –0.1057 –0.1013 –0.0969 –0.0924 –0.0880 –0.0835 –0.0790 –0.0745 –0.0699 –60 –59 –58 –57 –56 –55 –54 –53 –52 –51 –50 –49 –48 –47 –46 –45 –44 –43 –42 –41 –40 –39 –38 –37 –36 –35 –34 –33 –32 –31 –30 –29 –28 –27 –26 –25 –24 –23 –22 –21 –20 This file is licensed to John Murray (). Publication Date: 6/1/2017 Psychrometrics 1.3 Table 2 Thermodynamic Properties of Moist Air at Standard Atmospheric Pressure, 101.325 kPa (Continued) Licensed for single user. © 2017 ASHRAE, Inc. Temp., °C Humidity Ratio t Ws, kgw/kgda –19 –18 –17 –16 –15 –14 –13 –12 –11 –10 –9 –8 –7 –6 –5 –4 –3 –2 –1 0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24 25 26 27 28 29 30 31 32 33 34 35 36 37 38 39 40 41 42 0.0007013 0.0007711 0.0008473 0.0009303 0.0010207 0.0011191 0.0012261 0.0013425 0.0014689 0.0016062 0.0017551 0.0019166 0.0020916 0.0022812 0.0024863 0.0027083 0.0029482 0.0032076 0.0034877 0.0037900 0.004076 0.004382 0.004708 0.005055 0.005425 0.005819 0.006238 0.006684 0.007158 0.007663 0.008199 0.008768 0.009372 0.010013 0.010694 0.011415 0.012181 0.012991 0.013851 0.014761 0.015724 0.016744 0.017823 0.018965 0.020173 0.021451 0.022802 0.024229 0.025738 0.027333 0.029018 0.030797 0.032677 0.034663 0.036760 0.038975 0.041313 0.043783 0.046391 0.049145 0.052053 0.055124 Specific Volume, m3/kgda Specific Enthalpy, kJ/kgda Specific Entropy, kJ/(kgda·K) vda vas vs hda has hs sda ss Temp., °C t 0.7193 0.7222 0.7250 0.7279 0.7307 0.7336 0.7364 0.7392 0.7421 0.7449 0.7478 0.7506 0.7534 0.7563 0.7591 0.7620 0.7648 0.7677 0.7705 0.7733 0.7762 0.7790 0.7819 0.7847 0.7875 0.7904 0.7932 0.7961 0.7989 0.8017 0.8046 0.8074 0.8103 0.8131 0.8159 0.8188 0.8216 0.8245 0.8273 0.8301 0.8330 0.8358 0.8387 0.8415 0.8443 0.8472 0.8500 0.8529 0.8557 0.8585 0.8614 0.8642 0.8671 0.8699 0.8727 0.8756 0.8784 0.8813 0.8841 0.8869 0.8898 0.8926 0.0008 0.0009 0.0010 0.0011 0.0012 0.0013 0.0014 0.0016 0.0017 0.0019 0.0021 0.0023 0.0025 0.0028 0.0030 0.0033 0.0036 0.0039 0.0043 0.0047 0.0051 0.0055 0.0059 0.0064 0.0068 0.0074 0.0079 0.0085 0.0092 0.0098 0.0106 0.0113 0.0122 0.0131 0.0140 0.0150 0.0160 0.0172 0.0184 0.0196 0.0210 0.0224 0.0240 0.0256 0.0273 0.0291 0.0311 0.0331 0.0353 0.0376 0.0400 0.0426 0.0454 0.0483 0.0514 0.0546 0.0581 0.0618 0.0657 0.0698 0.0741 0.0788 0.7201 0.7231 0.7260 0.7290 0.7319 0.7349 0.7378 0.7408 0.7438 0.7468 0.7499 0.7529 0.7560 0.7591 0.7622 0.7653 0.7684 0.7716 0.7748 0.7780 0.7813 0.7845 0.7878 0.7911 0.7944 0.7978 0.8012 0.8046 0.8081 0.8116 0.8152 0.8188 0.8224 0.8262 0.8299 0.8338 0.8377 0.8416 0.8457 0.8498 0.8540 0.8583 0.8626 0.8671 0.8716 0.8763 0.8811 0.8860 0.8910 0.8961 0.9014 0.9069 0.9124 0.9182 0.9241 0.9302 0.9365 0.9430 0.9498 0.9567 0.9639 0.9714 –19.106 –18.100 –17.095 –16.089 –15.084 –14.078 –13.073 –12.067 –11.062 –10.056 –9.050 –8.045 –7.039 –6.034 –5.028 –4.023 –3.017 –2.011 –1.006 0.000 1.006 2.011 3.017 4.023 5.029 6.034 7.040 8.046 9.052 10.058 11.063 12.069 13.075 14.081 15.087 16.093 17.099 18.105 19.111 20.117 21.124 22.130 23.136 24.142 25.148 26.155 27.161 28.167 29.174 30.180 31.187 32.193 33.200 34.207 35.213 36.220 37.227 38.233 39.240 40.247 41.254 42.261 1.728 1.902 2.091 2.298 2.523 2.769 3.036 3.326 3.642 3.986 4.358 4.763 5.202 5.677 6.192 6.750 7.354 8.007 8.712 9.475 10.198 10.970 11.794 12.673 13.611 14.610 15.674 16.807 18.013 19.297 20.661 22.111 23.653 25.290 27.028 28.873 30.830 32.906 35.108 37.441 39.914 42.534 45.308 48.246 51.355 54.647 58.129 61.813 65.709 69.829 74.186 78.792 83.661 88.807 94.247 99.995 106.069 112.488 119.272 126.440 134.016 142.023 –17.377 –16.198 –15.003 –13.791 –12.560 –11.310 –10.037 –8.741 –7.419 –6.070 –4.692 –3.282 –1.838 –0.356 1.164 2.728 4.337 5.995 7.707 9.475 11.203 12.981 14.811 16.696 18.639 20.644 22.714 24.853 27.065 29.354 31.724 34.181 36.728 39.371 42.115 44.966 47.929 51.011 54.219 57.559 61.038 64.663 68.444 72.388 76.504 80.801 85.290 89.980 94.883 100.010 105.373 110.986 116.861 123.014 129.460 136.215 143.296 150.722 158.512 166.687 175.270 184.284 –0.0725 –0.0685 –0.0646 –0.0607 –0.0568 –0.0529 –0.0490 –0.0452 –0.0413 –0.0375 –0.0337 –0.0299 –0.0261 –0.0223 –0.0186 –0.0148 –0.0111 –0.0074 –0.0037 0.0000 0.0037 0.0073 0.0110 0.0146 0.0182 0.0219 0.0254 0.0290 0.0326 0.0362 0.0397 0.0432 0.0468 0.0503 0.0538 0.0573 0.0607 0.0642 0.0676 0.0711 0.0745 0.0779 0.0813 0.0847 0.0881 0.0915 0.0948 0.0982 0.1015 0.1048 0.1081 0.1115 0.1147 0.1180 0.1213 0.1246 0.1278 0.1311 0.1343 0.1375 0.1407 0.1439 –0.0653 –0.0607 –0.0560 –0.0513 –0.0465 –0.0416 –0.0367 –0.0317 –0.0267 –0.0215 –0.0163 –0.0110 –0.0055 0.0000 0.0057 0.0115 0.0175 0.0236 0.0299 0.0364 0.0427 0.0492 0.0559 0.0627 0.0697 0.0769 0.0843 0.0919 0.0997 0.1078 0.1162 0.1248 0.1338 0.1430 0.1525 0.1624 0.1726 0.1832 0.1942 0.2057 0.2175 0.2298 0.2426 0.2560 0.2698 0.2842 0.2992 0.3148 0.3311 0.3481 0.3658 0.3843 0.4035 0.4236 0.4447 0.4666 0.4895 0.5135 0.5386 0.5649 0.5923 0.6211 –19 –18 –17 –16 –15 –14 –13 –12 –11 –10 –9 –8 –7 –6 –5 –4 –3 –2 –1 0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24 25 26 27 28 29 30 31 32 33 34 35 36 37 38 39 40 41 42 This file is licensed to John Murray (). Publication Date: 6/1/2017 1.4 2017 ASHRAE Handbook—Fundamentals (SI) Table 2 Thermodynamic Properties of Moist Air at Standard Atmospheric Pressure, 101.325 kPa (Continued) Temp., °C Humidity Ratio t Ws, kgw/kgda Licensed for single user. © 2017 ASHRAE, Inc. 43 44 45 46 47 48 49 50 51 52 53 54 55 56 57 58 59 60 61 62 63 64 65 66 67 68 69 70 71 72 73 74 75 76 77 78 79 80 81 82 83 84 85 86 87 88 89 90 4. 0.058368 0.061795 0.065416 0.069242 0.073286 0.077561 0.082081 0.086863 0.091922 0.097278 0.102949 0.108958 0.115326 0.122080 0.129248 0.136858 0.144945 0.153545 0.162697 0.172446 0.182842 0.193937 0.205794 0.218478 0.232067 0.246645 0.262309 0.279167 0.297343 0.316979 0.338237 0.361304 0.386399 0.413774 0.443727 0.476610 0.512842 0.552926 0.597470 0.647218 0.703089 0.766233 0.838105 0.920580 1.016105 1.127952 1.260579 1.420235 Specific Volume, m3/kgda Specific Enthalpy, kJ/kgda vda vas vs hda has hs 0.8955 0.8983 0.9011 0.9040 0.9068 0.9096 0.9125 0.9153 0.9182 0.9210 0.9238 0.9267 0.9295 0.9324 0.9352 0.9380 0.9409 0.9437 0.9465 0.9494 0.9522 0.9551 0.9579 0.9607 0.9636 0.9664 0.9692 0.9721 0.9749 0.9778 0.9806 0.9834 0.9863 0.9891 0.9919 0.9948 0.9976 1.0005 1.0033 1.0061 1.0090 1.0118 1.0146 1.0175 1.0203 1.0232 1.0260 1.0288 0.0837 0.0888 0.0943 0.1002 0.1063 0.1129 0.1198 0.1272 0.1350 0.1433 0.1521 0.1614 0.1714 0.1819 0.1932 0.2051 0.2179 0.2315 0.2460 0.2615 0.2780 0.2957 0.3147 0.3350 0.3568 0.3803 0.4056 0.4328 0.4622 0.4941 0.5287 0.5663 0.6072 0.6520 0.7010 0.7550 0.8145 0.8805 0.9539 1.0360 1.1283 1.2328 1.3519 1.4887 1.6473 1.8332 2.0539 2.3198 0.9791 0.9871 0.9955 1.0041 1.0131 1.0225 1.0323 1.0425 1.0531 1.0643 1.0759 1.0881 1.1009 1.1143 1.1284 1.1432 1.1587 1.1752 1.1925 1.2108 1.2302 1.2508 1.2726 1.2957 1.3204 1.3467 1.3748 1.4049 1.4372 1.4719 1.5093 1.5497 1.5935 1.6411 1.6930 1.7497 1.8121 1.8809 1.9572 2.0421 2.1373 2.2446 2.3665 2.5062 2.6676 2.8564 3.0799 3.3487 43.268 44.275 45.282 46.289 47.297 48.304 49.311 50.319 51.326 52.334 53.341 54.349 55.356 56.364 57.372 58.380 59.388 60.396 61.404 62.412 63.420 64.428 65.436 66.445 67.453 68.462 69.470 70.479 71.488 72.496 73.505 74.514 75.523 76.532 77.542 78.551 79.560 80.570 81.579 82.589 83.598 84.608 85.618 86.628 87.638 88.648 89.658 90.668 150.486 159.432 168.890 178.892 189.470 200.660 212.501 225.034 238.305 252.362 267.256 283.047 299.794 317.567 336.439 356.490 377.809 400.493 424.650 450.398 477.868 507.204 538.570 572.145 608.133 646.762 688.288 733.004 781.240 833.375 889.844 951.149 1017.871 1090.688 1170.398 1257.941 1354.439 1461.236 1579.961 1712.604 1861.625 2030.099 2221.922 2442.105 2697.204 2995.967 3350.325 3776.998 193.754 203.707 214.172 225.181 236.766 248.964 261.812 275.353 289.631 304.695 320.598 337.395 355.151 373.931 393.811 414.869 437.197 460.889 486.054 512.810 541.287 571.632 604.006 638.590 675.587 715.224 757.759 803.483 852.728 905.872 963.350 1025.663 1093.394 1167.220 1247.939 1336.492 1433.999 1541.806 1661.540 1795.193 1945.223 2114.707 2307.539 2528.732 2784.842 3084.614 3439.983 3867.666 THERMODYNAMIC PROPERTIES OF WATER AT SATURATION Table 3 shows thermodynamic properties of water at saturation for temperatures from –60 to 160°C, calculated by the formulations described by IAPWS (2007, 2009, 2011, 2014). Symbols in the table follow standard steam table nomenclature. These properties are based on ITS-90. The internal energy and entropy of saturated liquid water are both assigned the value zero at the triple point, 0.01°C. Between the triple-point and critical-point temperatures of Specific Entropy, kJ/(kgda·K) sda 0.1471 0.1503 0.1535 0.1566 0.1598 0.1629 0.1660 0.1692 0.1723 0.1754 0.1785 0.1816 0.1846 0.1877 0.1908 0.1938 0.1968 0.1999 0.2029 0.2059 0.2089 0.2119 0.2149 0.2179 0.2208 0.2238 0.2268 0.2297 0.2326 0.2356 0.2385 0.2414 0.2443 0.2472 0.2501 0.2529 0.2558 0.2587 0.2615 0.2644 0.2672 0.2701 0.2729 0.2757 0.2785 0.2813 0.2841 0.2869 ss Temp., °C t 0.6512 0.6828 0.7159 0.7507 0.7871 0.8254 0.8655 0.9078 0.9522 0.9989 1.0481 1.0999 1.1545 1.2121 1.2729 1.3371 1.4050 1.4769 1.5530 1.6338 1.7195 1.8105 1.9075 2.0107 2.1209 2.2386 2.3647 2.4998 2.6449 2.8012 2.9697 3.1520 3.3497 3.5645 3.7989 4.0554 4.3371 4.6478 4.9921 5.3755 5.8048 6.2886 6.8377 7.4661 8.1920 9.0397 10.0422 11.2459 43 44 45 46 47 48 49 50 51 52 53 54 55 56 57 58 59 60 61 62 63 64 65 66 67 68 69 70 71 72 73 74 75 76 77 78 79 80 81 82 83 84 85 86 87 88 89 90 water, both saturated liquid and saturated vapor may coexist in equilibrium; below the triple-point temperature, both saturated ice and saturated vapor may coexist in equilibrium. The water vapor saturation pressure is required to determine a number of moist air properties, principally the saturation humidity ratio. Values may be obtained from Table 3 or calculated from the following formulas (Hyland and Wexler 1983b). The 1983 formulas are within 300 ppm of the latest IAPWS formulations. For higher accuracy, developers of software and others are referred to IAPWS (2007, 2011). This file is licensed to John Murray (). Publication Date: 6/1/2017 Psychrometrics 1.5 Table 3 Thermodynamic Properties of Water at Saturation Licensed for single user. © 2017 ASHRAE, Inc. Temp., °C t –60 –59 –58 –57 –56 –55 –54 –53 –52 –51 –50 –49 –48 –47 –46 –45 –44 –43 –42 –41 –40 –39 –38 –37 –36 –35 –34 –33 –32 –31 –30 –29 –28 –27 –26 –25 –24 –23 –22 –21 –20 –19 –18 –17 –16 –15 –14 –13 –12 –11 –10 –9 –8 –7 –6 –5 –4 –3 –2 –1 0 Absolute Pressure pws, kPa 0.00108 0.00124 0.00141 0.00161 0.00184 0.00209 0.00238 0.00271 0.00307 0.00348 0.00394 0.00445 0.00503 0.00568 0.00640 0.00720 0.00810 0.00910 0.01022 0.01146 0.01284 0.01437 0.01607 0.01795 0.02004 0.02234 0.02489 0.02771 0.03081 0.03423 0.03801 0.04215 0.04672 0.05173 0.05724 0.06327 0.06989 0.07714 0.08508 0.09376 0.10324 0.11360 0.12490 0.13722 0.15065 0.16527 0.18119 0.19849 0.21729 0.23771 0.25987 0.28391 0.30995 0.33817 0.36871 0.40174 0.43745 0.47604 0.51770 0.56266 0.61115 Specific Volume, m3/kgw Sat. Solid vi /vf 0.001081 0.001082 0.001082 0.001082 0.001082 0.001082 0.001082 0.001083 0.001083 0.001083 0.001083 0.001083 0.001083 0.001084 0.001084 0.001084 0.001084 0.001084 0.001084 0.001085 0.001085 0.001085 0.001085 0.001085 0.001085 0.001086 0.001086 0.001086 0.001086 0.001086 0.001086 0.001087 0.001087 0.001087 0.001087 0.001087 0.001088 0.001088 0.001088 0.001088 0.001088 0.001088 0.001089 0.001089 0.001089 0.001089 0.001089 0.001090 0.001090 0.001090 0.001090 0.001090 0.001090 0.001091 0.001091 0.001091 Specific Enthalpy, kJ/kgw Specific Entropy, kJ/(kgw ·K) Evap. vig /vfg Sat. Vapor vg Sat. Solid hi /hf Evap. hig /hfg Sat. Vapor hg Sat. Solid si /sf Evap. sig /sfg Sat. Vapor sg Temp., °C t 90971.58 79885.31 70235.77 61826.23 54488.28 48077.54 42470.11 37559.49 33254.07 29474.87 26153.80 23232.03 20658.70 18389.75 16387.03 14617.39 13052.07 11666.02 10437.46 9347.38 8379.20 7518.44 6752.43 6070.08 5461.68 4918.69 4433.64 3999.95 3611.82 3264.15 2952.46 2672.77 2421.58 2195.80 1992.68 1809.79 1644.99 1496.36 1362.21 1241.03 1131.49 1032.38 942.64 861.34 787.61 720.70 659.94 604.72 554.51 508.81 467.19 429.25 394.66 363.09 334.26 307.92 283.82 261.78 241.60 223.10 206.15 90971.58 79885.31 70235.78 61826.24 54488.28 48077.54 42470.11 37559.50 33254.07 29474.87 26153.80 23232.04 20658.70 18389.75 16387.03 14617.39 13052.07 11666.02 10437.46 9347.38 8379.20 7518.44 6752.43 6070.08 5461.68 4918.69 4433.64 3999.95 3611.82 3264.16 2952.46 2672.77 2421.58 2195.80 1992.68 1809.79 1644.99 1496.36 1362.21 1241.03 1131.49 1032.38 942.65 861.34 787.61 720.70 659.94 604.73 554.51 508.81 467.19 429.26 394.66 363.09 334.26 307.92 283.83 261.78 241.60 223.11 206.15 –446.12 –444.46 –442.79 –441.11 –439.42 –437.73 –436.03 –434.32 –432.61 –430.88 –429.16 –427.42 –425.68 –423.93 –422.17 –420.40 –418.63 –416.85 –415.06 –413.27 –411.47 –409.66 –407.85 –406.02 –404.19 –402.36 –400.51 –398.66 –396.80 –394.94 –393.06 –391.18 –389.29 –387.40 –385.50 –383.59 –381.67 –379.75 –377.81 –375.88 –373.93 –371.98 –370.01 –368.05 –366.07 –364.09 –362.10 –360.10 –358.10 –356.08 –354.06 –352.04 –350.00 –347.96 –345.91 –343.86 –341.79 –339.72 –337.64 –335.56 –333.47 2836.27 2836.45 2836.63 2836.81 2836.97 2837.13 2837.28 2837.42 2837.56 2837.69 2837.81 2837.93 2838.04 2838.14 2838.23 2838.32 2838.39 2838.47 2838.53 2838.59 2838.64 2838.68 2838.72 2838.74 2838.76 2838.78 2838.78 2838.78 2838.77 2838.75 2838.73 2838.70 2838.66 2838.61 2838.56 2838.49 2838.42 2838.35 2838.26 2838.17 2838.07 2837.96 2837.84 2837.72 2837.59 2837.45 2837.30 2837.14 2836.98 2836.80 2836.62 2836.44 2836.24 2836.03 2835.82 2835.60 2835.37 2835.13 2834.88 2834.63 2834.36 2390.14 2391.99 2393.85 2395.70 2397.55 2399.40 2401.25 2403.10 2404.95 2406.81 2408.66 2410.51 2412.36 2414.21 2416.06 2417.91 2419.76 2421.62 2423.47 2425.32 2427.17 2429.02 2430.87 2432.72 2434.57 2436.42 2438.27 2440.12 2441.97 2443.82 2445.67 2447.51 2449.36 2451.21 2453.06 2454.91 2456.75 2458.60 2460.45 2462.29 2464.14 2465.98 2467.83 2469.67 2471.51 2473.36 2475.20 2477.04 2478.88 2480.72 2482.56 2484.40 2486.23 2488.07 2489.91 2491.74 2493.57 2495.41 2497.24 2499.07 2500.90 –1.6842 –1.6764 –1.6687 –1.6609 –1.6531 –1.6453 –1.6375 –1.6298 –1.6220 –1.6142 –1.6065 –1.5987 –1.5909 –1.5832 –1.5754 –1.5677 –1.5599 –1.5522 –1.5444 –1.5367 –1.5289 –1.5212 –1.5135 –1.5057 –1.4980 –1.4903 –1.4825 –1.4748 –1.4671 –1.4594 –1.4516 –1.4439 –1.4362 –1.4285 –1.4208 –1.4131 –1.4054 –1.3977 –1.3899 –1.3822 –1.3745 –1.3668 –1.3591 –1.3514 –1.3437 –1.3360 –1.3284 –1.3207 –1.3130 –1.3053 –1.2976 –1.2899 –1.2822 –1.2745 –1.2668 –1.2592 –1.2515 –1.2438 –1.2361 –1.2284 –1.2208 13.3064 13.2452 13.1845 13.1243 13.0646 13.0054 12.9468 12.8886 12.8310 12.7738 12.7171 12.6609 12.6051 12.5498 12.4950 12.4406 12.3867 12.3331 12.2801 12.2274 12.1752 12.1234 12.0720 12.0210 11.9704 11.9202 11.8703 11.8209 11.7718 11.7231 11.6748 11.6269 11.5793 11.5321 11.4852 11.4386 11.3925 11.3466 11.3011 11.2559 11.2110 11.1665 11.1223 11.0784 11.0348 10.9915 10.9485 10.9058 10.8634 10.8213 10.7795 10.7380 10.6967 10.6558 10.6151 10.5747 10.5345 10.4946 10.4550 10.4157 10.3766 11.6222 11.5687 11.5158 11.4634 11.4115 11.3601 11.3092 11.2589 11.2090 11.1596 11.1106 11.0622 11.0142 10.9666 10.9196 10.8729 10.8267 10.7810 10.7356 10.6907 10.6462 10.6022 10.5585 10.5152 10.4724 10.4299 10.3878 10.3461 10.3047 10.2638 10.2232 10.1830 10.1431 10.1036 10.0644 10.0256 9.9871 9.9489 9.9111 9.8736 9.8365 9.7996 9.7631 9.7269 9.6910 9.6554 9.6201 9.5851 9.5504 9.5160 9.4819 9.4481 9.4145 9.3812 9.3482 9.3155 9.2830 9.2508 9.2189 9.1872 9.1558 –60 –59 –58 –57 –56 –55 –54 –53 –52 –51 –50 –49 –48 –47 –46 –45 –44 –43 –42 –41 –40 –39 –38 –37 –36 –35 –34 –33 –32 –31 –30 –29 –28 –27 –26 –25 –24 –23 –22 –21 –20 –19 –18 –17 –16 –15 –14 –13 –12 –11 –10 –9 –8 –7 –6 –5 –4 –3 –2 –1 0 This file is licensed to John Murray (). Publication Date: 6/1/2017 1.6 2017 ASHRAE Handbook—Fundamentals (SI) Table 3 Thermodynamic Properties of Water at Saturation (Continued) Temp., °C t Absolute Pressure pws, kPa Specific Volume, m3/kgw Sat. Solid vi /vf Evap. vig /vfg Sat. Vapor vg Specific Enthalpy, kJ/kgw Specific Entropy, kJ/(kgw ·K) Sat. Solid hi /hf Evap. hig /hfg Sat. Vapor hg Sat. Solid si /sf Evap. sig /sfg Sat. Vapor sg Temp., °C t –0.04 4.18 8.39 12.60 16.81 21.02 25.22 29.43 33.63 37.82 42.02 46.22 50.41 54.60 58.79 62.98 67.17 71.36 75.55 79.73 83.92 88.10 92.29 96.47 100.66 104.84 109.02 113.20 117.38 121.56 125.75 129.93 134.11 138.29 142.47 146.64 150.82 155.00 159.18 163.36 167.54 171.72 175.90 180.08 184.26 188.44 192.62 196.80 200.98 205.16 209.34 213.52 217.70 221.88 226.06 230.24 234.42 238.61 242.79 246.97 251.15 255.34 259.52 263.71 267.89 272.08 276.27 280.45 284.64 288.83 2500.93 2498.55 2496.17 2493.80 2491.42 2489.05 2486.68 2484.31 2481.94 2479.58 2477.21 2474.84 2472.48 2470.11 2467.75 2465.38 2463.01 2460.65 2458.28 2455.92 2453.55 2451.18 2448.81 2446.45 2444.08 2441.71 2439.33 2436.96 2434.59 2432.21 2429.84 2427.46 2425.08 2422.70 2420.32 2417.94 2415.56 2413.17 2410.78 2408.39 2406.00 2403.61 2401.21 2398.82 2396.42 2394.02 2391.61 2389.21 2386.80 2384.39 2381.97 2379.56 2377.14 2374.72 2372.30 2369.87 2367.44 2365.01 2362.57 2360.13 2357.69 2355.25 2352.80 2350.35 2347.89 2345.43 2342.97 2340.50 2338.03 2335.56 2500.89 2502.73 2504.57 2506.40 2508.24 2510.07 2511.91 2513.74 2515.57 2517.40 2519.23 2521.06 2522.89 2524.71 2526.54 2528.36 2530.19 2532.01 2533.83 2535.65 2537.47 2539.29 2541.10 2542.92 2544.73 2546.54 2548.35 2550.16 2551.97 2553.78 2555.58 2557.39 2559.19 2560.99 2562.79 2564.58 2566.38 2568.17 2569.96 2571.75 2573.54 2575.33 2577.11 2578.89 2580.67 2582.45 2584.23 2586.00 2587.77 2589.54 2591.31 2593.08 2594.84 2596.60 2598.35 2600.11 2601.86 2603.61 2605.36 2607.10 2608.85 2610.58 2612.32 2614.05 2615.78 2617.51 2619.23 2620.96 2622.67 2624.39 –0.0002 0.0153 0.0306 0.0459 0.0611 0.0763 0.0913 0.1064 0.1213 0.1362 0.1511 0.1659 0.1806 0.1953 0.2099 0.2245 0.2390 0.2534 0.2678 0.2822 0.2965 0.3108 0.3250 0.3391 0.3532 0.3673 0.3813 0.3952 0.4091 0.4230 0.4368 0.4506 0.4643 0.4780 0.4916 0.5052 0.5187 0.5322 0.5457 0.5591 0.5724 0.5858 0.5990 0.6123 0.6255 0.6386 0.6517 0.6648 0.6778 0.6908 0.7038 0.7167 0.7296 0.7424 0.7552 0.7680 0.7807 0.7934 0.8060 0.8186 0.8312 0.8438 0.8563 0.8687 0.8811 0.8935 0.9059 0.9182 0.9305 0.9428 9.1559 9.1138 9.0721 9.0306 8.9895 8.9486 8.9081 8.8678 8.8278 8.7882 8.7488 8.7096 8.6708 8.6322 8.5939 8.5559 8.5181 8.4806 8.4434 8.4064 8.3696 8.3331 8.2969 8.2609 8.2251 8.1895 8.1542 8.1192 8.0843 8.0497 8.0153 7.9812 7.9472 7.9135 7.8800 7.8467 7.8136 7.7807 7.7480 7.7155 7.6832 7.6512 7.6193 7.5876 7.5561 7.5248 7.4937 7.4628 7.4320 7.4015 7.3711 7.3409 7.3109 7.2811 7.2514 7.2219 7.1926 7.1634 7.1344 7.1056 7.0770 7.0485 7.0201 6.9919 6.9639 6.9361 6.9083 6.8808 6.8534 6.8261 9.1558 9.1291 9.1027 9.0765 9.0506 9.0249 8.9994 8.9742 8.9492 8.9244 8.8998 8.8755 8.8514 8.8275 8.8038 8.7804 8.7571 8.7341 8.7112 8.6886 8.6661 8.6439 8.6218 8.6000 8.5783 8.5568 8.5355 8.5144 8.4934 8.4727 8.4521 8.4317 8.4115 8.3914 8.3715 8.3518 8.3323 8.3129 8.2936 8.2746 8.2557 8.2369 8.2183 8.1999 8.1816 8.1634 8.1454 8.1276 8.1099 8.0923 8.0749 8.0576 8.0405 8.0235 8.0066 7.9899 7.9733 7.9568 7.9405 7.9243 7.9082 7.8922 7.8764 7.8607 7.8451 7.8296 7.8142 7.7990 7.7839 7.7689 0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24 25 26 27 28 29 30 31 32 33 34 35 36 37 38 39 40 41 42 43 44 45 46 47 48 49 50 51 52 53 54 55 56 57 58 59 60 61 62 63 64 65 66 67 68 69 Licensed for single user. © 2017 ASHRAE, Inc. Transition from saturated solid to saturated liquid 0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24 25 26 27 28 29 30 31 32 33 34 35 36 37 38 39 40 41 42 43 44 45 46 47 48 49 50 51 52 53 54 55 56 57 58 59 60 61 62 63 64 65 66 67 68 69 0.6112 0.6571 0.7060 0.7581 0.8135 0.8726 0.9354 1.0021 1.0730 1.1483 1.2282 1.3129 1.4028 1.4981 1.5989 1.7057 1.8188 1.9383 2.0647 2.1982 2.3392 2.4881 2.6452 2.8109 2.9856 3.1697 3.3637 3.5679 3.7828 4.0089 4.2467 4.4966 4.7592 5.0351 5.3247 5.6286 5.9475 6.2818 6.6324 6.9997 7.3844 7.7873 8.2090 8.6503 9.1118 9.5944 10.0988 10.6259 11.1764 11.7512 12.3513 12.9774 13.6305 14.3116 15.0215 15.7614 16.5322 17.3350 18.1708 19.0407 19.9458 20.8873 21.8664 22.8842 23.9421 25.0411 26.1827 27.3680 28.5986 29.8756 0.001000 0.001000 0.001000 0.001000 0.001000 0.001000 0.001000 0.001000 0.001000 0.001000 0.001000 0.001000 0.001000 0.001001 0.001001 0.001001 0.001001 0.001001 0.001001 0.001002 0.001002 0.001002 0.001002 0.001003 0.001003 0.001003 0.001003 0.001004 0.001004 0.001004 0.001004 0.001005 0.001005 0.001005 0.001006 0.001006 0.001006 0.001007 0.001007 0.001008 0.001008 0.001009 0.001009 0.001010 0.001010 0.001011 0.001011 0.001012 0.001012 0.001013 0.001013 0.001014 0.001014 0.001015 0.001015 0.001016 0.001016 0.001017 0.001017 0.001018 0.001018 0.001019 0.001019 0.001020 0.001020 0.001021 0.001021 0.001022 0.001022 206.139 192.444 179.763 168.013 157.120 147.016 137.637 128.927 120.833 113.308 106.308 99.792 93.723 88.069 82.797 77.880 73.290 69.005 65.002 61.260 57.760 54.486 51.421 48.551 45.862 43.340 40.976 38.757 36.674 34.718 32.881 31.153 29.528 28.000 26.561 25.207 23.931 22.728 21.594 20.525 19.516 18.564 17.664 16.815 16.012 15.252 14.534 13.855 13.212 12.603 12.027 11.481 10.963 10.472 10.006 9.5639 9.1444 8.7461 8.3678 8.0083 7.6666 7.3418 7.0328 6.7389 6.4591 6.1928 5.9392 5.6976 5.4674 5.2479 206.140 192.445 179.764 168.014 157.121 147.017 137.638 128.928 120.834 113.309 106.309 99.793 93.724 88.070 82.798 77.881 73.291 69.006 65.003 61.261 57.761 54.487 51.422 48.552 45.863 43.341 40.977 38.758 36.675 34.719 32.882 31.154 29.529 28.001 26.562 25.208 23.932 22.729 21.595 20.526 19.517 18.565 17.665 16.816 16.013 15.253 14.535 13.856 13.213 12.604 12.028 11.482 10.964 10.473 10.007 9.5649 9.1454 8.7471 8.3688 8.0093 7.6677 7.3428 7.0338 6.7399 6.4601 6.1938 5.9402 5.6986 5.4684 5.2490 This file is licensed to John Murray (). 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Temp., °C t 70 71 72 73 74 75 76 77 78 79 80 81 82 83 84 85 86 87 88 89 90 91 92 93 94 95 96 97 98 99 100 101 102 103 104 105 106 107 108 109 110 111 112 113 114 115 116 117 118 119 120 122 124 126 128 130 132 134 136 138 140 142 144 146 148 150 152 154 156 158 160 Absolute Pressure pws, kPa 31.2006 32.5750 34.0001 35.4775 37.0088 38.5954 40.2389 41.9409 43.7031 45.5271 47.4147 49.3676 51.3875 53.4762 55.6355 57.8675 60.1738 62.5565 65.0174 67.5587 70.1824 72.8904 75.6849 78.5681 81.5420 84.6089 87.7711 91.0308 94.3902 97.8518 101.4180 105.0910 108.8735 112.7678 116.7765 120.9021 125.1472 129.5145 134.0065 138.6261 143.3760 148.2588 153.2775 158.4348 163.7337 169.1770 174.7678 180.5090 186.4036 192.4547 198.6654 211.5782 225.1676 239.4597 254.4813 270.2596 286.8226 304.1989 322.4175 341.5081 361.5010 382.4271 404.3178 427.2053 451.1220 476.1014 502.1771 529.3834 557.7555 587.3287 618.1392 Specific Volume, m3/kgw Sat. Liquid vi /vf 0.001023 0.001023 0.001024 0.001025 0.001025 0.001026 0.001026 0.001027 0.001028 0.001028 0.001029 0.001030 0.001030 0.001031 0.001032 0.001032 0.001033 0.001034 0.001035 0.001035 0.001036 0.001037 0.001037 0.001038 0.001039 0.001040 0.001040 0.001041 0.001042 0.001043 0.001043 0.001044 0.001045 0.001046 0.001047 0.001047 0.001048 0.001049 0.001050 0.001051 0.001052 0.001052 0.001053 0.001054 0.001055 0.001056 0.001057 0.001058 0.001059 0.001059 0.001060 0.001062 0.001064 0.001066 0.001068 0.001070 0.001072 0.001074 0.001076 0.001078 0.001080 0.001082 0.001084 0.001086 0.001088 0.001091 0.001093 0.001095 0.001097 0.001100 0.001102 Specific Enthalpy, kJ/kgw Specific Entropy, kJ/(kgw ·K) Evap. vig /vfg Sat. Vapor vg Sat. Liquid hi /hf Evap. hig /hfg Sat. Vapor hg Sat. Liquid si /sf Evap. sig /sfg Sat. Vapor sg Temp., °C t 5.0387 4.8392 4.6488 4.4671 4.2937 4.1281 3.9699 3.8188 3.6743 3.5363 3.4042 3.2780 3.1572 3.0415 2.9309 2.8249 2.7234 2.6262 2.5330 2.4437 2.3581 2.2760 2.1973 2.1217 2.0492 1.9796 1.9128 1.8486 1.7870 1.7277 1.6708 1.6161 1.5635 1.5129 1.4642 1.4174 1.3724 1.3290 1.2873 1.2471 1.2083 1.1710 1.1351 1.1005 1.0671 1.0349 1.0038 0.9739 0.9450 0.9171 0.8902 0.8392 0.7916 0.7472 0.7058 0.6670 0.6308 0.5969 0.5651 0.5353 0.5074 0.4813 0.4567 0.4336 0.4118 0.3914 0.3722 0.3541 0.3370 0.3209 0.3057 5.0397 4.8402 4.6498 4.4681 4.2947 4.1291 3.9709 3.8198 3.6754 3.5373 3.4053 3.2790 3.1582 3.0426 2.9319 2.8259 2.7244 2.6272 2.5341 2.4448 2.3591 2.2771 2.1983 2.1228 2.0502 1.9806 1.9138 1.8497 1.7880 1.7288 1.6719 1.6171 1.5645 1.5140 1.4653 1.4185 1.3734 1.3301 1.2883 1.2481 1.2094 1.1721 1.1362 1.1015 1.0681 1.0359 1.0049 0.9750 0.9461 0.9182 0.8913 0.8403 0.7927 0.7483 0.7068 0.6681 0.6318 0.5979 0.5662 0.5364 0.5085 0.4823 0.4577 0.4346 0.4129 0.3925 0.3733 0.3552 0.3381 0.3220 0.3068 293.02 297.21 301.40 305.59 309.78 313.97 318.17 322.36 326.56 330.75 334.95 339.15 343.34 347.54 351.74 355.95 360.15 364.35 368.56 372.76 376.97 381.18 385.38 389.59 393.81 398.02 402.23 406.45 410.66 414.88 419.10 423.32 427.54 431.76 435.99 440.21 444.44 448.67 452.90 457.13 461.36 465.60 469.83 474.07 478.31 482.55 486.80 491.04 495.29 499.53 503.78 512.29 520.80 529.32 537.85 546.39 554.93 563.49 572.05 580.62 589.20 597.79 606.39 615.00 623.62 632.25 640.89 649.55 658.21 666.89 675.57 2333.08 2330.60 2328.11 2325.62 2323.13 2320.63 2318.13 2315.62 2313.11 2310.59 2308.07 2305.54 2303.01 2300.47 2297.93 2295.38 2292.83 2290.27 2287.70 2285.14 2282.56 2279.98 2277.39 2274.80 2272.20 2269.60 2266.98 2264.37 2261.74 2259.11 2256.47 2253.83 2251.18 2248.52 2245.85 2243.18 2240.50 2237.81 2235.12 2232.41 2229.70 2226.99 2224.26 2221.53 2218.78 2216.03 2213.27 2210.51 2207.73 2204.94 2202.15 2196.53 2190.88 2185.19 2179.47 2173.70 2167.89 2162.04 2156.15 2150.22 2144.24 2138.22 2132.15 2126.04 2119.88 2113.67 2107.41 2101.10 2094.74 2088.32 2081.86 2626.10 2627.81 2629.51 2631.21 2632.91 2634.60 2636.29 2637.98 2639.66 2641.34 2643.01 2644.68 2646.35 2648.01 2649.67 2651.33 2652.98 2654.62 2656.26 2657.90 2659.53 2661.16 2662.78 2664.39 2666.01 2667.61 2669.22 2670.81 2672.40 2673.99 2675.57 2677.15 2678.72 2680.28 2681.84 2683.39 2684.94 2686.48 2688.02 2689.55 2691.07 2692.58 2694.09 2695.60 2697.09 2698.58 2700.07 2701.55 2703.02 2704.48 2705.93 2708.82 2711.69 2714.52 2717.32 2720.09 2722.83 2725.53 2728.20 2730.84 2733.44 2736.01 2738.54 2741.04 2743.50 2745.92 2748.30 2750.64 2752.95 2755.21 2757.43 0.9550 0.9672 0.9793 0.9915 1.0035 1.0156 1.0276 1.0396 1.0516 1.0635 1.0754 1.0873 1.0991 1.1109 1.1227 1.1344 1.1461 1.1578 1.1694 1.1811 1.1927 1.2042 1.2158 1.2273 1.2387 1.2502 1.2616 1.2730 1.2844 1.2957 1.3070 1.3183 1.3296 1.3408 1.3520 1.3632 1.3743 1.3854 1.3965 1.4076 1.4187 1.4297 1.4407 1.4517 1.4626 1.4735 1.4844 1.4953 1.5062 1.5170 1.5278 1.5494 1.5708 1.5922 1.6134 1.6346 1.6557 1.6767 1.6977 1.7185 1.7393 1.7600 1.7806 1.8011 1.8216 1.8420 1.8623 1.8825 1.9027 1.9228 1.9428 6.7990 6.7720 6.7452 6.7185 6.6920 6.6656 6.6393 6.6132 6.5872 6.5613 6.5356 6.5100 6.4846 6.4592 6.4340 6.4090 6.3840 6.3592 6.3345 6.3099 6.2854 6.2611 6.2368 6.2127 6.1887 6.1648 6.1411 6.1174 6.0938 6.0704 6.0471 6.0238 6.0007 5.9777 5.9548 5.9320 5.9092 5.8866 5.8641 5.8417 5.8194 5.7972 5.7750 5.7530 5.7310 5.7092 5.6874 5.6658 5.6442 5.6227 5.6013 5.5587 5.5165 5.4746 5.4330 5.3918 5.3508 5.3102 5.2698 5.2298 5.1900 5.1505 5.1112 5.0723 5.0335 4.9951 4.9569 4.9189 4.8811 4.8436 4.8063 7.7540 7.7392 7.7245 7.7100 7.6955 7.6812 7.6669 7.6528 7.6388 7.6248 7.6110 7.5973 7.5837 7.5701 7.5567 7.5434 7.5301 7.5170 7.5039 7.4909 7.4781 7.4653 7.4526 7.4400 7.4275 7.4150 7.4027 7.3904 7.3782 7.3661 7.3541 7.3421 7.3303 7.3185 7.3068 7.2951 7.2836 7.2721 7.2607 7.2493 7.2380 7.2268 7.2157 7.2047 7.1937 7.1827 7.1719 7.1611 7.1504 7.1397 7.1291 7.1081 7.0873 7.0668 7.0465 7.0264 7.0066 6.9869 6.9675 6.9483 6.9293 6.9105 6.8918 6.8734 6.8551 6.8370 6.8191 6.8014 6.7838 6.7664 6.7491 70 71 72 73 74 75 76 77 78 79 80 81 82 83 84 85 86 87 88 89 90 91 92 93 94 95 96 97 98 99 100 101 102 103 104 105 106 107 108 109 110 111 112 113 114 115 116 117 118 119 120 122 124 126 128 130 132 134 136 138 140 142 144 146 148 150 152 154 156 158 160 This file is licensed to John Murray (). Publication Date: 6/1/2017 1.8 2017 ASHRAE Handbook—Fundamentals (SI) The saturation (sublimation) pressure over ice for the temperature range of 100 to 0°C is given by ln pws = C1/T + C2 + C3T + C4T 2 + C5T 3 + C6T 4 + C7 ln T (5) dv = Mw /V Density of a moist air mixture is the ratio of total mass to total volume: where C1 C2 C3 C4 C5 C6 C7 = = = = = = = (Mda + Mw)/V = (1/v)(1 + W) –5.674 535 9 E+03 6.392 524 7 E+00 –9.677 843 0 E–03 6.221 570 1 E–07 2.074 782 5 E–09 –9.484 024 0 E–13 4.163 501 9 E00 Humidity Parameters Involving Saturation ln pws = C8/T + C9 + C10T + C11T 2 + C12T 3 + C13 ln T (6) Licensed for single user. © 2017 ASHRAE, Inc. where = = = = = = –5.800 220 6 E+03 1.391 499 3 E+00 –4.864 023 9 E–02 4.176 476 8 E–05 –1.445 209 3 E–08 6.545 967 3 E+00 The following definitions of humidity parameters involve the concept of moist air saturation: Saturation humidity ratio Ws(t, p) is the humidity ratio of moist air saturated with respect to water (or ice) at the same temperature t and pressure p. Relative humidity is the ratio of the actual water vapor partial pressure in moist air at the dew-point pressure and temperature to the reference saturation water vapor partial pressure at the dry-bulb pressure and temperature: = (pwv _ enh /pwvs _ ref |p,t) = [f(p, tdp)e(tdp)]/[f(p, tdb)e(tdb)] (12) In both Equations (5) and (6), pws = saturation pressure, Pa T = absolute temperature, K = °C + 273.15 The coefficients of Equations (5) and (6) were derived from the Hyland-Wexler equations. Because of rounding errors in the derivations and in some computers' calculating precision, results from Equations (5) and (6) may not agree precisely with Table 3 values. The vapor pressure ps of water in saturated moist air differs negligibly from the saturation vapor pressure pws of pure water at the same temperature. Consequently, ps can be used in equations in place of pws with very little error: ps = xws p where xws is the mole fraction of water vapor in saturated moist air at temperature t and pressure p, and p is the total barometric pressure of moist air. 5. (11) where v is the moist air specific volume, m3/kgda, as defined by Equation (24). The saturation pressure over liquid water for the temperature range of 0 to 200°C is given by C8 C9 C10 C11 C12 C13 (10) Note that Equations (12) and (22) have been revised so that they cover both the normal range of relative humidity where e(tdb) p and the extended range (e.g., atmospheric pressure drying kilns) where e(tdb) p. The definitions in earlier editions applied only to the normal range. Dew-point temperature td is the temperature of moist air saturated at pressure p, with the same humidity ratio W as that of the given sample of moist air. It is defined as the solution td (p, W) of the following equation: Ws (p, td) = W Thermodynamic wet-bulb temperature t* is the temperature at which water (liquid or solid), by evaporating into moist air at drybulb temperature t and humidity ratio W, can bring air to saturation adiabatically at the same temperature t* while total pressure p is constant. This parameter is considered separately in the section on Thermodynamic Wet-Bulb and Dew-Point Temperature. 6. HUMIDITY PARAMETERS Basic Parameters Humidity ratio W (or mixing ratio) of a given moist air sample is defined as the ratio of the mass of water vapor to the mass of dry air in the sample: W = Mw /Mda W = 0.621 945xw /xda Water vapor: pwV = nw RT (14) (15) where pda pw V nda nw R T (8) (9a) In terms of the humidity ratio, = W/(1 + W) pdaV = nda RT Dry air: Specific humidity is the ratio of the mass of water vapor to total mass of the moist air sample: = Mw /(Mw + Mda) PERFECT GAS RELATIONSHIPS FOR DRY AND MOIST AIR When moist air is considered a mixture of independent perfect gases (i.e., dry air and water vapor), each is assumed to obey the perfect gas equation of state as follows: (7) W equals the mole fraction ratio xw /xda multiplied by the ratio of molecular masses (18.015 268/28.966 = 0.621 945): (13) = = = = = = = partial pressure of dry air partial pressure of water vapor total mixture volume number of moles of dry air number of moles of water vapor universal gas constant, 8314.472 J/(kmol·K) absolute temperature, K The mixture also obeys the perfect gas equation: (9b) Absolute humidity (alternatively, water vapor density) dv is the ratio of the mass of water vapor to total volume of the sample: pV = nRT (16) (pda + pw)V = (nda + nw)RT (17) or This file is licensed to John Murray (). Publication Date: 6/1/2017 Psychrometrics 1.9 where p = pda + pw is the total mixture pressure and n = nda + nw is the total number of moles in the mixture. From Equations (14) to (17), the mole fractions of dry air and water vapor are, respectively, xda = pda /(pda + pw) = pda /p (18) xw = pw /(pda + pw) = pw /p (19) and The enthalpy of a mixture of perfect gases equals the sum of the individual partial enthalpies of the components. Therefore, the specific enthalpy of moist air can be written as follows: h = hda + Whg where hda is the specific enthalpy for dry air in kJ/kgda and hg is the specific enthalpy for saturated water vapor in kJ/kgw at the mixture's temperature. As an approximation, From Equations (8), (18), and (19), the humidity ratio W is pw W = 0.621 945 ---------------p – pw (20) Licensed for single user. © 2017 ASHRAE, Inc. 7. (21) (22) Substituting Equation (21) for Ws into Equation (13),   = ---1 – 1 –   p ws p (23) Both and are zero for dry air and unity for saturated moist air. At intermediate states, their values differ, substantially at higher temperatures. The specific volume v of a moist air mixture is expressed in terms of a unit mass of dry air: v = V/Mda = V/(28.966nda) (24) where V is the total volume of the mixture, Mda is the total mass of dry air, and nda is the number of moles of dry air. By Equations (14) and (24), with the relation p = pda + pw , R da T RT v = ------------------------------------- = ---------------p – pw 28.966 p – p w (25) Using Equation (18), R da T 1 + 1.607 858 W RT 1 + 1.607 858 W v = -- = --p 28.966p (26) In Equations (25) and (26), v is specific volume, T is absolute temperature, p is total pressure, pw is partial pressure of water vapor, and W is humidity ratio. In specific units, Equation (26) may be expressed as v = 0.287 042(t + 273.15)(1 + 1.607 858W)/p where v t W p = = = = specific volume, m3/kgda dry-bulb temperature, °C humidity ratio, kgw /kgda total pressure, kPa (28) hg 2501 + 1.86t (29) h = 1.006t + W(2501 + 1.86t) The term pws represents the saturation pressure of water vapor in the absence of air at the given temperature t. This pressure pws is a function only of temperature and differs slightly from the vapor pressure of water in saturated moist air. The relative humidity is defined in Equation (12). Using the second equality and eliminating the enhancement factors, which are not applicable using the perfect gas assumption, gives = e(tdp)/e(tdb) hda 1.006t where t is the dry-bulb temperature in °C. The moist air specific enthalpy in kJ/kg da then becomes The saturation humidity ratio Ws is p ws Ws = 0.621 945 ---------------p – p ws (27) (30) THERMODYNAMIC WET-BULB AND DEW-POINT TEMPERATURE For any state of moist air, a temperature t* exists at which liquid (or solid) water evaporates into the air to bring it to saturation at exactly this same temperature and total pressure (Harrison 1965). During adiabatic saturation, saturated air is expelled at a temperature equal to that of the injected water. In this constant-pressure process, • Humidity ratio increases from initial value W to Ws*, corresponding to saturation at temperature t* • Enthalpy increases from initial value h to hs*, corresponding to saturation at temperature t* • Mass of water added per unit mass of dry air is (Ws* – W), which adds energy to the moist air of amount (Ws* – W)hw*, where hw* denotes specific enthalpy in kJ/kgw of water added at temperature t* Therefore, if the process is strictly adiabatic, conservation of enthalpy at constant total pressure requires that h + (Ws* – W)hw* = hs* (31) Ws*, hw*, and hs* are functions only of temperature t* for a fixed value of pressure. The value of t* that satisfies Equation (31) for given values of h, W, and p is the thermodynamic wet-bulb temperature. A psychrometer consists of two thermometers; one thermometer's bulb is covered by a wick that has been thoroughly wetted with water. When the wet bulb is placed in an airstream, water evaporates from the wick, eventually reaching an equilibrium temperature called the wet-bulb temperature. This process is not one of adiabatic saturation, which defines the thermodynamic wet-bulb temperature, but one of simultaneous heat and mass transfer from the wet bulb. The fundamental mechanism of this process is described by the Lewis relation [Equation (40) in Chapter 6]. Fortunately, only small corrections must be applied to wet-bulb thermometer readings to obtain the thermodynamic wet-bulb temperature. As defined, thermodynamic wet-bulb temperature is a unique property of a given moist air sample independent of measurement techniques. Equation (31) is exact because it defines the thermodynamic wetbulb temperature t*. Substituting the approximate perfect gas relation [Equation (30)] for h, the corresponding expression for hs*, and the approximate relation for saturated liquid water h*w 4.186t* (32) into Equation (31), and solving for the humidity ratio, 2501 – 2.326t* W*s – 1.006 t – t* W = --2501 + 1.86t – 4.186t* (33) This file is licensed to John Murray (). Publication Date: 6/1/2017 1.10 2017 ASHRAE Handbook—Fundamentals (SI) where t and t* are in °C. Below freezing, the corresponding equations are h*w –333.4 + 2.1t* (34) 2830 – 0.24t* W*s – 1.006 t – t* W = ---2830 + 1.86t – 2.1t* (35) A wet/ice-bulb thermometer is imprecise when determining moisture content at 0°C. The dew-point temperature td of moist air with humidity ratio W and pressure p was defined as the solution td (p, W) of Ws (p, td). For perfect gases, this reduces to pws(td) = pw = (pW)/(0.621 945 + W) Licensed for single user. © 2017 ASHRAE, Inc. (37) Below 0°C, td = 6.09 + 12.608 + 0.49592 To Obtain Use Comments pw = pws (td) W pws (t) Ws v h t* Table 3 or Equation (5) or (6) Equation (20) Table 3 or Equation (5) or (6) Equation (21) Equation (23) Equation (26) Equation (30) Equation (21) and (33) or (35) with Table 3 or with Equation (5) or (6) Sat. press. for temp. td (36) where pw is the water vapor partial pressure for the moist air sample and pws(td) is the saturation vapor pressure at temperature td . The saturation vapor pressure is obtained from Table 3 or by using Equation (5) or (6). Alternatively, the dew-point temperature can be calculated directly by one of the following equations (Peppers 1988): Between dew points of 0 and 93°C, td = C14 + C15 + C162 + C173 + C18 (pw)0.1984 SITUATION 2. Given: Dry-bulb temperature t, Dew-point temperature td , Pressure p (38) 8. = = = = = = = = dew-point temperature, °C ln pw water vapor partial pressure, kPa 6.54 14.526 0.7389 0.09486 0.4569 Requires trial-and-error or numerical solution method SITUATION 3. Given: Dry-bulb temperature t, Relative humidity Pressure p To Obtain pws(t) pw W Ws v h td t* where td pw C14 C15 C16 C17 C18 Sat. press. for temp. t Using pws (t) Using pws (t) Use Comments Table 3 or Equation (5) or (6) Equation (22) Equation (20) Equation (21) Equation (26) Equation (30) Table 3 with Equation (36), (37), or (38) Equation (21) and (33) or (35) with Table 3 or with Equation (5) or (6) Sat. press. for temp. t Using pws (t) Requires trial-and-error or numerical solution method Moist Air Property Tables for Standard Pressure Table 2 shows thermodynamic properties for standard atmospheric pressure at temperatures from –60 to 90°C calculated using the ASHRAE RP-1485 (Herrmann et al. 2009) research project numerical model. Properties of intermediate moist air states can be calculated using the degree of saturation : NUMERICAL CALCULATION OF MOIST AIR PROPERTIES The following are outlines, citing equations and tables already presented, for calculating moist air properties using perfect gas relations. These relations are accurate enough for most engineering calculations in air-conditioning practice, and are readily adapted to either hand or computer calculating methods. For more details, refer to Tables 15 through 18 in Chapter 1 of Olivieri (1996). Graphical procedures are discussed in the section on Psychrometric Charts. SITUATION 1. Given: Dry-bulb temperature t, Wet-bulb temperature t*, Pressure p To Obtain Use Comments pws (t*) W s* W pws (t) Ws v h pw td Table 3 or Equation (5) or (6) Sat. press. for temp. t* Equation (21) Using pws (t*) Equation (33) or (35) Table 3 or Equation (5) or (6) Sat. press. for temp. t Equation (21) Using pws (t) Equation (23) Using pws (t) Equation (26) Equation (30) Equation (36) Table 3 with Equation (36), (37), or (38) Volume v = vda + vas (39) Enthalpy h = hda + has (40) These equations are accurate to about 350°C. At higher temperatures, errors can be significant. 9. PSYCHROMETRIC CHARTS A psychrometric chart graphically represents the thermodynamic properties of moist air. The choice of coordinates for a psychrometric chart is arbitrary. A chart with coordinates of enthalpy and humidity ratio provides convenient graphical solutions of many moist air problems with a minimum of thermodynamic approximations. ASHRAE developed five such psychrometric charts. Chart 1 is shown as Figure 1; the others may be obtained through ASHRAE. Charts 1, 2, 3 and 4 are for sea-level pressure (101.325 kPa). Chart 5 is for 750 m altitude (92.634 kPa), Chart 6 is for 1500 m altitude (84.54 kPa), and Chart 7 is for 2250 m altitude (77.058 kPa). All charts use oblique-angle coordinates of enthalpy and humidity ratio, and are consistent with the data of Table 2 and the properties computation methods of Hyland and Wexler (1983a) and ASHRAE research project RP-1485. Palmatier (1963) describes the geometry of chart construction applying specifically to Charts 1 and 4. The dry-bulb temperature ranges covered by the charts are Charts 1, 5, 6, 7 Chart 2 Normal temperature Low temperature 0 to 50°C –40 to 10°C This file is licensed to John Murray (). Publication Date: 6/1/2017 1.11 Licensed for single user. © 2017 ASHRAE, Inc. Psychrometrics Fig. 1 ASHRAE Psychrometric Chart No. 1 This file is licensed to John Murray (). Publication Date: 6/1/2017 1.12 2017 ASHRAE Handbook—Fundamentals (SI) Chart 3 Chart 4 High temperature Very high temperature 10 to 120°C 100 to 200°C Charts 8 to 16 are for 200 to 320°C and cover the same pressures as 1, 5, 6, and 7 plus the additional pressures of 0.2, 0.5,1.0, 2.0, and 5.0 MPa. They were produced by Nelson and Sauer (2002) and are available as a download with Gatley (2013). Psychrometric properties or charts for other barometric pressures can be derived by interpolation. Sufficiently exact values for most purposes can be derived by methods described in the section on Perfect Gas Relationships for Dry and Moist Air. Constructing charts for altitude conditions has been discussed by Haines (1961), Karig (1946), and

Rohsenow (1946). Comparison of charts 1 and 6 by overlay reveals the following: Licensed for single user. © 2017 ASHRAE, Inc. • The dry-bulb lines coincide. • Wet-bulb lines for a given temperature originate at the intersections of the corresponding dry-bulb line and the two saturation curves, and they have the same slope. • Humidity ratio and enthalpy for a given dry- and wet-bulb temperature increase with altitude, but there is little change in relative humidity. • Volume changes rapidly; for a given dry-bulb and humidity ratio, it is practically inversely proportional to barometric pressure. The following table compares properties at sea level (chart 1) and 1500 m (chart 6): Chart No. db wb h W rh v 1 6 40 40 30 30 99.5 114.1 23.0 28.6 49 50 0.920 1.111 Figure 1 shows humidity ratio lines (horizontal) for the range from 0 (dry air) to 30 grams moisture per kilogram dry air. Enthalpy lines are oblique lines across the chart precisely parallel to each other. Dry-bulb temperature lines are straight, not precisely parallel to each other, and inclined slightly from the vertical position. Thermodynamic wet-bulb temperature lines are oblique and in a slightly different direction from enthalpy lines. They are straight but are not precisely parallel to each other. Relative humidity lines are shown in intervals of 10%. The saturation curve is the line of 100% rh, whereas the horizontal line for W = 0 (dry air) is the line for 0% rh. Specific volume lines are straight but are not precisely parallel to each other. A narrow region above the saturation curve has been developed for fog conditions of moist air. This two-phase region represents a mechanical mixture of saturated moist air and liquid water, with the two components in thermal equilibrium. Isothermal lines in the fog region coincide with extensions of thermodynamic wet-bulb temperature lines. If required, the fog region can be further expanded by extending humidity ratio, enthalpy, and thermodynamic wet-bulb temperature lines. The protractor to the left of the chart shows two scales: one for sensible/total heat ratio, and one for the ratio of enthalpy difference to humidity ratio difference. The protractor is used to establish the direction of a condition line on the psychrometric chart. Example 1 shows use of the ASHRAE psychrometric chart to determine moist air properties. Example 1. Moist air exists at 40°C dry-bulb temperature, 20°C thermodynamic wet-bulb temperature, and 101.325 kPa pressure. Determine the humidity ratio, enthalpy, dew-point temperature, relative humidity, and specific volume. Solution: Locate state point on chart 1 (Figure 1) at the intersection of 40°C dry-bulb temperature and 20°C thermodynamic wet-bulb temperature lines. Read humidity ratio W = 6.5 gw /kgda. The enthalpy can be found by using two triangles to draw a line parallel to the nearest enthalpy line (60 kJ/kgda) through the state point to the nearest edge scale. Read h = 56.7 kJ/kgda. Dew-point temperature can be read at the intersection of W = 6.5 gw /kgda with the saturation curve. Thus, td = 7°C. Relative humidity can be estimated directly. Thus, = 14%. Specific volume can be found by linear interpolation between the volume lines for 0.88 and 0.90 m3/kgda. Thus, v = 0.896 m3/kgda. 10. TYPICAL AIR-CONDITIONING PROCESSES The ASHRAE psychrometric chart can be used to solve numerous process problems with moist air. Its use is best explained through illustrative examples. In each of the following examples, the process takes place at a constant total pressure of 101.325 kPa. Moist Air Sensible Heating or Cooling Adding heat alone to or removing heat alone from moist air is represented by a horizontal line on the ASHRAE chart, because the humidity ratio remains unchanged. Figure 2 shows a device that adds heat to a stream of moist air. For steady-flow conditions, the required rate of heat addition is 1q2 = m· da (h2 – h1) (41) Example 2. Moist air, saturated at 2°C, enters a heating coil at a rate of 10 m3/s. Air leaves the coil at 40°C. Find the required rate of heat addition. Solution: Figure 3 schematically shows the solution. State 1 is located on the saturation curve at 2°C. Thus, h1 = 13.0 kJ/kgda, W1 = 4.38 gw /kgda, and v1 = 0.785 m3/kgda. State 2 is located at the intersection of t = 40°C and W2 = W1 = 4.38 g wT/kgda. Thus, h2 = 51.5 kJ/kgda. The mass flow of dry air is m· da = 10/0.785 = 12.74 kgda/s Fig. 2 Schematic of Device for Heating Moist Air Fig. 3 Schematic Solution for Example 2 This file is licensed to John Murray (). Publication Date: 6/1/2017 Psychrometrics 1.13 Fig. 4 Schematic of Device for Cooling Moist Air From Equation (41), 1q2 Fig. 5 Schematic Solution for Example 3 = 12.74(51.5 – 13.0) = 490 kW Licensed for single user. © 2017 ASHRAE, Inc. Moist Air Cooling and Dehumidification Moisture condensation occurs when moist air is cooled to a temperature below its initial dew point. Figure 4 shows a schematic cooling coil where moist air is assumed to be uniformly processed. Although water can be removed at various temperatures ranging from the initial dew point to the final saturation temperature, it is assumed that condensed water is cooled to the final air temperature t2 before it drains from the system. For the system in Figure 4, the steady-flow energy and material balance equations are m· da h 1 = m· da h 2 + 1q2 + m· w h w2 m· da W 1 = m· da W 2 + m· w Fig. 6 Adiabatic Mixing of Two Moist Airstreams Thus, m· w = m· da (W1 – W2) 1q2 = m· da [(h1 – h2) – (W1 – W2)hw2] (42) (43) Example 3. Moist air at 30°C dry-bulb temperature and 50% rh enters a cooling coil at 5 m3/s and is processed to a final saturation condition at 10°C. Find the kW of refrigeration required. Solution: Figure 5 shows the schematic solution. State 1 is located at the intersection of t = 30°C and = 50%. Thus, h1 = 64.3 kJ/kgda, W1 = 13.3 gw /kgda, and v1 = 0.877 m3/kgda. State 2 is located on the saturation curve at 10°C. Thus, h2 = 29.5 kJ/kgda and W2 = 7.66 gw /kgda. From Table 3, hw2 = 42.02 kJ/kgw . The mass flow of dry air is m· da = 5/0.877 = 5.70 kgda/s From Equation (43), 1q2 = 5.70[(64.3 – 29.5) – (0.0133 – 0.00766)42.02] = 197 kW Adiabatic Mixing of Two Moist Airstreams A common process in air-conditioning systems is the adiabatic mixing of two moist airstreams. Figure 6 schematically shows the problem. Adiabatic mixing is governed by three equations: m· da1 h 1 + m· da2 h 2 = m· da3 h 3 m· da1 + m· da2 = m· da3 W + m· W = m· W m· da1 1 da2 2 da3 3 Eliminating m· da3 gives h2 – h3 W2 – W3 m· da1 ---------------- = -------------------- = ----------m· da2 h3 – h1 W3 – W1 (44) according to which, on the ASHRAE chart, the state point of the resulting mixture lies on the straight line connecting the state points of the two streams being mixed, and divides the line into two segments, in the same ratio as the masses of dry air in the two streams. Example 4. A stream of 2 m3/s of outdoor air at 4°C dry-bulb temperature and 2°C thermodynamic wet-bulb temperature is adiabatically mixed with 6.25 m3/s of recirculated air at 25°C dry-bulb temperature and 50% rh. Find the dry-bulb temperature and thermodynamic wet-bulb temperature of the resulting mixture. Solution: Figure 7 shows the schematic solution. States 1 and 2 are located on the ASHRAE chart: v1 = 0.789 m3/kgda, and v2 = 0.858 m3/kgda. Therefore, m· da1 = 2/0.789 = 2.535 kgda /s m· da2 = 6.25/0.858 = 7.284 kgda /s This file is licensed to John Murray (). Publication Date: 6/1/2017 Licensed for single user. © 2017 ASHRAE, Inc. 1.14 2017 ASHRAE Handbook—Fundamentals (SI) Fig. 9 Fig. 7 Schematic Solution for Example 4 Schematic Solution for Example 5 specific enthalpy of the injected water, drawn through the initial state point of the moist air. Example 5. Moist air at 20°C dry-bulb and 8°C thermodynamic wet-bulb temperature is to be processed to a final dew-point temperature of 13°C by adiabatic injection of saturated steam at 110°C. The rate of dry airflow m· da is 2 kgda /s. Find the final dry-bulb temperature of the moist air and the rate of steam flow. Solution: Figure 9 shows the schematic solution. By Table 3, the enthalpy of the steam hg = 2691.07 kJ/kgw . Therefore, according to Equation (45), the condition line on the ASHRAE chart connecting states 1 and 2 must have a direction: h/W = 2.691 kJ/gw Fig. 8 Schematic Showing Injection of Water into Moist Air According to Equation (44), m· da2 m· da1 Line 1–3 7.284 Line 3–2 -------------------- = ----------- or -------------------- = ----------- = ------------- = 0.742 · m da2 m· da3 Line 1–2 9.819 Line 1–3 Consequently, the length of line segment 1-3 is 0.742 times the length of entire line 1-2. Using a ruler, state 3 is located, and the values t3 = 19.5°C and t3* = 14.6°C found. The condition line can be drawn with the h/W protractor. First, establish the reference line on the protractor by connecting the origin with the value h/W = 2.691 kJ/gw . Draw a second line parallel to the reference line and through the initial state point of the moist air. This second line is the condition line. State 2 is established at the intersection of the condition line with the horizontal line extended from the saturation curve at 13°C (td2 = 13°C). Thus, t2 = 21°C. Values of W2 and W1 can be read from the chart. The required steam flow is m· w = m· da (W2 – W1) = 2 1000(0.0093 – 0.0018) = 15.0 gw /s, or 0.015 kgw /s Adiabatic Mixing of Water Injected into Moist Air Steam or liquid water can be injected into a moist airstream to raise its humidity, as shown in Figure 8. If mixing is adiabatic, the following equations apply: m· da h 1 + m· w h w = m· da h 2 m· da W 1 + m· w = m· da W 2 Therefore, h2 – h1 h -------------------- = -------- = hw W2 – W1 W (45) according to which, on the ASHRAE chart, the final state point of the moist air lies on a straight line in the direction fixed by the Space Heat Absorption and Moist Air Moisture Gains Air conditioning required for a space is usually determined by (1) the quantity of moist air to be supplied, and (2) the supply air condition necessary to remove given amounts of energy and water from the space at the exhaust condition specified. Figure 10 shows a space with incident rates of energy and moisture gains. The quantity qs denotes the net sum of all rates of heat gain in the space, arising from transfers through boundaries and from sources within the space. This heat gain involves energy addition alone and does not include energy contributions from water (or water vapor) addition. It is usually called the sensible heat gain. The quantity m· w denotes the net sum of all rates of moisture gain on the space arising from transfers through boundaries and from sources within the space. Each kilogram of water This file is licensed to John Murray (). Publication Date: 6/1/2017 Psychrometrics 1.15 Fig. 10 Schematic of Air Conditioned Space vapor added to the space adds an amount of energy equal to its specific enthalpy. Assuming steady-state conditions, governing equations are m· h + q + m· h = m· h Licensed for single user. © 2017 ASHRAE, Inc. da 1 or m· da W 1 + m· w = m· da W 2 s Fig. 11 da 2 w w q s + m· w h w = m· da (h2 – h1) scale instead of the h/W scale. The quantity Hs /Ht is the ratio of rate of sensible heat gain for the space to rate of total energy gain for the space. Therefore, H s qs 9 --------- = --------------------------------- = -- = 0.701 H t q s + m· w h w 9 + 0.0015 2555.58 (46) · · mw = mda (W2 – W1) (47) Note that Hs /Ht = 0.701 on the protractor coincides closely with h/W = 8.555 kJ/gw . The flow of dry air can be calculated from either Equation (46) or (47). From Equation (46), The left side of Equation (46) represents the total rate of energy addition to the space from all sources. By Equations (46) and (47), q s + m· w h w h2 – h1 h -------------------- = -------- = -------------------------------------W2 – W1 W m· Schematic Solution for Example 6 q s + m· w h w 9 + 0.0015 2555.58 m· da = --------------------------------------- = --h2 – h1 53.9 – 39.2 = 0.873 kg s (48) w according to which, on the ASHRAE chart and for a given state of withdrawn air, all possible states (conditions) for supply air must lie on a straight line drawn through the state point of withdrawn air, with its direction specified by the numerical value of q s + m· w h w m· w . This line is the condition line for the given problem. Example 6. Moist air is withdrawn from a room at 25°C dry-bulb temperature and 19°C thermodynamic wet-bulb temperature. The sensible rate of heat gain for the space is 9 kW. A rate of moisture gain of 0.0015 kgw /s occurs from the space occupants. This moisture is assumed as saturated water vapor at 30°C. Moist air is introduced into the room at a dry-bulb temperature of 15°C. Find the required thermodynamic wet-bulb temperature and volume flow rate of the supply air. Solution: Figure 11 shows the schematic solution. State 2 is located on the ASHRAE chart. From Table 3, the specific enthalpy of the added water vapor is hg = 2555.58 kJ/kgw . From Equation (48), 9 + 0.0015 2555.58 h -------- = --- = 8555 kJ/kgw W 0.0015 With the h/W protractor, establish a reference line of direction h/W = 8.555 kJ/gw . Parallel to this reference line, draw a straight line on the chart through state 2. The intersection of this line with the 15°C dry-bulb temperature line is state 1. Thus, t1* = 14.0°C. An alternative (and approximately correct) procedure in establishing the condition line is to use the protractor's sensible/total heat ratio At state 1, v1 = 0.829 m3/kgda. Therefore, supply volume = m· da v 1 = 0.873 0.829 = 0.724 m3/s. 11. TRANSPORT PROPERTIES OF MOIST AIR For certain scientific and experimental work, particularly in the heat transfer field, many other moist air properties are important. Generally classified as transport properties, these include diffusion coefficient, viscosity, thermal conductivity, and thermal diffusion factor. Mason and Monchick (1965) derive these properties by calculation. Table 4 and Figures 12 and 13 summarize the authors' results on the first three properties listed. Note that, within the boundaries of ASHRAE psychrometric charts 1, 2, and 3, viscosity varies little from that of dry air at normal atmospheric pressure, and thermal conductivity is essentially independent of moisture content. 12. SYMBOLS C1 to C18 = constants in Equations (5), (6), and (37) dv = absolute humidity of moist air, mass of water per unit volume of mixture, kgw /m3 h = specific enthalpy of moist air, kJ/kgda Hs = rate of sensible heat gain for space, kW hs* = specific enthalpy of saturated moist air at thermodynamic wetbulb temperature, kJ/kgda Ht = rate of total energy gain for space, kW This file is licensed to John Murray (). Publication Date: 6/1/2017 1.16 2017 ASHRAE Handbook—Fundamentals (SI) Table 4 Calculated Diffusion Coefficients for Water/Air at 101.325 kPa Temp., °C mm2/s Temp., °C mm2/s Temp., °C mm2/s –70 –50 –40 –35 –30 –25 –20 –15 –10 –5 13.2 15.6 16.9 17.5 18.2 18.8 19.5 20.2 20.8 21.5 0 5 10 15 20 25 30 35 40 45 22.2 22.9 23.6 24.3 25.1 25.8 26.5 27.3 28.0 28.8 50 55 60 70 100 130 160 190 220 250 29.5 30.3 31.1 32.7 37.6 42.8 48.3 54.0 60.0 66.3 Licensed for single user. © 2017 ASHRAE, Inc. Fig. 13 Thermal Conductivity of Moist Air V v vT W W s* = = = = = xda = xw = xws = Z = total volume of moist air sample, m3 specific volume, m3/kgda or m3/kgw total gas volume, m3 humidity ratio of moist air, kgw /kgda or gw /kgda humidity ratio of moist air at saturation at thermodynamic wet-bulb temperature, kgw /kgda or gw /kgda mole fraction of dry air, moles of dry air per mole of mixture mole fraction of water, moles of water per mole of mixture mole fraction of water vapor under saturated conditions, moles of vapor per mole of saturated mixture altitude, m Greek Fig. 12 Viscosity of Moist Air hw* = specific enthalpy of condensed water (liquid or solid) at thermodynamic wet-bulb temperature and a pressure of 101.325 kPa, kJ/kgw Mda = mass of dry air in moist air sample, kgda m· da = mass flow of dry air, per unit time, kgda /s Mw = mass of water vapor in moist air sample, kgw m· w = mass flow of water (any phase), per unit time, kgw /s n = nda + nw , total number of moles in moist air sample nda = moles of dry air nw = moles of water vapor p = total pressure of moist air, kPa pda = partial pressure of dry air, kPa ps = vapor pressure of water in moist air at saturation, kPa. Differs slightly from saturation pressure of pure water because of presence of air. pw = partial pressure of water vapor in moist air, kPa pws = pressure of saturated pure water, kPa qs = rate of addition (or withdrawal) of sensible heat, kW R = universal gas constant, 8314.472 J/(kg mole·K) Rda = gas constant for dry air, kJ/(kgda ·K) Rw = gas constant for water vapor, kJ/(kgw ·K) s = specific entropy, kJ/(kgda ·K) or kJ/(kgw ·K) T = absolute temperature, K t = dry-bulb temperature of moist air, °C td = dew-point temperature of moist air, °C t* = thermodynamic wet-bulb temperature of moist air, °C = ln(pw), parameter used in Equations (37) and (38) = specific humidity of moist air, mass of water per unit mass of mixture = moist air density = relative humidity, dimensionless Subscripts as da f fg = = = = g i ig s t w = = = = = = difference between saturated moist air and dry air dry air saturated liquid water difference between saturated liquid water and saturated water vapor saturated water vapor saturated ice difference between saturated ice and saturated water vapor saturated moist air total water in any phase REFERENCES ASHRAE members can access ASHRAE Journal articles and ASHRAE research project final reports at technologyportal.ashrae .org. 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Thermodynamic properties of dry air, moist air and water and SI psychrometric charts. ASHRAE Research Projects RP-216 and RP-257, Final Report. Related Commercial Resources This file is licensed to John Murray (). Publication Date: 6/1/2017 Related Commercial Resources CHAPTER 2 Licensed for single user. © 2017 ASHRAE, Inc. THERMODYNAMICS AND REFRIGERATION CYCLES THERMODYNAMICS.. 2.1 Stored Energy .... 2.1 Energy in Transition .... 2.1 First Law of Thermodynamics .... 2.2 Second Law of Thermodynamics .... 2.2 Thermodynamic Analysis of Refrigeration Cycles..................... 2.3 Equations of State .... 2.4 Calculating Thermodynamic Properties................................... 2.4 COMPRESSION REFRIGERATION CYCLES......................... 2.6 Carnot Cycle .... 2.6 Theoretical Single-Stage Cycle Using a Pure Refrigerant or Azeotropic Mixture... 2.7 Lorenz Refrigeration Cycle.. 2.9 Theoretical Single-Stage Cycle Using Zeotropic Refrigerant Mixture... 2.10 Multistage Vapor Compression Refrigeration Cycles .... Actual Refrigeration Systems .... ABSORPTION REFRIGERATION CYCLES .... Ideal Thermal Cycle .... Working-Fluid Phase Change Constraints .... Working Fluids.. Effect of Fluid Properties on Cycle Performance .... Absorption Cycle Representations .... Conceptualizing the Cycle.. Absorption Cycle Modeling.. Ammonia/Water Absorption Cycles .... ADSORPTION REFRIGERATION SYSTEMS......................... Symbols .... HERMODYNAMICS is the study of energy, its transformations, and its relation to states of matter. This chapter covers the application of thermodynamics to refrigeration cycles. The first part reviews the first and second laws of thermodynamics and presents methods for calculating thermodynamic properties. The second and third parts address compression and absorption refrigeration cycles, two common methods of thermal energy transfer. Chemical energy is caused by the arrangement of atoms composing the molecules. Nuclear (atomic) energy derives from the cohesive forces holding protons and neutrons together as the atom's nucleus. T 1. THERMODYNAMICS A thermodynamic system is a region in space or a quantity of matter bounded by a closed surface. The surroundings include everything external to the system, and the system is separated from the surroundings by the system boundaries. These boundaries can be movable or fixed, real or imaginary. Entropy and energy are important in any thermodynamic system. Entropy measures the molecular disorder of a system. The more mixed a system, the greater its entropy; an orderly or unmixed configuration is one of low entropy. Energy has the capacity for producing an effect and can be categorized into either stored or transient forms. 1.1 STORED ENERGY Thermal (internal) energy is caused by the motion of molecules and/or intermolecular forces. Potential energy (PE) is caused by attractive forces existing between molecules, or the elevation of the system. PE = mgz (1) 1.2 2.10 2.11 2.13 2.13 2.14 2.15 2.16 2.16 2.16 2.17 2.19 2.20 2.21 ENERGY IN TRANSITION Heat Q is the mechanism that transfers energy across the boundaries of systems with differing temperatures, always toward the lower temperature. Heat is positive when energy is added to the system (see Figure 1). Work is the mechanism that transfers energy across the boundaries of systems with differing pressures (or force of any kind), always toward the lower pressure. If the total effect produced in the system can be reduced to the raising of a weight, then nothing but work has crossed the boundary. Work is positive when energy is removed from the system (see Figure 1). Mechanical or shaft work W is the energy delivered or absorbed by a mechanism, such as a turbine, air compressor, or internal combustion engine. Flow work is energy carried into or transmitted across the system boundary because a pumping process occurs somewhere outside the system, causing fluid to enter the system. It can be more easily understood as the work done by the fluid just outside the system on the adjacent fluid entering the system to force or push it into the system. Flow work also occurs as fluid leaves the system. Flow work (per unit mass) = pv where m = mass g = local acceleration of gravity z = elevation above horizontal reference plane Kinetic energy (KE) is the energy caused by the velocity of molecules and is expressed as KE = mV 2/2 (2) where V is the velocity of a fluid stream crossing the system boundary. The preparation of the first and second parts of this chapter is assigned to TC 1.1, Thermodynamics and Psychrometrics. The third and fourth parts are assigned to TC 8.3, Absorption and Heat Operated Machines. 2.1 Copyright © 2017, ASHRAE Fig. 1 Energy Flows in General Thermodynamic System (3) This file is licensed to John Murray (). Publication Date: 6/1/2017 2.2 2017 ASHRAE Handbook—Fundamentals (SI) where p is pressure and v is specific volume, or the volume displaced per unit mass evaluated at the inlet or exit. A property of a system is any observable characteristic of the system. The state of a system is defined by specifying the minimum set of independent properties. The most common thermodynamic properties are temperature T, pressure p, and specific volume v or density . Additional thermodynamic properties include entropy, stored forms of energy, and enthalpy. Frequently, thermodynamic properties combine to form other properties. Enthalpy h is an important property that includes internal energy and flow work and is defined as Licensed for single user. © 2017 ASHRAE, Inc. h u + pv min u + pv + -----2- + gz in V2 V – m out u + pv + ------ + gz +Q–W out 2 2 V2 V = mf u + ------ + gz – m i u + ------ + gz i 2 2 f Net amount of energy = Net increase of stored added to system energy in system or [Energy in] – [Energy out] = [Increase of stored energy in system] system where subscripts i and f refer to the initial and final states, respectively. Nearly all important engineering processes are commonly modeled as steady-flow processes. Steady flow signifies that all quantities associated with the system do not vary with time. Consequently, all streams entering – V2 m· h + ------ + gz 2 all streams leaving V2 m· h + ------ + gz + Q· – W· = 0 2 (6) where h u + pv as described in Equation (4). A second common application is the closed stationary system for which the first law equation reduces to Q – W = [m(uf – ui)]system 1.4 (7) SECOND LAW OF THERMODYNAMICS The second law of thermodynamics differentiates and quantifies processes that only proceed in a certain direction (irreversible) from those that are reversible. The second law may be described in several ways. One method uses the concept of entropy flow in an open system and the irreversibility associated with the process. The concept of irreversibility provides added insight into the operation of cycles. For example, the larger the irreversibility in a refrigeration cycle operating with a given refrigeration load between two fixed temperature levels, the larger the amount of work required to operate the cycle. Irreversibilities include pressure drops in lines and heat exchangers, heat transfer between fluids of different temperature, and mechanical friction. Reducing total irreversibility in a cycle improves cycle performance. In the limit of no irreversibilities, a cycle attains its maximum ideal efficiency. In an open system, the second law of thermodynamics can be described in terms of entropy as Q dSsystem = ------- + mi si – me se + dI T FIRST LAW OF THERMODYNAMICS The first law of thermodynamics is often called the law of conservation of energy. The following form of the first-law equation is valid only in the absence of a nuclear or chemical reaction. Based on the first law or the law of conservation of energy, for any system, open or closed, there is an energy balance as (5) 2 (4) where u is the internal energy per unit mass. Each property in a given state has only one definite value, and any property always has the same value for a given state, regardless of how the substance arrived at that state. A process is a change in state that can be defined as any change in the properties of a system. A process is described by specifying the initial and final equilibrium states, the path (if identifiable), and the interactions that take place across system boundaries during the process. A cycle is a process or a series of processes wherein the initial and final states of the system are identical. Therefore, at the conclusion of a cycle, all the properties have the same value they had at the beginning. Refrigerant circulating in a closed system undergoes a cycle. A pure substance has a homogeneous and invariable chemical composition. It can exist in more than one phase, but the chemical composition is the same in all phases. If a substance is liquid at the saturation temperature and pressure, it is called a saturated liquid. If the temperature of the liquid is lower than the saturation temperature for the existing pressure, it is called either a subcooled liquid (the temperature is lower than the saturation temperature for the given pressure) or a compressed liquid (the pressure is greater than the saturation pressure for the given temperature). When a substance exists as part liquid and part vapor at the saturation temperature, its quality is defined as the ratio of the mass of vapor to the total mass. Quality has meaning only when the substance is saturated (i.e., at saturation pressure and temperature). Pressure and temperature of saturated substances are not independent properties. If a substance exists as a vapor at saturation temperature and pressure, it is called a saturated vapor. (Sometimes the term dry saturated vapor is used to emphasize that the quality is 100%.) When the vapor is at a temperature greater than the saturation temperature, it is a superheated vapor. Pressure and temperature of a superheated vapor are independent properties, because the temperature can increase while pressure remains constant. Gases such as air at room temperature and pressure are highly superheated vapors. 1.3 Figure 1 illustrates energy flows into and out of a thermodynamic system. For the general case of multiple mass flows with uniform properties in and out of the system, the energy balance can be written (8) where total change within system in time dt during process entropy increase caused by mass entering (incoming) entropy decrease caused by mass leaving (exiting) entropy change caused by reversible heat transfer between system and surroundings at temperature T dI = entropy caused by irreversibilities (always positive) dSsystem mi si me se Q/T = = = = Equation (8) accounts for all entropy changes in the system. Rearranged, this equation becomes Q = T [(me se – mi si) + dSsys – dI] (9) This file is licensed to John Murray (). Publication Date: 6/1/2017 Thermodynamics and Refrigeration Cycles 2.3 In integrated form, if inlet and outlet properties, mass flow, and interactions with the surroundings do not vary with time, the general equation for the second law is (Sf – Si)system = rev ------T- + ms in – ms out + I Q (10) In many applications, the process can be considered to operate steadily with no change in time. The change in entropy of the system is therefore zero. The irreversibility rate, which is the rate of entropy production caused by irreversibilities in the process, can be determined by rearranging Equation (10): · I = Q m· s out – m· s in – ----------T · · X in – X· out Rate of net exergy transfer by heat, work, and mass Licensed for single user. © 2017 ASHRAE, Inc. (12) In a cycle, the reduction of work produced by a power cycle (or the increase in work required by a refrigeration cycle) equals the absolute ambient temperature multiplied by the sum of irreversibilities in all processes in the cycle. Thus, the difference in reversible and actual work for any refrigeration cycle, theoretical or real, operating under the same conditions, becomes · · · Wactual = Wreversible + T 0 I (13) Another second-law method to describe performance of engineering devices is the concept of exergy (also called the availability, potential energy, or work potential), which is the maximum useful work that could be obtained from the system at a given state in a specified environment. There is always a difference between exergy and the actual work delivered by a device; this difference represents the room for improvement. Note that exergy is a property of the system/environment combination and not of the system alone. The exergy of a system in equilibrium with its environment is zero. The state of the environment is referred to as the dead state, because the system cannot do any work. Exergy transfer is in three forms (heat, work, and mass flow), and is given by T 0 Xheat = 1 – ----- Q T W – W surr Xwork = W (for boundary work) (for other forms of work) Xmass = m where = (h – h0) – T0(s – s0) + (V 2/2) + gz is flow exergy. Exergy balance for any system undergoing any process can be expressed as Xin – Xout Net exergy transfer by heat, work, and mass – Xdestroyed Exergy destruction = Xsystem Change in exergy (general) = Rate of exergy destruction dXsystem /dt Rate of change in exergy (general, in rate form) T0 Q – W – P0 V2 – V1 1 – ----Tk k + m – m – X destroyed = X 2 – X 1 in surr h out – h in · I = m· s out – s in – ---------------------T surr · X destroyed Taking the positive direction of heat transfer as to the system and the positive direction of work transfer as from the system, the general exergy balance relations can be expressed explicitly as (11) Equation (6) can be used to replace the heat transfer quantity. Note that the absolute temperature of the surroundings with which the system is exchanging heat is used in the last term. If the temperature of the surroundings is equal to the system temperature, heat is transferred reversibly and the last term in Equation (11) equals zero. Equation (11) is commonly applied to a system with one mass flow in, the same mass flow out, no work, and negligible kinetic or potential energy flows. Combining Equations (6) and (11) yields – 1.5 out THERMODYNAMIC ANALYSIS OF REFRIGERATION CYCLES Refrigeration cycles transfer thermal energy from a region of low temperature TR to one of higher temperature. Usually the highertemperature heat sink is the ambient air or cooling water, at temperature T0 , the temperature of the surroundings. The first and second laws of thermodynamics can be applied to individual components to determine mass and energy balances and the irreversibility of the components. This procedure is shown in later sections in this chapter. Performance of a refrigeration cycle is usually described by a coefficient of performance (COP), defined as the benefit of the cycle (amount of heat removed) divided by the required energy input to operate the cycle: Useful refrigerating effect COP --Net energy supplied from external sources (14) For a mechanical vapor compression system, the net energy supplied is usually in the form of work, mechanical or electrical, and may include work to the compressor and fans or pumps. Thus, Q evap COP = -------------W net (15) In an absorption refrigeration cycle, the net energy supplied is usually in the form of heat into the generator and work into the pumps and fans, or Q evap COP = -----------------------------Q gen + W net (16) In many cases, work supplied to an absorption system is very small compared to the amount of heat supplied to the generator, so the work term is often neglected. Applying the second law to an entire refrigeration cycle shows that a completely reversible cycle operating under the same conditions has the maximum possible COP. Departure of the actual cycle from an ideal reversible cycle is given by the refrigerating efficiency: COP R = --------------------- COP rev (17) The Carnot cycle usually serves as the ideal reversible refrigeration cycle. For multistage cycles, each stage is described by a reversible cycle. This file is licensed to John Murray (). Publication Date: 6/1/2017 2.4 2017 ASHRAE Handbook—Fundamentals (SI) 1.6 EQUATIONS OF STATE The equation of state of a pure substance is a mathematical relation between pressure, specific volume, and temperature. When the system is in thermodynamic equilibrium, f (p,v,T) = 0 (18) Licensed for single user. © 2017 ASHRAE, Inc. The principles of statistical mechanics are used to (1) explore the fundamental properties of matter, (2) predict an equation of state based on the statistical nature of a particular system, or (3) propose a functional form for an equation of state with unknown parameters that are determined by measuring thermodynamic properties of a substance. A fundamental equation with this basis is the virial equation, which is expressed as an expansion in pressure p or in reciprocal values of volume per unit mass v as pv ------- = 1 + B'p + C'p2 + D'p3 + ... RT (19) pv ------- = 1 + (B/v) + (C/v2) + (D/v3) + ... RT (20) where coefficients B', C', D', etc., and B, C, D, etc., are the virial coefficients. B' and B are the second virial coefficients; C' and C are the third virial coefficients, etc. The virial coefficients are functions of temperature only, and values of the respective coefficients in Equations (19) and (20) are related. For example, B' = B/RT and C' = (C – B2)/(RT) 2. The universal gas constant R is defined as pv R = lim ------------Tp 0 T (21) P = (RT/v) + (BoRT – Ao – Co/T 2)/v2 + (bRT – a)/v3 2 + (a)/v6 + [c(1 + /v 2)e(–/v)]/v3T 2 – kT T c – kT T c A4 + B4 T A5 + B5 T + C5 e av + -------------------- + A 6 + B 6 T e - + ---(24) 5 4 v – b v – b where the constant coefficients are Ai , Bi , Ci , k, b, and a. Strobridge (1962) suggested an equation of state that was developed for nitrogen properties and used for most cryogenic fluids. This equation combines the B-W-R equation of state with an equation for high-density nitrogen suggested by Benedict (1937). These equations have been used successfully for liquid and vapor phases, extending in the liquid phase to the triple-point temperature and the freezing line, and in the vapor phase from 10 to 1000 K, with pressures to 1 GPa. The Strobridge equation is accurate within the uncertainty of the measured p-v-T data: n3 n4 n5 2 p = RT + Rn 1 T + n 2 + ----- + -----2 + -----4 T T T 3 (22) An advantage of the virial form is that statistical mechanics can be used to predict the lower-order coefficients and provide physical significance to the virial coefficients. For example, in Equation (22), the term B/v is a function of interactions between two molecules, C/v 2 between three molecules, etc. Because lower-order interactions are common, contributions of the higher-order terms are successively less. Thermodynamicists use the partition or distribution function to determine virial coefficients; however, experimental values of the second and third coefficients are preferred. For dense fluids, many higher-order terms are necessary that can neither be satisfactorily predicted from theory nor determined from experimental measurements. In general, a truncated virial expansion of four terms is valid for densities of less than one-half the value at the critical point. For higher densities, additional terms can be used and determined empirically. Computers allow the use of very complex equations of state in calculating p-v-T values, even to high densities. The BenedictWebb-Rubin (B-W-R) equation of state (Benedict et al. 1940) and Martin-Hou equation (1955) have had considerable use, but should generally be limited to densities less than the critical value. Strobridge (1962) suggested a modified Benedict-Webb-Rubin relation that gives excellent results at higher densities and can be used for a p-v-T surface that extends into the liquid phase. The B-W-R equation has been used extensively for hydrocarbons (Cooper and Goldfrank 1967): – kT T c A3 + B3 T + C3 e RT A 2 + B 2 T + C 2 e p = ---------- + --- + ---3 2 v–b v – b v – b + Rn 6 T + n 7 + n 8 T where pv T is the product of the pressure and the molar specific volume along an isotherm with absolute temperature T. The current best value of R is 8314.41 J/(kg mol·K). The gas constant R is equal to the universal gas constant R divided by the molecular mass M of the gas or gas mixture. The quantity pv/RT is also called the compressibility factor Z, or Z = 1 + (B/v) + (C/v2) + (D/v3) + ... (23) where the constant coefficients are Ao, Bo, Co, a, b, c, , and . The Martin-Hou equation, developed for fluorinated hydrocarbon properties, has been used to calculate the thermodynamic property tables in Chapter 30 and in ASHRAE Thermodynamic Properties of Refrigerants (Stewart et al. 1986). The Martin-Hou equation is 4 3 n 9 n 10 n 11 2 + -----2 + ------+ ------- exp – n 16 T T3 T 4 5 n 12 n 13 n 14 2 6 + -------2 + -------3 + -------4 exp – n 16 + n 15 T T T (25) The 15 coefficients of this equation's linear terms are determined by a least-square fit to experimental data. Hust and McCarty (1967) and Hust and Stewart (1966) give further information on methods and techniques for determining equations of state. In the absence of experimental data, van der Waals' principle of corresponding states can predict fluid properties. This principle relates properties of similar substances by suitable reducing factors (i.e., the p-v-T surfaces of similar fluids in a given region are assumed to be of similar shape). The critical point can be used to define reducing parameters to scale the surface of one fluid to the dimensions of another. Modifications of this principle, as suggested by Kamerlingh Onnes, a Dutch cryogenic researcher, have been used to improve correspondence at low pressures. The principle of corresponding states provides useful approximations, and numerous modifications have been reported. More complex treatments for predicting properties, which recognize similarity of fluid properties, are by generalized equations of state. These equations ordinarily allow adjustment of the p-v-T surface by introducing parameters. One example (Hirschfelder et al. 1958) allows for departures from the principle of corresponding states by adding two correlating parameters. 1.7 CALCULATING THERMODYNAMIC PROPERTIES Although equations of state provide p-v-T relations, thermodynamic analysis usually requires values for internal energy, This file is licensed to John Murray (). Publication Date: 6/1/2017 Licensed for single user. © 2017 ASHRAE, Inc. Thermodynamics and Refrigeration Cycles 2.5 enthalpy, and entropy. These properties have been tabulated for many substances, including refrigerants (see Chapters 1, 30, and 33), and can be extracted from such tables by interpolating manually or with a suitable computer program. This approach is appropriate for hand calculations and for relatively simple computer models; however, for many computer simulations, the overhead in memory or input and output required to use tabulated data can make this approach unacceptable. For large thermal system simulations or complex analyses, it may be more efficient to determine internal energy, enthalpy, and entropy using fundamental thermodynamic relations or curves fit to experimental data. Some of these relations are discussed in the following sections. Also, the thermodynamic relations discussed in those sections are the basis for constructing tables of thermodynamic property data. Further information on the topic may be found in references covering system modeling and thermodynamics (Howell and Buckius 1992; Stoecker 1989). At least two intensive properties (properties independent of the quantity of substance, such as temperature, pressure, specific volume, and specific enthalpy) must be known to determine the remaining properties. If two known properties are either p, v, or T (these are relatively easy to measure and are commonly used in simulations), the third can be determined throughout the range of interest using an equation of state. Furthermore, if the specific heats at zero pressure are known, specific heat can be accurately determined from spectroscopic measurements using statistical mechanics (NASA 1971). Entropy may be considered a function of T and p, and from calculus an infinitesimal change in entropy can be written as s s ds = ------ dT + ------ dp T pT p (26) Likewise, a change in enthalpy can be written as h h dh = ------ dT + ------ dp pT Tp (27) Using the Gibbs relation Tds dh – vdp and the definition of specific heat at constant pressure, cp (h/T)p , Equation (27) can be rearranged to yield cp dp h ds = ----- dT + – v -----T T p T (29) This is an expression for an exact derivative, so it follows that 2 c p v (30) T = – 2 p T T p Integrating this expression at a fixed temperature yields 2 v cp = cp0 – T 2 dpT 0 T v dh = cpdT + v – T dp T p p (31) where cp0 is the known zero-pressure specific heat, and dpT is used to indicate that integration is performed at a fixed temperature. The second partial derivative of specific volume with respect to temperature can be determined from the equation of state. Thus, Equation (31) can be used to determine the specific heat at any pressure. (32) Equations (28) and (32) may be integrated at constant pressure to obtain ----T- dTp T1 cp s(T1, p0) = s(T0, p0) + (33) T0 cp dT T1 h(T1, p0) = h(T0, p0) + and (34) T0 Integrating the Maxwell relation (s/p)T = –(v/T)p gives an equation for entropy changes at a constant temperature as Tp dpT p p1 s(T0, p1) = s(T0, p0) – v (35) 0 Likewise, integrating Equation (32) along an isotherm yields the following equation for enthalpy changes at a constant temperature: v – T Tp dp p p1 h(T0, p1) = h(T0, p0) + v (36) 0 Internal energy can be calculated from u = h – pv. When entropy or enthalpy are known at a reference temperature T0 and pressure p0, values at any temperature and pressure may be obtained by combining Equations (33) and (35) or Equations (34) and (36). Combinations (or variations) of Equations (33) to (36) can be incorporated directly into computer subroutines to calculate properties with improved accuracy and efficiency. However, these equations are restricted to situations where the equation of state is valid and the properties vary continuously. These restrictions are violated by a change of phase such as evaporation and condensation, which are essential processes in air-conditioning and refrigerating devices. Therefore, the Clapeyron equation is of particular value; for evaporation or condensation, it gives h fg s fg dp = ------ = -------------- v fg Tv fg dT sat (28) Equations (26) and (28) combine to yield (s/T)p = cp /T. Then, using the Maxwell relation (s/p)T = –(v/T)p , Equation (26) may be rewritten as cp v ds = ----- dT – ------ dp T Tp Using Tds dh – vdp, Equation (29) can be written as (37) where sfg = entropy of vaporization hfg = enthalpy of vaporization vfg = specific volume difference between vapor and liquid phases If vapor pressure and liquid and vapor density data (all relatively easy measurements to obtain) are known at saturation, then changes in enthalpy and entropy can be calculated using Equation (37). Phase Equilibria for Multicomponent Systems To understand phase equilibria, consider a container full of a liquid made of two components; the more volatile component is designated i and the less volatile component j (Figure 2A). This mixture is all liquid because the temperature is low (but not so low that a solid appears). Heat added at a constant pressure raises the mixture's temperature, and a sufficient increase causes vapor to form, as shown in Figure 2B. If heat at constant pressure continues to be added, eventually the temperature becomes so high that only vapor remains in the container (Figure 2C). A temperature-concentration (T- x) diagram is useful for exploring details of this situation. Figure 3 is a typical T- x diagram valid at a fixed pressure. The case shown in Figure 2A, a container full of liquid mixture with This file is licensed to John Murray (). Publication Date: 6/1/2017 2.6 2017 ASHRAE Handbook—Fundamentals (SI) Fig. 2 Mixture of i and j Components in Constant-Pressure Container Licensed for single user. © 2017 ASHRAE, Inc. Fig. 4 Azeotropic Behavior Shown on T-x Diagram Fig. 3 Temperature-Concentration (T-x) Diagram for Zeotropic Mixture mole fraction xi,0 at temperature T0 , is point 0 on the T- x diagram. When heat is added, the mixture's temperature increases. The point at which vapor begins to form is the bubble point. Starting at point 0, the first bubble forms at temperature T1 (point 1 on the diagram). The locus of bubble points is the bubble-point curve, which provides bubble points for various liquid mole fractions xi. When the first bubble begins to form, vapor in the bubble may not have the same mole fraction as the liquid mixture. Rather, the mole fraction of the more volatile species is higher in the vapor than in the liquid. Boiling prefers more volatile species, and the T- x diagram shows this behavior. At Tl, the vapor-forming bubbles have an i mole fraction of yi,l. If heat continues to be added, this preferential boiling depletes the liquid of species i and the temperature required to continue the process increases. Again, the T- x diagram reflects this fact; at point 2 the i mole fraction in the liquid is reduced to xi,2 and the vapor has a mole fraction of yi,2. The temperature required to boil the mixture is increased to T2. Position 2 on the T-x diagram could correspond to the physical situation shown in Figure 2B. If constant-pressure heating continues, all the liquid eventually becomes vapor at temperature T3. The vapor at this point is shown as position 3 in Figure 3. At this point the i mole fraction in the vapor yi,3 equals the starting mole fraction in the all-liquid mixture xi,1. This equality is required for mass and species conservation. Further addition of heat simply raises the vapor temperature. The final position 4 corresponds to the physical situation shown in Figure 2C. Starting at position 4 in Figure 3, heat removal leads to initial liquid formation when position 3 (the dew point) is reached. The locus of dew points is called the dew-point curve. Heat removal causes the liquid phase of the mixture to reverse through points 3, 2, 1, and to starting point 0. Because the composition shifts, the temperature required to boil (or condense) this mixture changes as the process proceeds. This is known as temperature glide. This mixture is therefore called zeotropic. Most mixtures have T- x diagrams that behave in this fashion, but some have a markedly different feature. If the dew-point and bubble-point curves intersect at any point other than at their ends, the mixture exhibits azeotropic behavior at that composition. This case is shown as position a in the T- x diagram of Figure 4. If a container of liquid with a mole fraction xa were boiled, vapor would be formed with an identical mole fraction ya . The addition of heat at constant pressure would continue with no shift in composition and no temperature glide. Perfect azeotropic behavior is uncommon, although nearazeotropic behavior is fairly common. The azeotropic composition is pressure dependent, so operating pressures should be considered for their effect on mixture behavior. Azeotropic and near-azeotropic refrigerant mixtures are widely used. The properties of an azeotropic mixture are such that they may be conveniently treated as pure substance properties. Phase equilibria for zeotropic mixtures, however, require special treatment, using an equation-of-state approach with appropriate mixing rules or using the fugacities with the standard state method (Tassios 1993). Refrigerant and lubricant blends are a zeotropic mixture and can be treated by these methods (Martz et al. 1996a, 1996b; Thome 1995). 2. COMPRESSION REFRIGERATION CYCLES 2.1 CARNOT CYCLE The Carnot cycle, which is completely reversible, is a perfect model for a refrigeration cycle operating between two fixed temperatures, or between two fluids at different temperatures and each with infinite heat capacity. Reversible cycles have two important properties: (1) no refrigerating cycle may have a coefficient of performance higher than that for a reversible cycle operated between the same temperature limits, and (2) all reversible cycles, when This file is licensed to John Murray (). Publication Date: 6/1/2017 Thermodynamics and Refrigeration Cycles 2.7 operated between the same temperature limits, have the same coefficient of performance. Proof of both statements may be found in almost any textbook on elementary engineering thermodynamics. Figure 5 shows the Carnot cycle on temperature-entropy coordinates. Heat is withdrawn at constant temperature TR from the region to be refrigerated. Heat is rejected at constant ambient temperature T0. The cycle is completed by an isentropic expansion and an isentropic compression. The energy transfers are given by Q0 = T0(S2 – S3) Qi = TR (S1 – S4) = TR (S2 – S3) Wnet = Qo – Qi Thus, by Equation (15), TR COP = -----------------T0 – TR (38) Example 1. Determine entropy change, work, and COP for the cycle shown in Figure 6. Temperature of the refrigerated space TR is 250 K, and that of the atmosphere T0 is 300 K. Refrigeration load is 125 kJ. Solution: Licensed for single user. © 2017 ASHRAE, Inc. S = S1 – S4 = Qi/TR = 125/250 = 0.5 kJ/K W = S(T0 – TR) = 0.5(300 – 250) = 25 kJ COP = Qi /(Qo – Qi) = Qi /W = 125/25 = 5 Flow of energy and its area representation in Figure 6 are Energy kJ Area Qi Qo W 125 150 25 b a+b a The net change of entropy of any refrigerant in any cycle is always zero. In Example 1, the change in entropy of the refrigerated space is SR = –125/250 = –0.5 kJ/K and that of the atmosphere is So = 125/ 250 = 0.5 kJ/K. The net change in entropy of the isolated system is Stotal = SR + So = 0. The Carnot cycle in Figure 7 shows a process in which heat is added and rejected at constant pressure in the two-phase region of a refrigerant. Saturated liquid at state 3 expands isentropically to the low temperature and pressure of the cycle at state d. Heat is added isothermally and isobarically by evaporating the liquid-phase refrigerant from state d to state 1. The cold saturated vapor at state 1 is compressed isentropically to the high temperature in the cycle at state b. However, the pressure at state b is below the saturation pressure corresponding to the high temperature in the cycle. The compression process is completed by an isothermal compression process from state b to state c. The cycle is completed by an isothermal and isobaric heat rejection or condensing process from state c to state 3. Applying the energy equation for a mass of refrigerant m yields (all work and heat transfer are positive) 3W d = m(h3 – hd) 1Wb = m(hb – h1) bW c = T0(Sb – Sc) – m(hb – hc) d Q1 = m(h1 – hd) = Area def1d The net work for the cycle is Wnet = 1Wb + bWc – 3Wd = Area d1bc3d TR d Q1 COP = ----------- = -----------------W net T0 – TR and Fig. 5 Carnot Refrigeration Cycle 2.2 THEORETICAL SINGLE-STAGE CYCLE USING A PURE REFRIGERANT OR AZEOTROPIC MIXTURE A system designed to approach the ideal model shown in Figure 7 is desirable. A pure refrigerant or azeotropic mixture can be used to maintain constant temperature during phase changes by maintaining constant pressure. Because of concerns such as high initial cost and increased maintenance requirements, a practical machine Fig. 6 Temperature-Entropy Diagram for Carnot Refrigeration Cycle of Example 1 Fig. 7 Carnot Vapor Compression Cycle This file is licensed to John Murray (). Publication Date: 6/1/2017 2.8 2017 ASHRAE Handbook—Fundamentals (SI) has one compressor instead of two and the expander (engine or turbine) is replaced by a simple expansion valve, which throttles refrigerant from high to low pressure. Figure 8 shows the theoretical single-stage cycle used as a model for actual systems. Applying the energy equation for a mass m of refrigerant yields = m(h1 – h4) (39a) 1W2 = m(h2 – h1) (39b) = m(h2 – h3) (39c) h3 = h4 (39d) 4Q1 2Q3 Constant-enthalpy throttling assumes no heat transfer or change in potential or kinetic energy through the expansion valve. The coefficient of performance is h1 – h4 COP = --------- = ----------------h2 – h1 1W 2 4Q1 h4 – hf – 173.64 x4 = --------------- = 241.72 --- = 0.3198 hg – hf 386.55 – 173.64 v4 = vf + x4(vg – vf) = 0.0007362 + 0.3198(0.14739 – 0.0007362) = 0.04764 m3/kg s4 = sf + x4(sg – sf) = 0.9002 + 0.3198(1.7413 – 0.9002) = 1.16918 kJ/(kg·K) The property data are tabulated in Table 1. (b) By Equation (40), 386.55 – 241.71 COP = --- = 3.97 423.07 – 386.55 (40) The theoretical compressor displacement CD (at 100% volumetric efficiency) is CD = m· v (41) 1 Licensed for single user. © 2017 ASHRAE, Inc. Chapter 30. Specific volume and specific entropy values for state 4 are obtained by determining the quality of the liquid-vapor mixture from the enthalpy. which is a measure of the physical size or speed of the compressor required to handle the prescribed refrigeration load. Example 2. A theoretical single-stage cycle using R-134a as the refrigerant operates with a condensing temperature of 30°C and an evaporating temperature of –20°C. The system produces 50 kW of refrigeration. Determine the (a) thermodynamic property values at the four main state points of the cycle, (b) COP, (c) cycle refrigerating efficiency, and (d) rate of refrigerant flow. Solution: (a) Figure 9 shows a schematic p-h diagram for the problem with numerical property data. Saturated vapor and saturated liquid properties for states 1 and 3 are obtained from the saturation table for R-134a in Chapter 30. Properties for superheated vapor at state 2 are obtained by linear interpolation of the superheat tables for R-134a in (c) By Equations (17) and (38), COP T 3 – T 1 3.97 50 R = ---------------------------------- = ------------------------- = 0.78 or 78% T1 253.15 (d) The mass flow of refrigerant is obtained from an energy balance on the evaporator. Thus, · m· h 1 – h 4 = Q i = 50 kW and · Qi 50 m· = --------------------- = --- = 0.345 kg/s h1 – h4 386.55 – 241.72 The saturation temperatures of the single-stage cycle strongly influence the magnitude of the coefficient of performance. This influence may be readily appreciated by an area analysis on a temperature-entropy (T- s) diagram. The area under a reversible process line on a T- s diagram is directly proportional to the thermal energy added or removed from the working fluid. This observation follows directly from the definition of entropy [see Equation (8)]. Table 1 Thermodynamic Property Data for Example 2 State 1 2 3 4 Fig. 8 Theoretical Single-Stage Vapor Compression Refrigeration Cycle t, °C p, kPa v, m3/kg h, kJ/kg s, kJ/(kg·K) –20.0 37.8 30.0 –20.0 132.73 770.20 770.20 132.73 0.14739 0.02798 0.000842 0.047636 386.55 423.07 241.72 241.72 1.7413 1.7413 1.1435 1.16918 Fig. 9 Schematic p-h Diagram for Example 2 This file is licensed to John Murray (). Publication Date: 6/1/2017 Thermodynamics and Refrigeration Cycles 2.9 Fig. 11 Processes of Lorenz Refrigeration Cycle Licensed for single user. © 2017 ASHRAE, Inc. Fig. 10 Areas on T- s Diagram Representing Refrigerating Effect and Work Supplied for Theoretical Single-Stage Cycle In Figure 10, the area representing Qo is the total area under the constant-pressure curve between states 2 and 3. The area representing the refrigerating capacity Qi is the area under the constantpressure line connecting states 4 and 1. The net work required Wnet equals the difference (Qo – Qi), which is represented by the entire shaded area shown on Figure 10. Because COP = Qi /Wnet , the effect on the COP of changes in evaporating temperature and condensing temperature may be observed. For example, a decrease in evaporating temperature TE significantly increases Wnet and slightly decreases Qi. An increase in condensing temperature TC produces the same results but with less effect on Wnet . Therefore, for maximum coefficient of performance, the cycle should operate at the lowest possible condensing temperature and maximum possible evaporating temperature. 2.3 Carnot refrigeration cycle operating between T0 and TR would lie between states 1, a, 3, and b; the Lorenz cycle has a smaller refrigerating effect and requires more work, but this cycle is a more practical reference when a refrigeration system operates between two single-phase fluids such as air or water. The energy transfers in a Lorenz refrigeration cycle are as follows, where T is the temperature change of the refrigerant during each of the two heat exchange processes. Qo = (T0 + T/2)(S2 – S3) Qi = (TR – T/2)(S1 – S4) = (TR – T/2)(S2 – S3) Wnet = Qo – QR Thus by Equation (15), T R – T 2 COP = -------------------------------T 0 – T R + T LORENZ REFRIGERATION CYCLE The Carnot refrigeration cycle includes two assumptions that make it impractical. The heat transfer capacities of the two external fluids are assumed to be infinitely large so the external fluid temperatures remain fixed at T0 and TR (they become infinitely large thermal reservoirs). The Carnot cycle also has no thermal resistance between the working refrigerant and external fluids in the two heat exchange processes. As a result, the refrigerant must remain fixed at T0 in the condenser and at TR in the evaporator. The Lorenz cycle eliminates the first restriction in the Carnot cycle by allowing the temperature of the two external fluids to vary during heat exchange. The second assumption of negligible thermal resistance between the working refrigerant and two external fluids remains. Therefore, the refrigerant temperature must change during the two heat exchange processes to equal the changing temperature of the external fluids. This cycle is completely reversible when operating between two fluids that each have a finite but constant heat capacity. Figure 11 is a schematic of a Lorenz cycle. Note that this cycle does not operate between two fixed temperature limits. Heat is added to the refrigerant from state 4 to state 1. This process is assumed to be linear on T-s coordinates, which represents a fluid with constant heat capacity. The refrigerant temperature is increased in isentropic compression from state 1 to state 2. Process 2-3 is a heat rejection process in which the refrigerant temperature decreases linearly with heat transfer. The cycle ends with isentropic expansion between states 3 and 4. The heat addition and heat rejection processes are parallel so the entire cycle is drawn as a parallelogram on T- s coordinates. A (42) Example 3. Determine the entropy change, work required, and COP for the Lorenz cycle shown in Figure 11 when the temperature of the refrigerated space is TR = 250 K, ambient temperature is T0 = 300 K, T of the refrigerant is 5 K, and refrigeration load is 125 kJ. Solution: - = 4 -------T 1 S = Q i Qi 125 ------------------------------ = ------------- = 0.5051 kJ/K T R – T 2 247.5 Qo = [T0 + (T/2)]S = (300 + 2.5)0.5051 = 152.79 kJ Wnet = Qo – QR = 152.79 – 125 = 27.79 kJ T R – T 2 250 – 5 2 247.5 COP = ------------------------------- = --------------------------------- = ------------- = 4.50 T 0 – T R + T 300 – 250 + 5 55 Note that the entropy change for the Lorenz cycle is larger than for the Carnot cycle when both operate between the same two temperature reservoirs and have the same capacity (see Example 1). That is, both the heat rejection and work requirement are larger for the Lorenz cycle. This difference is caused by the finite temperature difference between the working fluid in the cycle compared to the bounding temperature reservoirs. However, as discussed previously, the assumption of constant-temperature heat reservoirs is not necessarily a good representation of an actual refrigeration system because of the temperature changes that occur in the heat exchangers. This file is licensed to John Murray (). Publication Date: 6/1/2017 2.10 2017 ASHRAE Handbook—Fundamentals (SI) such as in a supermarket, or when evaporator temperature becomes very low. Low evaporator temperature indicates low evaporator pressure and low refrigerant density into the compressor. Two small compressors in series have a smaller displacement and usually operate more efficiently than one large compressor that covers the entire pressure range from the evaporator to the condenser. This is especially true in ammonia refrigeration systems because of the large amount of superheating that occurs during the compression process. Thermodynamic analysis of multistage cycles is similar to analysis of single-stage cycles, except that mass flow differs through various components of the system. A careful mass balance and energy balance on individual components or groups of components ensures correct application of the first law of thermodynamics. Care must also be used when performing second-law calculations. Often, the refrigerating load is comprised of more than one evaporator, so the total system capacity is the sum of the loads from all evaporators. Likewise, the total energy input is the sum of the work into all compressors. For multistage cycles, the expression for the coefficient of performance given in Equation (15) should be written as Licensed for single user. © 2017 ASHRAE, Inc. Fig. 12 Areas on T-s Diagram Representing Refrigerating Effect and Work Supplied for Theoretical Single-Stage Cycle Using Zeotropic Mixture as Refrigerant 2.4 THEORETICAL SINGLE-STAGE CYCLE USING ZEOTROPIC REFRIGERANT MIXTURE A practical method to approximate the Lorenz refrigeration cycle is to use a fluid mixture as the refrigerant and the four system components shown in Figure 8. When the mixture is not azeotropic and the phase change occurs at constant pressure, the temperatures change during evaporation and condensation and the theoretical single-stage cycle can be shown on T-s coordinates as in Figure 12. In comparison, Figure 10 shows the system operating with a pure simple substance or an azeotropic mixture as the refrigerant. Equations (14), (15), (39), (40), and (41) apply to this cycle and to conventional cycles with constant phase change temperatures. Equation (42) should be used as the reversible cycle COP in Equation (17). For zeotropic mixtures, the concept of constant saturation temperatures does not exist. For example, in the evaporator, the refrigerant enters at T4 and exits at a higher temperature T1. The temperature of saturated liquid at a given pressure is the bubble point and the temperature of saturated vapor at a given pressure is called the dew point. The temperature T3 in Figure 12 is at the bubble point at the condensing pressure and T1 is at the dew point at the evaporating pressure. Areas on a T-s diagram representing additional work and reduced refrigerating effect from a Lorenz cycle operating between the same two temperatures T1 and T3 with the same value for T can be analyzed. The cycle matches the Lorenz cycle most closely when counterflow heat exchangers are used for both the condenser and evaporator. In a cycle that has heat exchangers with finite thermal resistances and finite external fluid capacity rates, Kuehn and Gronseth (1986) showed that a cycle using a refrigerant mixture has a higher coefficient of performance than one using a simple pure substance as a refrigerant. However, the improvement in COP is usually small. Mixture performance can be improved further by reducing the heat exchangers' thermal resistance and passing fluids through them in a counterflow arrangement. 2.5 MULTISTAGE VAPOR COMPRESSION REFRIGERATION CYCLES Multistage or multipressure vapor compression refrigeration is used when several evaporators are needed at various temperatures, COP = Qi /Wnet (43) When compressors are connected in series, vapor between stages should be cooled to bring the vapor to saturated conditions before proceeding to the next stage of compression. Intercooling usually minimizes displacement of the compressors, reduces the work requirement, and increases the cycle's COP. If the refrigerant temperature between stages is above ambient, a simple intercooler that removes heat from the refrigerant can be used. If the temperature is below ambient, which is the usual case, the refrigerant itself must be used to cool the vapor. This is accomplished with a flash intercooler. Figure 13 shows a cycle with a flash intercooler installed. The superheated vapor from compressor I is bubbled through saturated liquid refrigerant at the intermediate pressure of the cycle. Some of this liquid is evaporated when heat is added from the superheated refrigerant. The result is that only saturated vapor at the intermediate pressure is fed to compressor II. A common approach is to operate the intercooler at about the geometric mean of the evaporating and condensing pressures. This operating point provides the same pressure ratio and nearly equal volumetric efficiencies for the two compressors. Example 4 illustrates the thermodynamic analysis of this cycle. Example 4. Determine the thermodynamic properties of the eight state points shown in Figure 13, mass flows, and COP of this theoretical multistage refrigeration cycle using R-134a. The saturated evaporator temperature is –20°C, the saturated condensing temperature is 30°C, and the refrigeration load is 50 kW. The saturation temperature of the refrigerant in the intercooler is 0°C, which is nearly at the geometric mean pressure of the cycle. Solution: Thermodynamic property data are obtained from the saturation and superheat tables for R-134a in Chapter 30. States 1, 3, 5, and 7 are obtained directly from the saturation table. State 6 is a mixture of liquid and vapor. The quality is calculated by h6 – h7 241.72 – 200 x6 = ---------------- = ------------------------------ = 0.21007 h3 – h7 398.60 – 200 Then, v6 = v7 + x6 (v3 – v7) = 0.000772 + 0.21007(0.06931 – 0.000772) = 0.01517 m3/kg s6 = s7 + x6 (s3 – s7) = 1.0 + 0.21007(1.7282 – 1.0) = 1.15297 kJ/(kg·K) Similarly for state 8, x8 = 0.12381, v8 = 0.01889 m3/kg, s8 = 1.00434 kJ/(kg·K) This file is licensed to John Murray (). Publication Date: 6/1/2017 Thermodynamics and Refrigeration Cycles 2.11 Licensed for single user. © 2017 ASHRAE, Inc. Fig. 14 Schematic of Real, Direct-Expansion, Single-Stage Mechanical Vapor-Compression Refrigeration System Rearranging and solving for m· 3 , h7 – h2 200 – 401.51 m· 3 = m· 2 ---------------- = 0.2680 --------------------------------------- = 0.3442 kg/s h6 – h3 241.72 – 398.60 · W I = m· 1 h 2 – h 1 = 0.2680 401.51 – 386.55 = 4.009 kW · W II = m· 3 h 4 – h 3 = 0.3442 418.68 – 398.60 Fig. 13 Schematic and Pressure-Enthalpy

Diagram for Dual-Compression, Dual-Expansion Cycle of Example 4 Table 2 Thermodynamic Property Values for Example 4 Temperature, Pressure, State °C kPa 1 2 3 4 5 6 7 8 –20.0 2.8 0.0 33.6 30.0 0.0 0.0 –20.0 132.73 292.80 292.80 770.20 770.20 292.80 292.80 132.73 Specific Volume, m3/kg Specific Enthalpy, kJ/kg Specific Entropy, kJ/(kg · K) 0.14739 0.07097 0.06931 0.02726 0.00084 0.01517 0.000772 0.01889 386.55 401.51 398.60 418.68 241.72 241.72 200.00 200.00 1.7413 1.7413 1.7282 1.7282 1.1435 1.15297 1.0000 1.00434 States 2 and 4 are obtained from the superheat tables by linear interpolation. The thermodynamic property data are summarized in Table 2. Mass flow through the lower circuit of the cycle is determined from an energy balance on the evaporator. · Qi 50 m · 1 = -------------- = ------------------------------------- = 0.2680 kg/s h1 – h8 386.55 – 200 m · 1 = m · 2 = m · 7 = m · 8 For the upper circuit of the cycle, m · 3 = m · 4 = m · 5 = m · 6 Assuming the intercooler has perfect external insulation, an energy balance on it is used to compute m · 3 . m · 6 h 6 + m · 2 h 2 = m · 7 h 7 + m · 3 h 3 = 6.912 kW · Qi 50 COP = -------------------- = 4.58 · · · = -------------------------------------4.009 + 6.912 W I + W II Examples 2 and 4 have the same refrigeration load and operate with the same evaporating and condensing temperatures. The twostage cycle in Example 4 has a higher COP and less work input than the single-stage cycle. Also, the highest refrigerant temperature leaving the compressor is about 34°C for the two-stage cycle versus about 38°C for the single-stage cycle. These differences are more pronounced for cycles operating at larger pressure ratios. 2.6 ACTUAL REFRIGERATION SYSTEMS Actual systems operating steadily differ from the ideal cycles considered in the previous sections in many respects. Pressure drops occur everywhere in the system except in the compression process. Heat transfers between the refrigerant and its environment in all components. The actual compression process differs substantially from isentropic compression. The working fluid is not a pure substance but a mixture of refrigerant and oil. All of these deviations from a theoretical cycle cause irreversibilities in the system. Each irreversibility requires additional power into the compressor. It is useful to understand how these irreversibilities are distributed throughout a real system; this insight can be useful when design changes are contemplated or operating conditions are modified. Example 5 illustrates how the irreversibilities can be computed in a real system and how they require additional compressor power to overcome. Input data have been rounded off for ease of computation. Example 5. An air-cooled, direct-expansion, single-stage mechanical vaporcompression refrigerator uses R-22 and operates under steady conditions. A schematic of this system is shown in Figure 14. Pressure drops occur in all piping, and heat gains or losses occur as indicated. Power input includes compressor power and the power required to operate both fans. The following performance data are obtained: This file is licensed to John Murray (). Publication Date: 6/1/2017 2.12 2017 ASHRAE Handbook—Fundamentals (SI) Table 3 Measured and Computed Thermodynamic Properties of R-22 for Example 5 Measured Pressure, Temperature, kPa °C State 1 2 3 4 5 6 7 8 9 310.0 304.0 1450.0 1435.0 1410.0 1405.0 320.0 –10.0 4.0 82.0 70.0 34.0 33.0 –12.8 Computed Specific Enthalpy, kJ/kg 402.08 406.25 454.20 444.31 421.46 240.13 240.13 Specific Entropy, kJ/(kg·K) Specific Volume, m3/kg 1.7810 1.7984 1.8165 1.7891 1.1400 1.1359 1.1561 0.07558 0.07946 0.02057 0.01970 0.00086 0.00086 0.01910 Second law · 11 2 Q· 1 2 = m · s 2 – s 1 – -------T0 = 0.04322 1.7984 – 1.7810 – 303.15 = 0.1575 W/K Licensed for single user. © 2017 ASHRAE, Inc. Fig. 15 Pressure-Enthalpy Diagram of Actual System and Theoretical Single-Stage System Operating Between Same Inlet Air Temperatures tR and t0 Ambient air temperature Refrigerated space temperature Refrigerant flow rate Compressor power input Condenser fan input Evaporator fan input t0 tR · Q evap · W comp · W CF · W EF = = = = = = 30°C –10°C 7.0 kW 0.25 kW 0.15 kW 0.11 kW Refrigerant pressures and temperatures are measured at the seven locations shown in Figure 14. Table 3 lists the measured and computed thermodynamic properties of the refrigerant, neglecting the dissolved oil. A pressure-enthalpy diagram of this cycle is shown in Figure 15 and is compared with a theoretical single-stage cycle operating between the air temperatures tR and t0. Compute the energy transfers to the refrigerant in each component of the system and determine the second-law irreversibility rate in each component. Show that the total irreversibility rate multiplied by the absolute ambient temperature is equal to the difference between the actual power input and the power required by a Carnot cycle operating between tR and t0 with the same refrigerating load. Solution: The mass flow of refrigerant is the same through all components, so it is only computed once through the evaporator. Each component in the system is analyzed sequentially, beginning with the evaporator. Equation (6) is used to perform a first-law energy balance on each component, and Equations (11) and (13) are used for the second-law analysis. Note that the temperature used in the second-law analysis is the absolute temperature. Compressor: Energy balance · · · Q 2 3 = m h 3 – h 2 + 2W3 = 0.04322 454.20 – 406.25 – 2.5 = – 0.4276 kW Second law · Q 2 3 · s – s = ------- m 2 3 2 T0 = 0.04322 1.8165 – 1.7984 – – 0.4276 303.15 = 0.2357 W/K Condenser: Energy balance · Q 3 4 = m · h4 – h3 = 0.04322 444.31 – 454.20 = – 0.4274 kW Second law · 34 = 0.04322 1.7891 – 1.8165 – – 0.4274 303.15 = 0.2258 W/K Condenser: Energy balance · 4Q5 = m · h 5 – h 4 = 0.04322 421.46 – 444.31 = – 0.9876 kW Second law · 4 5 = m · h 1 – h 7 = 7.0 kW· 4l 5 7 1 7.0 m = --- = 0.04322 kg/s 402.08 – 240.13 · Q 7 1 s m · s 1 – s 7 – -------T0 = 0.0747 W/K Liquid Line: Energy balance · 5Q6 7.0 = 0.04322 1.7810 – 1.1561 – ----------------- = 0.0747 W/K 263.15 Suction Line: Energy balance · Q9 1 = m h2 – h1 = 0.04322 406.25 – 402.08 = 0.1802 kW · 4Q5 = m · s 5 – s 4 – -------T0 = 0.04322 1.1400 – 1.7891 – – 8.7698 303.15 = 0.0043 W/K Second law 7l 1 · 3Q4 = m · s 4 – s 3 – -------T0 = m h6 – h5 = 0.04322 240.13 – 241.40 = – 0.0549 kW Second law · 5l 6 · 5Q6 = m · s 6 – s 5 – -------T0 = 0.04322 1.1359 – 1.1400 – – 0.0549 303.15 = 0.0039 W/K This file is licensed to John Murray (). Publication Date: 6/1/2017 Thermodynamics and Refrigeration Cycles 2.13 pressure drop causes a very small loss and refrigerant pressure drop causes a negligible loss. This finding indicates that promoting refrigerant heat transfer at the expense of increasing the pressure drop often improves performance. Using a thermoeconomic technique is required to determine the cost/benefits associated with reducing component irreversibilities. Expansion Device: Energy balance · Q = m · h7 – h6 = 0 6 7 Second law · 6l 7 = m · s 7 – s 6 = 0.04322(1.1561 – 1.1359) = 0.8730 W/K These results are summarized in Table 4. For the Carnot cycle, TR 263.15 COPCarnot = ---------------- = ----------------- = 6.579 T0 – TR 40 Q · 0 = 7.0 · W Carnot = -------------------- = 1.064 kW COP Carnot 6.579 The actual power requirement for the compressor is · · · W comp = m · W Carnot + I total T 0 = 1.064 + 0.7331 303.15 = 1.064 + -- = 2.4994 kW 1000 Licensed for single user. © 2017 ASHRAE, Inc. ABSORPTION REFRIGERATION CYCLES An absorption cycle is a heat-activated thermal cycle. It exchanges only thermal energy with its surroundings; no appreciable mechanical energy is exchanged. Furthermore, no appreciable conversion of heat to work or work to heat occurs in the cycle. Absorption cycles are used in applications where one or more of the exchanges of heat with the surroundings is the useful product (e.g., refrigeration, air conditioning, and heat pumping). The two great advantages of this type of cycle in comparison to other cycles with similar product are The Carnot power requirement for the 7 kW load is This result is within computational error of the measured power input to the compressor of 2.5 kW. The analysis in Example 5 can be applied to any actual vapor compression refrigeration system. The only required information for second-law analysis is the refrigerant thermodynamic state points and mass flow rates and the temperatures in which the system is exchanging heat. In this example, the extra compressor power required to overcome the irreversibility in each component is determined. The component with the largest loss is the compressor. This loss is due to motor inefficiency, friction losses, and irreversibilities caused by pressure drops, mixing, and heat transfer between the compressor and the surroundings. Unrestrained expansion in the expansion device is the next largest (also a large loss), but could be reduced by using an expander rather than a throttling process. An expander may be economical on large machines. All heat transfer irreversibilities on both the refrigerant side and the air side of the condenser and evaporator are included in the analysis. Refrigerant pressure drop is also included. Air-side pressure drop irreversibilities of the two heat exchangers are not included, but these are equal to the fan power requirements because all the fan power is dissipated as heat. An overall second-law analysis, such as in Example 5, shows the designer components with the most losses, and helps determine which components should be replaced or redesigned to improve performance. However, it does not identify the nature of the losses; this requires a more detailed second-law analysis of the actual processes in terms of fluid flow and heat transfer (Liang and Kuehn 1991). A detailed analysis shows that most irreversibilities associated with heat exchangers are due to heat transfer, whereas air-side Table 4 3. • No large, rotating mechanical equipment is required • Any source of heat can be used, including low-temperature sources (e.g., waste heat, solar heat) 3.1 IDEAL THERMAL CYCLE All absorption cycles include at least three thermal energy exchanges with their surroundings (i.e., energy exchange at three different temperatures). The highest- and lowest-temperature heat flows are in one direction, and the mid-temperature one (or two) is in the opposite direction. In the forward cycle, the extreme (hottest and coldest) heat flows are into the cycle. This cycle is also called the heat amplifier, heat pump, conventional cycle, or Type I cycle. When extreme-temperature heat flows are out of the cycle, it is called a reverse cycle, heat transformer, temperature amplifier, temperature booster, or Type II cycle. Figure 16 illustrates both types of thermal cycles. This fundamental constraint of heat flow into or out of the cycle at three or more different temperatures establishes the first limitation on cycle performance. By the first law (at steady state), Qhot + Qcold = –Qmid (positive heat quantities are into the cycle) The second law requires that Energy Transfers and Irreversibility Rates for Refrigeration System in Example 5 q, kW · W , kW · I , W/K · I · I total , % 7.0000 0.1802 –0.4276 –0.4274 –8.7698 –0.0549 0 0 0 2.5 0 0 0 0 0.0474 0.1575 2.1928 0.2258 8.8747 0.0039 0.8730 3 46 5 18 0 18 Totals –2.4995 2.5 4.7351 Component Evaporator Suction line Compressor Discharge line Condenser Liquid line Expansion device Fig. 16 Thermal Cycles (44) This file is licensed to John Murray (). Publication Date: 6/1/2017 2.14 2017 ASHRAE Handbook—Fundamentals (SI) Q hot Q cold Q mid -------- + --------- + -------- ≥ 0 T hot T cold T mid (45) with equality holding in the ideal case. From these two laws alone (i.e., without invoking any further assumptions) it follows that, for the ideal forward cycle, Q cold T hot – T mid T cold COPideal = ------- = -------------------------------------Q hot T mid – T cold T hot Licensed for single user. © 2017 ASHRAE, Inc. (47) This expression results from assigning all the entropy flow to the single temperature Tmid. The ideal COP for the four-temperature cycle requires additional assumptions, such as the relationship between the various heat quantities. Under the assumptions that Qcold = Qmid cold and Qhot = Qmid hot , the following expression results: T hot – T mid hot T cold T cold COPideal = --T hot T mid cold T mid hot 3.2 Qhot Qgen (48) WORKING-FLUID PHASE CHANGE CONSTRAINTS Absorption cycles require at least two working substances: a sorbent and a fluid refrigerant; these substances undergo phase changes. Given this constraint, many combinations are not achievable. The first result of invoking the phase change constraints is that the various heat fluxes assume known identities. As shown in Figure 17, the refrigerant phase changes occur in an evaporator and a condenser, and the sorbent phase changes in an absorber and a desorber (generator). (Note that two lines connect the evaporator to the absorber and the desorber to the condenser, with one indicating vapor flow, the second carryover of liquid. In both cases, the carry- (49) and the coldest heat is supplied to the evaporator: Qcold Qevap (46) The heat ratio Qcold /Qhot is commonly called the coefficient of performance (COP), which is the cooling realized divided by the driving heat supplied. Heat rejected to ambient may be at two different temperatures, creating a four-temperature cycle. The ideal COP of the fourtemperature cycle is also expressed by Equation (46), with Tmid signifying the entropic mean heat rejection temperature. In that case, Tmid is calculated as follows: Q mid hot + Q mid cold Tmid = ---Q mid hot Q mid cold ---------------- + ------------------T mid hot T mid cold over of liquid is detrimental to system performance.) For the forward absorption cycle, the highest-temperature heat is always supplied to the generator, (50) For the reverse absorption cycle (also called heat transformer or type II absorption cycle), the highest-temperature heat is rejected from the absorber, and the lowest-temperature heat is supplied to the generator. The second law requires that, for all known refrigerants and sorbents over pressure ranges of interest, and Qevap Qcond (51) Qgen Qabs (52) These two relations are true because the latent heat of phase change (vapor condensed phase) is relatively constant when far removed from the critical point. Thus, each heat input cannot be independently adjusted. The ideal single-effect forward-cycle COP expression is T gen – T abs T evap T cond COPideal ------------------------------------T cond – T evap T abs T gen (53) Equality holds only if the heat quantities at each temperature may be adjusted to specific values, which is not possible, as shown the following discussion. The third result of invoking the phase change constraint is that only three of the four temperatures Tevap, Tcond , Tgen, and Tabs may be independently selected. Practical liquid absorbents for absorption cycles have a significant negative deviation from behavior predicted by Raoult's law. This has the beneficial effect of reducing the required amount of absorbent recirculation, at the expense of reduced lift (Tcond – Tevap) and increased sorption duty. In practical terms, for most absorbents, Qabs /Qcond 1.2 to 1.3 (54) Tgen – Tabs 1.2(Tcond – Tevap) (55) and The net result of applying these approximations and constraints to the ideal-cycle COP for the single-effect forward cycle is T evap T cond Q cond COPideal 1.2 -- 0.8 T gen T abs Q abs (56) In practical terms, the temperature constraint reduces the ideal COP to about 0.9, and the heat quantity constraint further reduces it to about 0.8. Another useful result is Tgen min = Tcond + Tabs – Tevap Fig. 17 Single-Effect Absorption Cycle (57) where Tgen min is the minimum generator temperature necessary to achieve a given evaporator temperature. Alternative approaches are available that lead to nearly the same upper limit on ideal-cycle COP. For example, one approach equates the exergy production from a "driving" portion of the cycle to the exergy consumption in a "cooling" portion of the cycle (Tozer and James 1997). This leads to the expression This file is licensed to John Murray (). Publication Date: 6/1/2017 Thermodynamics and Refrigeration Cycles T evap T cond COPideal ------------ = ---------T gen T abs 2.15 (58) Another approach derives the idealized relationship between the two temperature differences that define the cycle: the cycle lift, defined previously, and drop (Tgen – Tabs). Licensed for single user. © 2017 ASHRAE, Inc. Temperature Glide One important limitation of simplified analysis of absorption cycle performance is that the heat quantities are assumed to be at fixed temperatures. In most actual applications, there is some temperature change (temperature glide) in the various fluids supplying or acquiring heat. It is most easily described by first considering situations wherein temperature glide is not present (i.e., truly isothermal heat exchanges). Examples are condensation or boiling of pure components (e.g., supplying heat by condensing steam). Any sensible heat exchange relies on temperature glide: for example, a circulating high-temperature liquid as a heat source; cooling water or air as a heat rejection medium; or circulating chilled glycol. Even latent heat exchanges can have temperature glide, as when a multicomponent mixture undergoes phase change. When the temperature glide of one fluid stream is small compared to the cycle lift or drop, that stream can be represented by an average temperature, and the preceding analysis remains representative. However, one advantage of absorption cycles is that they can maximize benefit from low-temperature, high-glide heat sources. That ability derives from the fact that the desorption process inherently embodies temperature glide, and hence can be tailored to match the heat source glide. Similarly, absorption also embodies glide, which can be made to match the glide of the heat rejection medium. Implications of temperature glide have been analyzed for power cycles (Ibrahim and Klein 1998), but not yet for absorption cycles. 3.3 WORKING FLUIDS Working fluids for absorption cycles fall into four categories, each requiring a different approach to cycle modeling and thermodynamic analysis. Liquid absorbents can be nonvolatile (i.e., vapor phase is always pure refrigerant, neglecting condensables) or volatile (i.e., vapor concentration varies, so cycle and component modeling must track both vapor and liquid concentration). Solid sorbents can be grouped by whether they are physisorbents (also known as adsorbents), for which, as for liquid absorbents, sorbent temperature depends on both pressure and refrigerant loading (bivariance); or chemisorbents (also known as complex compounds), for which sorbent temperature does not vary with loading, at least over small ranges. Beyond these distinctions, various other characteristics are either necessary or desirable for suitable liquid absorbent/refrigerant pairs, as follows: Absence of Solid Phase (Solubility Field). The refrigerant/ absorbent pair should not solidify over the expected range of composition and temperature. If a solid forms, it will stop flow and shut down equipment. Controls must prevent operation beyond the acceptable solubility range. Relative Volatility. The refrigerant should be much more volatile than the absorbent so the two can be separated easily. Otherwise, cost and heat requirements may be excessive. Many absorbents are effectively nonvolatile. Affinity. The absorbent should have a strong affinity for the refrigerant under conditions in which absorption takes place. Affinity means a negative deviation from Raoult's law and results in an activity coefficient of less than unity for the refrigerant. Strong affinity allows less absorbent to be circulated for the same refrigeration effect, reducing sensible heat losses, and allows a smaller liq- uid heat exchanger to transfer heat from the absorbent to the pressurized refrigerant/absorption solution. On the other hand, as affinity increases, extra heat is required in the generators to separate refrigerant from the absorbent, and the COP suffers. Pressure. Operating pressures, established by the refrigerant's thermodynamic properties, should be moderate. High pressure requires heavy-walled equipment, and significant electrical power may be needed to pump fluids from the low-pressure side to the highpressure side. Vacuum requires large-volume equipment and special means of reducing pressure drop in the refrigerant vapor paths. Stability. High chemical stability is required because fluids are subjected to severe conditions over many years of service. Instability can cause undesirable formation of gases, solids, or corrosive substances. Purity of all components charged into the system is critical for high performance and corrosion prevention. Corrosion. Most absorption fluids corrode materials used in construction. Therefore, corrosion inhibitors are used. Safety. Precautions as dictated by code are followed when fluids are toxic, inflammable, or at high pressure. Codes vary according to country and region. Transport Properties. Viscosity, surface tension, thermal diffusivity, and mass diffusivity are important characteristics of the refrigerant/absorbent pair. For example, low viscosity promotes heat and mass transfer and reduces pumping power. Latent Heat. The refrigerant latent heat should be high, so the circulation rate of the refrigerant and absorbent can be minimized. Environmental Soundness. The two parameters of greatest concern are the global warming potential (GWP) and the ozone depletion potential (ODP). For more information on GWP and ODP, see Chapter 29. No refrigerant/absorbent pair meets all requirements, and many requirements work at cross-purposes. For example, a greater solubility field goes hand in hand with reduced relative volatility. Thus, selecting a working pair is inherently a compromise. Water/lithium bromide and ammonia/water offer the best compromises of thermodynamic performance and have no known detrimental environmental effect (zero ODP and zero GWP). Ammonia/water meets most requirements, but its volatility ratio is low and it requires high operating pressures. Ammonia is also a Safety Code Group B2 fluid (ASHRAE Standard 34), which restricts its use indoors. Advantages of water/lithium bromide include high (1) safety, (2) volatility ratio, (3) affinity, (4) stability, and (5) latent heat. However, this pair tends to form solids and operates at deep vacuum. Because the refrigerant turns to ice at 0°C, it cannot be used for lowtemperature refrigeration. In fact, ice formation on desmisters can be observed at even slightly above 0°C. Lithium bromide (LiBr) crystallizes at moderate concentrations, as would be encountered in aircooled chillers, which ordinarily limits the pair to applications where the absorber is water cooled and the concentrations are lower. However, using a combination of salts as the absorbent can reduce this crystallization tendency enough to allow air cooling (Macriss 1968). Other disadvantages include low operating pressures and high viscosity. This is particularly detrimental to the absorption step; however, alcohols with a high relative molecular mass enhance LiBr absorption. Proper equipment design and additives can overcome these disadvantages. Other refrigerant/absorbent pairs are listed in Table 5 (Macriss and Zawacki 1989). Several appear suitable for certain cycles and may solve some problems associated with traditional pairs. However, information on properties, stability, and corrosion is limited. Also, some of the fluids are somewhat hazardous. Alcohols such as methanol are also being considered as refrigerants. A review of working pairs is given in Macriss et al. (1988). In addition, some absorption working pairs use a conventional refriger- This file is licensed to John Murray (). Publication Date: 6/1/2017 2.16 2017 ASHRAE Handbook—Fundamentals (SI) Licensed for single user. © 2017 ASHRAE, Inc. Table 5 Refrigerant/Absorbent Pairs Refrigerant Absorbents H2O Salts Alkali halides LiBr LiClO3 CaCl2 ZnCl2 ZnBr Alkali nitrates Alkali thiocyanates Bases Alkali hydroxides Acids H2SO4 H3PO4 NH3 H2O Alkali thiocyanates TFE (Organic) NMP E181 DMF Pyrrolidone SO2 Organic solvents ant, such as ammonia or water, but a solid absorbent. Examples are complex compounds in which ammonia is absorbed into solid salts that remain solid even when the salts absorb quantities of ammonia that exceed on a molar basis several times the moles of salt (Rockenfeller and Kirol 1989, 1996; Rockenfeller et al. 1992, 1993). 3.4 EFFECT OF FLUID PROPERTIES ON CYCLE PERFORMANCE Thermodynamic observations can predict general trends of how working fluids' properties affect a cycle's performance: in all four major heat exchangers (absorber, generator, condenser, and evaporator), the amount of heat exchanged is dominated by the latent heat of the refrigerant (i.e., the component that undergoes the phase change), when any phase change of the absorbent is neglected. There are two additional contributions to the absorber and generator: heat of mixing when the condensed refrigerant is mixed with the absorbent/ refrigerant solution, and heating or cooling of the refrigerant/absorbent mixture during the absorption or desorption process. Thus, the heat required in the generator can be estimated as the latent heat of the refrigerant plus the heat of mixing plus the heat required to heat the remaining absorbent/refrigerant solution. Both additional terms increase the generator heat requirement and thus reduce the overall cycle efficiency. Based on this observation, the ideal absorption working fluid should have high latent heat, no heat of mixing, and a low specific heat capacity. Furthermore, heat exchange in the solution heat exchanger is governed by the specific heat of the fluid mixture flowing through this device. Consequently, any ineffectiveness of the heat exchanger represents a loss in absorption-cycle performance that is directly related to the specific heat capacity of the fluid mixture, reinforcing the argument that a low specific heat capacity is desirable. Finally, losses in the expansion process of the refrigerant as it enters the evaporator are also governed by the specific heat of the refrigerant (preferably large) and its specific heat capacity (preferably small), so that as little refrigerant as possible evaporates as a result of the expansion process. Absorption working fluids should consist of refrigerants with large latent heat and absorbents with a small heat of mixing, and both absorbent and refrigerant should have as small a specific heat capacity as possible. Heat of mixing and latent heat are determined by functional groups within the molecule: specific heat capacity is minimized when the molecule is small and of low molecular mass. Therefore, ideal working fluids should be small molecules with as many functional groups as possible. This explains why ammonia and water are still the favored refrigerants to date for absorption cycles, and why organic fluids have not yet succeeded in commercial absorption cycle applications (because of their relatively large molecular mass). 3.5 ABSORPTION CYCLE REPRESENTATIONS The quantities of interest to absorption cycle designers are temperature, concentration, pressure, and enthalpy. The most useful plots use linear scales and plot the key properties as straight lines. Some of the following plots are used: • Absorption plots embody the vapor-liquid equilibrium of both the refrigerant and the sorbent. Plots on linear pressure-temperature coordinates have a logarithmic shape and hence are little used. • In the van't Hoff plot (ln P versus –1/T), the constant concentration contours plot as nearly straight lines. Thus, it is more readily constructed (e.g., from sparse data) in spite of the awkward coordinates. • The Dühring diagram (solution temperature versus reference temperature) retains the linearity of the van't Hoff plot but eliminates the complexity of nonlinear coordinates. Thus, it is used extensively (see Figure 20). The primary drawback is the need for a reference substance. • The Gibbs plot (solution temperature versus T ln P) retains most of the advantages of the Dühring plot (linear temperature coordinates, concentration contours are straight lines) but eliminates the need for a reference substance. • The Merkel plot (enthalpy versus concentration) is used to assist thermodynamic calculations and to solve the distillation problems that arise with volatile absorbents. It has also been used for basic cycle analysis. • Temperature/entropy coordinates are occasionally used to relate absorption cycles to their mechanical vapor compression counterparts. 3.6 CONCEPTUALIZING THE CYCLE The basic absorption cycle shown in Figure 17 must be altered in many cases to take advantage of the available energy. Examples include the following: (1) the driving heat is much hotter than the minimum required Tgen min: a multistage cycle boosts the COP; and (2) the driving heat temperature is below Tgen min: a different multistage cycle (half-effect cycle) can reduce the Tgen min. Multistage cycles have one or more of the four basic exchangers (generator, absorber, condenser, evaporator) present at two or more places in the cycle at different pressures or concentrations. A multieffect cycle is a special case of multistaging, signifying the number of times the driving heat is used in the cycle. Thus, there are several types of two-stage cycles: double-effect, half-effect, and tripleeffect. Two or more single-effect absorption cycles, such as shown in Figure 17, can be combined to form a multistage cycle by coupling any of the components. Coupling implies either (1) sharing component(s) between the cycles to form an integrated single hermetic cycle or (2) exchanging heat between components belonging to two hermetically separate cycles that operate at (nearly) the same temperature level. Figure 18 shows a double-effect absorption cycle formed by coupling the absorbers and evaporators of two single-effect cycles This file is licensed to John Murray (). Publication Date: 6/1/2017 Thermodynamics and Refrigeration Cycles Licensed for single user. © 2017 ASHRAE, Inc. into an integrated, single hermetic cycle. Heat is transferred between the high-pressure condenser and intermediate-pressure generator. The heat of condensation of the refrigerant (generated in the hightemperature generator) generates additional refrigerant in the lower-temperature generator. Thus, the prime energy provided to the high-temperature generator is cascaded (used) twice in the cycle, making it a double-effect cycle. With the generation of additional refrigerant from a given heat input, the cycle COP increases. Commercial water/lithium bromide chillers normally use this cycle. The cycle COP can be further increased by coupling additional components and by increasing the number of cycles that are combined. This way, several different multieffect cycles can be combined by pressure-staging and/or concentration-staging. The double-effect cycle, for example, is formed by pressure staging two single-effect cycles. Figure 19 shows twelve generic triple-effect cycles identified by Alefeld and Radermacher (1994). Cycle 5 is a pressure-staged cycle, and cycle 10 is a concentration-staged cycle. All other cycles are pressure and concentration staged. Cycle 1, which is a type of dualloop cycle, is the only cycle consisting of two loops that does not circulate absorbent in the low-temperature portion of the cycle. Each of the cycles shown in Figure 19 can be made with one, two, or sometimes three separate hermetic loops. Dividing a cycle into Fig. 18 Double-Effect Absorption Cycle 2.17 separate hermetic loops allows the use of a different working fluid in each loop. Thus, a corrosive and/or high-lift absorbent can be restricted to the loop where it is required, and a conventional additive-enhanced absorbent can be used in other loops to reduce system cost significantly. As many as 78 hermetic loop configurations can be synthesized from the twelve triple-effect cycles shown in Figure 19. For each hermetic loop configuration, further variations are possible according to the absorbent flow pattern (e.g., series or parallel), the absorption working pairs selected, and various other hardware details. Thus, literally thousands of distinct variations of the triple-effect cycle are possible. The ideal analysis can be extended to these multistage cycles (Alefeld and Radermacher 1994). A similar range of cycle variants is possible for situations calling for the half-effect cycle, in which the available heat source temperature is below Tgen min. 3.7 ABSORPTION CYCLE MODELING Analysis and Performance Simulation A physical-mathematical model of an absorption cycle consists of four types of thermodynamic equations: mass balances, energy balances, relations describing heat and mass transfer, and equations for thermophysical properties of the working fluids. As an example of simulation, Figure 20 shows a Dühring plot of a single-effect water/lithium bromide absorption chiller. The hotwater-driven chiller rejects waste heat from the absorber and the condenser to a stream of cooling water, and produces chilled water. A simulation of this chiller starts by specifying the assumptions (Table 6) and the design parameters and operating conditions at the design point (Table 7). Design parameters are the specified UA values and the flow regime (co/countercurrent/recurrent, pool, or film) of all heat exchangers (evaporator, condenser, generator, absorber, solution heat exchanger) and the flow rate of weak solution through the solution pump. One complete set of operating parameters could be the design point values of the chilled- and cooling water temperatures tchill in, tchill out , tcool in, tcool out , hot-water flow rate m· hot , and total cooling capacity Qe. With this information, a cycle simulation calculates the required hot-water temperatures; cooling-water flow rate; and temperatures, pressures, and concentrations at all internal state points. Some additional assumptions are made that reduce the number of unknown parameters. With these assumptions and the design parameters and operating conditions as specified in Table 7, the cycle simulation can be conducted by solving the following set of equations: Mass Balances Fig. 19 Generic Triple-Effect Cycles Fig. 20 Single-Effect Water/Lithium Bromide Absorption Cycle Dühring Plot This file is licensed to John Murray (). Publication Date: 6/1/2017 2.18 2017 ASHRAE Handbook—Fundamentals (SI) Table 6 Assumptions for Single-Effect Water/ Lithium Bromide Model (Figure 20) Table 8 Simulation Results for Single-Effect Water/Lithium Bromide Absorption Chiller Assumptions • Generator and condenser as well as evaporator and absorber are under same pressure • Refrigerant vapor leaving the evaporator is saturated pure water • Liquid refrigerant leaving the condenser is saturated pure water • Strong solution leaving generator is boiling • Refrigerant vapor leaving generator has equilibrium temperature of weak solution at generator pressure • Weak solution leaving absorber is saturated • No liquid carryover from evaporator • Flow restrictors are adiabatic • Pump is isentropic • No jacket heat losses • LMTD (log mean temperature difference) expression adequately estimates latent changes Licensed for single user. © 2017 ASHRAE, Inc. Table 7 Design Parameters and Operating Conditions for Single-Effect Water/Lithium Bromide Absorption Chiller Design Parameters Operating Conditions Evaporator UAevap = 319.2 kW/K, countercurrent film tchill in = 12°C tchill out = 6°C Condenser UAcond = 180.6 kW/K, countercurrent film tcool out = 35°C Absorber UAabs = 186.9 kW/K, countercurrent film-absorber tcool in = 27°C Generator UAgen = 143.4 kW/K, pool-countercurrent m· hot = 74.4 kg/s Solution UAsol = 33.8 kW/K, countercurrent m· gen = 12 kg/s General m· refr + m· strong = m· weak (59) m· strong strong = m· weak weak (60) Energy Balances · Q evap = m· refr h vapor, evap – h liq, cond = m· chill h chill in – h chill out · Q cond = m· refr h vapor, gen – h liq, cond = m· cool h cool out – h cool mean · Q abs = m· refr h vapor, evap + m· strong h strong, sol – m· weak h weak, abs · Q gen = m· refr h vapor, gen + m· strong h strong, gen – m· weak h weak, gen · Q hot = m· hot h hot in – h hot out (64) · Q sol = m· strong h strong, gen – h strong, sol = m· weak h weak, sol – h weak, abs Heat Transfer Equations Evaporator tvapor,evap = 1.8°C psat,evap = 0.697 kPa Condenser Tliq,cond = 46.2°C psat,cond = 10.2 kPa Absorber weak = 59.6% tweak = 40.7°C tstrong,abs = 49.9°C · Q evap = 2148 kW · m chill = 85.3 kg/s · Q cond = 2322 kW · m cool 1158.7 kg/s · Q abs = 2984 kW tcool,mean = 31.5°C Generator strong = 64.6% tstrong,gen = 103.5°C tweak,gen = 92.4°C tweak,sol = 76.1°C · Q gen = 3158 kW thot in = 125°C thot out = 115°C Solution tstrong,sol = 62.4°C tweak,sol = 76.1°C · Q sol = 825 kW = 65.4% General m· vapor = 0.93 kg/s m· strong = 11.06 kg/s COP = 0.68 t cool out – t cool mean · Q cond = UA cond -- t liq, cond – t cool out (66) (67) t strong, abs – t cool mean – t weak, abs – t cool in · Q abs = UA abs -- t strong, abs – t cool mean ln --- (68) t hot in – t strong, gen – t hot out – t weak, gen · Q gen = UA gen --- (69) t hot in – t strong, gen -- t hot out – t weak, gen t strong, sol – t weak, abs – t strong, gen – t weak, sol · Q sol = UA sol ln -- (70) t strong, sol – t weak, abs Fluid Property Equations at Each State Point · · Q gen = m· refr h vapor, gen + m· strong h strong, gen – m· weak h weak, gen · Q evap = UA evap --- t chill in – t vapor, evap ln -- t chill out – t vapor, evap · Q evap = 2148 kW Internal Parameters (65) Thermal Equations of State: Two-Phase Equilibrium: hwater (t,p), hsol (t, p,) twater,sat (p), tsol,sat (p,) The results are listed in Table 8. A baseline correlation for the thermodynamic data of the H2O/ LiBr absorption working pair is presented in Hellman and Grossman (1996). Thermophysical property measurements at higher temperatures are reported by Feuerecker et al. (1993). Additional high-temperature measurements of vapor pressure and specific heat appear in Langeliers et al. (2003), including correlations of the data. The UA values in Equations (66) to (70) account for the actual heat and mass transfer processes. This file is licensed to John Murray (). Publication Date: 6/1/2017 Thermodynamics and Refrigeration Cycles 2.19 Table 9 Inputs and Assumptions for Double-Effect Water/Lithium Bromide Model (Figure 21) Table 10 State Point Data for Double-Effect Lithium Bromide/Water Cycle of Figure 21 Inputs Capacity · Q evap 1760 kW Evaporator temperature t10 5.1°C Desorber solution exit temperature t14 170.7°C Condenser/absorber low temperature t1, t8 42.4°C Solution heat exchanger effectiveness 0.6 Assumptions • • • • • Licensed for single user. © 2017 ASHRAE, Inc. • • • • • • • • Steady state Refrigerant is pure water No pressure changes except through flow restrictors and pump State points at 1, 4, 8, 11, 14, and 18 are saturated liquid State point 10 is saturated vapor Temperature difference between high-temperature condenser and lowtemperature generator is 5 K Parallel flow Both solution heat exchangers have same effectiveness Upper level solution flow rate is selected such that upper condenser heat exactly matches lower generator heat requirement Flow restrictors are adiabatic Pumps are isentropic No jacket heat losses No liquid carryover from evaporator to absorber Vapor leaving both generators is at equilibrium temperature of entering solution stream Point 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 The results are given in Table 11. The double-effect (Figure 21) cycle can be performed in a manner similar to that for the single-effect cycle. Mass and energy balances of the model shown in Figure 21 were calculated using the inputs and assumptions listed in Table 9. The results are shown in Table 10. The COP is quite sensitive to several inputs and assumptions. In particular, the effectiveness of the solution heat exchangers and the driving temperature difference between the high-temperature condenser and the low-temperature generator influence the COP strongly. 117.7 117.7 182.3 247.3 177.2 177.2 2661.1 177.4 177.4 2510.8 201.8 201.8 301.2 378.8 270.9 270.9 2787.3 430.6 430.6 m, kg/s p, kPa 9.551 9.551 8.797 8.797 0.320 0.754 0.754 5.498 5.498 5.498 5.064 5.064 5.064 0.434 0.434 0.88 8.36 8.36 8.36 0.88 8.36 0.88 0.88 8.36 111.8 111.8 111.8 8.6 111.8 111.8 8.36 COP 1.21 · Q abs · Q gen , high-pressure · Q cond · Q evap · Q gen , low-pressure · Q shx1 · Q shx2 · W p1 · W p2 3.8 Fig. 21 Double-Effect Water/Lithium Bromide Absorption Cycle with State Points h, kJ/kg = = = = = = = = = = Q, Fraction 0.0 0.0 0.0 0.0 0.0 0.0 0.008 0.0 0.105 C, °C 42.4 42.4 75.6 97.8 8.8 53.2 85.6 42.4 5.5 0.0 85.6 85.6 136.7 170.7 110.9 99.1 155.7 102.8 42.4 x, % LiBr 59.5 59.5 59.5 64.6 64.6 0.0 0.0 59.5 59.5 5.6 64.6 64.6 0.0 0.0 1.0 0.195 58.6 0.000 2328 kW 1023 kW 905 kW 1760 kW 1472 kW 617 kW 546 kW 0.043 kW 0.346 kW AMMONIA/WATER ABSORPTION CYCLES Ammonia/water absorption cycles are similar to water/lithium bromide cycles, but with some important differences because of ammonia's lower latent heat compared to water, the volatility of the absorbent, and the different pressure and solubility ranges. Ammonia's latent heat is only about half that of water, so, for the same duty, the refrigerant and absorbent mass circulation rates are roughly double that of water/lithium bromide. As a result, the sensible heat loss associated with heat exchanger approaches is greater. Accordingly, ammonia/water cycles incorporate more heat exchange components to reclaim sensible heat, described in Hanna et al. (1995). The refrigerant heat exchanger (RHX), also known as refrigerant subcooler, which improves COP by about 8%, is the most important (Holldorff 1979). Next is the absorber heat exchanger (AHX), accompanied by a generator heat exchanger (GHX) (Phillips 1976). These either replace or supplement the traditional solution heat exchanger (SHX). These components would also benefit the water/lithium bromide cycle, except that the deep vacuum in that cycle makes them impractical there. The volatility of the water absorbent is also key. It makes the distinction between crosscurrent, cocurrent, and countercurrent mass exchange more important in all of the latent heat exchangers (Briggs 1971). It also requires a distillation column on the high-pressure side. When improperly implemented, this column can impose both cost and COP penalties. Those penalties are avoided by refluxing the column from an internal diabatic section [e.g., solution-cooled rectifier (SCR)] rather than with an external reflux pump. The high-pressure operating regime makes it impractical to achieve multieffect performance via pressure staging. On the other hand, the exceptionally wide solubility field facilitates concentration staging. The generator-absorber heat exchange (GAX) cycle is This file is licensed to John Murray (). Publication Date: 6/1/2017 2.20 2017 ASHRAE Handbook—Fundamentals (SI) Table 11 Inputs and Assumptions for Single-Effect Ammonia/Water Cycle (Figure 22) Table 12 State Point Data for Single-Effect Ammonia/Water Cycle (Figure 22) Point h, kJ/kg m, kg/s p, kPa 1 2 3 4 5 6 7 8 9 10 11 12 13 14 –57.2 –56.0 89.6 195.1 24.6 24.6 1349 178.3 82.1 82.1 1216 1313 1429 120.4 10.65 10.65 10.65 9.09 9.09 9.09 1.55 1.55 1.55 1.55 1.55 1.59 0.04 515.0 515.0 1461 1461 1461 1461 1461 1461 515.0 515.0 515.0 1461 1461 Inputs Capacity High-side pressure Low-side pressure Absorber exit temperature Generator exit temperature Rectifier vapor exit temperature Solution heat exchanger effectiveness Refrigerant heat exchanger effectiveness · Q evap phigh plow t1 t4 t7 shx rhx 1760 kW 1461 kPa 515 kPa 40.6°C 95°C 55°C 0.692 0.629 Assumptions • Steady state • No pressure changes except through flow restrictors and pump • States at points 1, 4, 8, 11, and 14 are saturated liquid • States at point 12 and 13 are saturated vapor • Flow restrictors are adiabatic • Pump is isentropic • No jacket heat losses • No liquid carryover from evaporator to absorber • Vapor leaving generator is at equilibrium temperature of entering solution stream COP = 0.571 trhx = 7.24 K tshx = 16.68 K Licensed for single user. © 2017 ASHRAE, Inc. rhx = 0.629 rhx = 0.692 · Q abs = 2869 kW · Q cond = 1862.2 kW 4. Fig. 22 Single-Effect Ammonia/Water Absorption Cycle an especially advantageous embodiment of concentration staging (Modahl and Hayes 1988). Ammonia/water cycles can equal the performance of water/ lithium bromide cycles. The single-effect or basic GAX cycle yields the same performance as a water/lithium bromide double-effect cycle; and the VX GAX cycle (Erickson and Rane 1994) yields the same performance as a water/lithium bromide triple-effect cycle. Additional advantages of the ammonia/water cycles include refrigeration capability, air-cooling capability, all mild steel construction, extreme compactness, and capability of direct integration into industrial processes. Between heat-activated refrigerators, gas-fired residential air conditioners, and large industrial refrigeration plants, this technology has accounted for the vast majority of absorption activity over the past century. Figure 22 shows the diagram of a typical single-effect ammonia/water absorption cycle. The inputs and assumptions in Table 11 are used to calculate a single-cycle solution, which is summarized in Table 12. Comprehensive correlations of the ammonia/water properties are available from Ibrahim and Klein (1993) and Tillner-Roth and Friend (1998a, 1998b), both of which are available as commercial software. Figure 33 in Chapter 30 of this volume was prepared using the Ibrahim and Klein correlation, which is also incorporated in REFPROP (National Institute of Standards and Technology). Transport properties for ammonia/water mixtures are available in IIR (1994) and in Melinder (1998). Q, Fraction 0.0 0.0 0.006 1.000 0.0 0.049 0.953 1.000 0.0 · Q evap · Q gen · Q rhx · Q rhx · Qr · Q shx · W t, °C x, Fraction NH3 40.56 40.84 72.78 95.00 57.52 55.55 55.00 37.82 17.80 5.06 6.00 30.57 79.15 79.15 0.50094 0.50094 0.50094 0.41612 0.41612 0.99809 0.99809 0.99809 0.99809 0.99809 0.99809 0.99809 0.50094 = 1760 kW = 3083 kW = 149 kW = 170 kW = 1550 kW = 6.88 kW ADSORPTION REFRIGERATION SYSTEMS Adsorption is the term frequently used for solid/vapor sorption systems in which the sorbent is a solid and the refrigerant is a gas. Although solid-vapor sorption systems actually comprise adsorption, absorption, and chemisorption, the term adsorption is often used to generically describe systems with liquid/vapor systems, such as lithium bromide/water and ammonia/water sorption pairs. Solid/vapor sorption media can be divided into two classes: • Bivariant systems thermodynamically behave identically to liquid/vapor systems, in which the two components [sorbent and sorbate (i.e., refrigerant)] define a thermodynamic equilibrium relation where vapor pressure, temperature, and refrigerant concentration are interrelated. These systems are commonly depicted in p-T-x or Dühring plots. • Monovariant systems thermodynamically behave like a single component substance, in which vapor pressure and temperature are interrelated via a traditional Clausius-Clapeyron relation, but are independent of refrigerant concentration within a certain refrigerant concentration range. These systems are often depicted in p-T-x or van't Hoff plots, in which each line represents a refrigerant concentration range. Typical examples of bivariant adsorption systems are zeolites, activated carbons, and silica gels. The common examples of monovariant systems are metal hydrides and coordinative complex compounds (including ammoniated and hydrated complex compounds). Practical ammonia or water refrigerant uptake concentrations for bivariant materials are typically lower than observed with monovariant systems. Monovariant metal hydride systems use hydrogen as the gaseous refrigerant. Although uptake concentrations are very low (typically in the single-digit mass percentage), the heat of reaction for metal hydrides is very high, yielding an overall energy density almost comparable to other solid/vapor sorption systems. Coordinative complex compounds can have refrigerant uptake that This file is licensed to John Murray (). Publication Date: 6/1/2017 Licensed for single user. © 2017 ASHRAE, Inc. Thermodynamics and Refrigeration Cycles exceeds the capability of other solid/gas systems. The concentration range within which temperature and vapor pressure are independent of the refrigerant concentration can exceed 50%; the heat of sorption is distinctively higher than for bivariant solid/gas systems, but much lower than observed with metal hydrides. The large refrigerant concentration range of constant vapor pressure, also referred to as the coordination sphere, lends itself to thermal energy storage applications. The most common coordinative complex compounds are ammoniated compounds using alkali, alkali/earth, or transition metal halides (e.g., strontium chloride, calcium chloride, calcium bromide). Although solid/vapor systems are designed to use solid sorbents and therefore do not carry the operational risk of equipment failure caused by solidifying from a liquid (as is the case with lithium bromide), some systems, particularly complex compounds, might melt at certain temperature/pressure/concentration conditions, which can cause operational failure. Refrigeration or heat pump cycles constructed with solid/vapor sorption systems are inherently batch-type cycles in which exothermic adsorption of refrigerant is followed by endothermic desorption of refrigerant. To obtain continuous refrigeration or heating, two or more solid sorbent pressure vessels (sorbers) need to operate at the same time and out of time sequence. The advantage of such cycles compared to liquid/vapor cycles is the fact that they do not require a solution makeup circuit with a solution pump; the disadvantage is the fact that the sorbent is firmly packed or situated in heat exchange hardware, inducing a higher thermal mass and requiring more involved means for recuperation. Advanced cycles with internal heat recovery, pressure staging, and temperature staging exist for solid/vapor systems. For more information, see Alefeld and Radermacher (1994). 4.1 = = = = = = = = m = p = Qw · Q = R s = S= t = T= u v = V W · W = = = x = z = Z = = = = = = cp COP g h I · I SYMBOLS specific heat at constant pressure, kJ/(kg·K) coefficient of performance local acceleration of gravity, m/s2 enthalpy, kJ/kg irreversibility, kJ/K irreversibility rate, kW/K mass, kg mass flow, kg/s pressure, kPa heat energy, kJ rate of heat flow, kJ/s ideal gas constant, (kPa·m3)/(kg·K) specific entropy, kJ/(kg·K) total entropy, kJ/K temperature, °C absolute temperature, K internal energy, kJ/kg specific volume, m3/kg velocity of fluid, m/s mechanical or shaft work, kJ rate of work, power, kW mass fraction (of either lithium bromide or ammonia) vapor quality (fraction) elevation above horizontal reference plane, m compressibility factor temperature difference, K heat exchanger effectiveness efficiency solution concentration density, kg/m3 Subscripts abs a b cg cr = absorber = condenser to generator 2.21 cond evap fg gen liq oh, 0 p R r rhx shx sat shx sol = = = = = = = = = = = = = = = = condenser evaporator liquid to vapor generator liquid high-temperature generator saturated liquid reference condition ambient pump refrigerating or evaporator conditions rectifier refrigerant heat exchanger saturated solution heat exchanger solution REFERENCES Alefeld, G., and R. 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This chapter introduces the basics of fluid mechanics related to HVAC processes, reviews pertinent flow processes, and presents a general discussion of single-phase fluid flow analysis. F Licensed for single user. © 2017 ASHRAE, Inc. 1. FLUID PROPERTIES Solids and fluids react differently to shear stress: solids deform only a finite amount, whereas fluids deform continuously until the stress is removed. Both liquids and gases are fluids, although the natures of their molecular interactions differ strongly in both degree of compressibility and formation of a free surface (interface) in liquid. In general, liquids are considered incompressible fluids; gases may range from compressible to nearly incompressible. Liquids have unbalanced molecular cohesive forces at or near the surface (interface), so the liquid surface tends to contract and has properties similar to a stretched elastic membrane. A liquid surface, therefore, is under tension (surface tension). Fluid motion can be described by several simplified models. The simplest is the ideal-fluid model, which assumes that the fluid has no resistance to shearing. Ideal fluid flow analysis is well developed [e.g., Schlichting (1979)], and may be valid for a wide range of applications. Viscous effects are taken into account by categorizing a fluid as either Newtonian or non-Newtonian. In Newtonian fluids, the rate of deformation is directly proportional to the shearing stress; most fluids in the HVAC industry (e.g., water, air, most refrigerants) can be treated as Newtonian. In non-Newtonian fluids, the relationship between the rate of deformation and shear stress is more complicated. Viscosity Fluid Flows per unit area required to slide one plate with velocity V parallel to the other is proportional to V/Y: F /A = (V /Y) (1) where the proportionality factor is the absolute or dynamic viscosity of the fluid. The ratio of F to A is the shearing stress , and V /Y is the lateral velocity gradient (Figure 1A). In complex flows, velocity and shear stress may vary across the flow field; this is expressed by dv = -----dy (2) water = 998 kg/m3 The velocity gradient associated with viscous shear for a simple case involving flow velocity in the x direction but of varying magnitude in the y direction is shown in Figure 1B. Absolute viscosity depends primarily on temperature. For gases (except near the critical point), viscosity increases with the square root of the absolute temperature, as predicted by the kinetic theory of gases. In contrast, a liquid's viscosity decreases as temperature increases. Absolute viscosities of various fluids are given in Chapter 33. Absolute viscosity has dimensions of force time/length2. At standard indoor conditions, the absolute viscosities of water and dry air (Fox et al. 2004) are water = 1.01 mN·s/m2 air = 18.1 N·s/m2 Density The density of a fluid is its mass per unit volume. The densities of air and water (Fox et al. 2004) at standard indoor conditions of 20°C and 101.325 kPa (sea-level atmospheric pressure) are air = 1.20 kg/m3 Viscosity Viscosity is the resistance of adjacent fluid layers to shear. A classic example of shear is shown in Figure 1, where a fluid is between two parallel plates, each of area A separated by distance Y. The bottom plate is fixed and the top plate is moving, which induces a shearing force in the fluid. For a Newtonian fluid, the tangential force F The preparation of this chapter is assigned to TC 1.3, Heat Transfer and Fluid Flow. 3.1 Copyright © 2017, ASHRAE Another common unit of viscosity is the centipoise [1 centipoise = 1 g/(sm) = 1 mPa]. At standard conditions, water has a viscosity close to 1.0 centipoise. In fluid dynamics, kinematic viscosity is sometimes used in lieu of absolute or dynamic viscosity. Kinematic viscosity is the ratio of absolute viscosity to density: = / This file is licensed to John Murray (). Publication Date: 6/1/2017 3.2 2017 ASHRAE Handbook—Fundamentals (SI) At standard indoor conditions, the kinematic viscosities of water and dry air (Fox et al. 2004) are water = 1.01 mm2/s air = 15.0 mm2/s The stoke (1 cm2/s) and centistoke (1 mm2/s) are common units for kinematic viscosity. 2. This section discusses fundamental principles of fluid flow for constant-property, homogeneous, incompressible fluids and introduces fluid dynamic considerations used in most analyses. Continuity in a Pipe or Duct Licensed for single user. © 2017 ASHRAE, Inc. (3) where m· is mass flow rate across the area normal to flow, v is fluid velocity normal to differential area dA, and is fluid density. Both and v may vary over the cross section A of the conduit. When flow is effectively incompressible (= constant) in a pipe or duct flow analysis, the average velocity is then m· = VA (4) and the mass flow rate can be written as m· = VA (4) (5) or where Q is volumetric flow rate. Bernoulli Equation and Pressure Variation in Flow Direction The Bernoulli equation is a fundamental principle of fluid flow analysis. It involves the conservation of momentum and energy along a streamline; it is not generally applicable across streamlines. Development is fairly straightforward. The first law of thermodynamics can apply to both mechanical flow energies (kinetic and potential energy) and thermal energies. The change in energy content E per unit mass of flowing fluid is a result of the work per unit mass w done on the system plus the heat per unit mass q absorbed or rejected: E = w + q (6) Fluid energy is composed of kinetic, potential (because of elevation z), and internal (u) energies. Per unit mass of fluid, the energy change relation between two sections of the system is v p ----- + gz + u = EM – --- + q 2 v ----- + gz + 2 p ---- = B (9) Alternative forms of this relation are obtained through multiplication by or division by g: 2 Conservation of mass applied to fluid flow in a conduit requires that mass not be created or destroyed. Specifically, the mass flow rate into a section of pipe must equal the mass flow rate out of that section of pipe if no mass is accumulated or lost (e.g., from leakage). This requires that vdA = constant (8) Where a constant (B) represents the sum of the kinetic energy, potential energy, internal energy, and flow work per unit mass flow rate. In cases with no work interaction, no heat transfer, and no viscous frictional forces that convert mechanical energy into internal energy, this expression is constant and is known as the Bernoulli constant B: 2 BASIC RELATIONS OF FLUID DYNAMICS m· = v2 p ------ + gz + u + --- EM q 2 2 (7) where g is the gravitational acceleration constant. Rearranging, the generalized Bernoulli equation: v p + -------- + gz = B 2 (10) p v2 B + ------ + ----- + z = ----- 2g g (11) where = g is the weight density (= weight/volume versus = mass/volume). Note that Equations (9) to (11) assume no frictional losses. The units in the first form of the Bernoulli equation [Equation (9)] are energy per unit mass; in Equation (10), energy per unit volume; in Equation (11), energy per unit weight, usually called head. Note that the units for head reduce to just length [i.e., (N·m)/N to m]. In gas flow analysis, Equation (10) is often used, and gz is negligible. Equation (10) should be used when density variations occur. For liquid flows, Equation (11) is commonly used. Identical results are obtained with the three forms if the units are consistent and fluids are homogeneous. Many systems of pipes, ducts, pumps, and blowers can be considered as one-dimensional flow along a streamline (i.e., variation in velocity across the pipe or duct is ignored, and local velocity v = average velocity V). When v varies significantly across the cross section, the kinetic energy term in the Bernoulli constant B is expressed as V 2/2, where the kinetic energy factor (> 1) expresses the ratio of the true kinetic energy of the velocity profile to that of the average velocity. For laminar flow in a wide rectangular channel, = 1.54, and in a pipe, = 2.0. For turbulent flow in a duct, 1. Heat transfer q may often be ignored. Conversion of mechanical energy to internal energy u may be expressed as a loss EL. The change in the Bernoulli constant (B = B2 – B1) between stations 1 and 2 along the conduit can be expressed as 2 p V ---- + ------ + gz + EM – EL = 2 1 2 p V   ---- + ------ + gz 2 1 2 (12) or, by dividing by g, in the form of 2 p V V ------ + --- + z + HM – HL = ---- + ----- + z + HM = --- + z 1 2g 2g 2 (13) Note that Equation (12) has units of energy per mass, whereas each term in Equation (13) has units of energy per unit weight, or head. The terms EM and EL are defined as positive, where gEM represents energy added to the conduit flow by pumps or blowers. A turbine or fluid motor thus has a negative HM or EM. The simple system in Figure 2 Bernoulli Equation (13) The total head or the sum of terms in the parentheses of Bernoulli's equation is the sum of position 1 (pressure head plus velocity head plus elevation head) plus the head added by a This file is licensed to John Murray (). Publication Date: 6/1/2017 Fluid Flow 3.3 Fig. 2 Dimensions for Steady, Fully Developed Laminar Flow Equations pump (HM) minus the head lost through friction (HL) is the total head at station 2. Licensed for single user. © 2017 ASHRAE, Inc. Laminar Flow When real-fluid effects of viscosity or turbulence are included, the continuity relation in Equation (5) is not changed, but V must be evaluated from the integral of the velocity profile, using local velocities. In fluid flow past fixed boundaries, velocity at the boundary is zero, velocity gradients exist, and shear stresses are produced. The equations of motion then become complex, and exact solutions are difficult to find except in simple cases for laminar flow between flat plates, between rotating cylinders, or within a pipe or tube. For steady, fully developed laminar flow between two parallel plates (Figure 2), shear stress varies linearly with distance y from the centerline (transverse to the flow; y = 0 in the center of the channel). For a wide rectangular channel 2b tall, can be written as y dv = w ---- = dy b (14) where w is wall shear stress [b(dp/ds)], and s is flow direction. Because velocity is zero at the wall (y = b), Equation (14) can be integrated to yield b – y dp v = ------------- 2 ds 2 (15) (16) A parabolic velocity profile can also be derived for a pipe of radius R. V is 1/2 of the maximum velocity, and the pressure drop can be written as 8V dp ------ = – -- ds R2 Velocity Fluctuation at Point in Turbulent Flow Turbulence is characterized by the root mean square (RMS) of the instantaneous variation in velocity about its mean. Turbulence causes the fluid to transfer momentum, heat, and mass very rapidly across the flow. Laminar and turbulent flows can be differentiated using the Reynolds number Re, which is a dimensionless relative ratio of inertial forces to viscous forces: ReL = VL/ (18) where L is the characteristic length scale and is the kinematic viscosity of the fluid. In flow through pipes, tubes, and ducts, the characteristic length scale is the hydraulic diameter Dh, given by Dh = 4A/Pw (19) where A is the cross-sectional area of the pipe, duct, or tube, and Pw is the wetted perimeter. For a round pipe, Dh equals the pipe diameter. In general, laminar flow in pipes or ducts exists when the Reynolds number (based on Dh)

is less than 2300. Fully turbulent flow exists when ReDh > 10 000. For 2300 < ReDh < 10 000, transitional flow exists, and predictions are unreliable. 3. BASIC FLOW PROCESSES Wall Friction The resulting parabolic velocity profile in a wide rectangular channel is commonly called Poiseuille flow. Maximum velocity occurs at the centerline (y = 0), and the average velocity V is 2/3 of the maximum velocity. From this, the longitudinal pressure drop in terms of V can be written as 3V dp ------ = – --------- ds b2 Fig. 3 (17) Turbulence Fluid flows are generally turbulent, involving random perturbations or fluctuations of the flow (velocity and pressure), characterized by an extensive hierarchy of scales or frequencies (Robertson 1963). Flow disturbances that are not chaotic but have some degree of periodicity (e.g., the oscillating vortex trail behind bodies) have been erroneously identified as turbulence. Only flows involving random perturbations without any order or periodicity are turbulent; velocity in such a flow varies with time or locale of measurement (Figure 3). Turbulence can be quantified statistically. The velocity most often used is the time-averaged velocity. The strength of turbulence At the boundary of real-fluid flow, the relative tangential velocity at the fluid surface is zero. Sometimes in turbulent flow studies, velocity at the wall may appear finite and nonzero, implying a fluid slip at the wall. However, this is not the case; the conflict results from difficulty in velocity measurements near the wall (Goldstein 1938). Zero wall velocity leads to high shear stress near the wall boundary, which slows adjacent fluid layers. Thus, a velocity profile develops near a wall, with velocity increasing from zero at the wall to an exterior value within a finite lateral distance. Laminar and turbulent flow differ significantly in their velocity profiles. Turbulent flow profiles are flat and laminar profiles are more pointed (Figure 4). As discussed, fluid velocities of the turbulent profile near the wall must drop to zero more rapidly than those of the laminar profile, so shear stress and friction are much greater in turbulent flow. Fully developed conduit flow may be characterized by the pipe factor, which is the ratio of average to maximum (centerline) velocity. Viscous velocity profiles result in pipe factors of 0.667 and 0.50 for wide rectangular and axisymmetric conduits. Figure 5 indicates much higher values for rectangular and circular conduits for turbulent flow. Because of the flat velocity profiles, the kinetic energy factor in Equations (12) and (13) ranges from 1.01 to 1.10 for fully developed turbulent pipe flow. Boundary Layer The boundary layer is the region close to the wall where wall friction affects flow. Boundary layer thickness (usually denoted by is thin compared to downstream flow distance. For external flow over a body, fluid velocity varies from zero at the wall to a maximum at This file is licensed to John Murray (). Publication Date: 6/1/2017 3.4 2017 ASHRAE Handbook—Fundamentals (SI) Fig. 4 Velocity Profiles of Flow in Pipes Licensed for single user. © 2017 ASHRAE, Inc. Fig. 7 Boundary Layer Flow to Separation Fig. 5 Pipe Factor for Flow in Conduits Fig. 8 Geometric Separation, Flow Development, and Loss in Flow Through Orifice Fig. 6 Flow in Conduit Entrance Region distance from the wall. Boundary layers are generally laminar near the start of their formation but may become turbulent downstream. A significant boundary-layer occurrence exists in a pipeline or conduit following a well-rounded entrance (Figure 6). Layers grow from the walls until they meet at the center of the pipe. Near the start of the straight conduit, the layer is very thin and most likely laminar, so the uniform velocity core outside has a velocity only slightly greater than the average velocity. As the layer grows in thickness, the slower velocity near the wall requires a velocity increase in the uniform core to satisfy continuity. As flow proceeds, the wall layers grow (and centerline velocity increases) until they join, after an entrance length Le. Applying the Bernoulli relation of Equation (10) to core flow indicates a decrease in pressure along the layer. Ross (1956) shows that, although the entrance length Le is many diameters, the length in which pressure drop significantly exceeds that for fully developed flow is on the order of 10 hydraulic diameters for turbulent flow in smooth pipes. In more general boundary-layer flows, as with wall layer development in a diffuser or for the layer developing along the surface of a strut or turning vane, pressure gradient effects can be severe and may even lead to boundary layer separation. When the outer flow velocity (v1 in Figure 7) decreases in the flow direction, an adverse pressure gradient can cause separation, as shown in the figure. Downstream from the separation point, fluid backflows near the wall. Separation is caused by frictional velocity (thus local kinetic energy) reduction near the wall. Flow near the wall no longer has energy to move into the higher pressure imposed by the decrease in v1 at the edge of the layer. The locale of this separation is difficult to predict, especially for the turbulent boundary layer. Analyses verify the experimental observation that a turbulent boundary layer is less subject to separation than a laminar one because of its greater kinetic energy. Flow Patterns with Separation In technical applications, flow with separation is common and often accepted if it is too expensive to avoid. Flow separation may be geometric or dynamic. Dynamic separation is shown in Figure 7. Geometric separation (Figures 8 and 9) results when a fluid stream passes over a very sharp corner, as with an orifice; the fluid generally leaves the corner irrespective of how much its velocity has been reduced by friction. For geometric separation in orifice flow (Figure 8), the outer streamlines separate from the sharp corners and, because of fluid inertia, contract to a section smaller than the orifice opening. The smallest section is known as the vena contracta and generally has a limiting area of about six-tenths of the orifice opening. After the vena contracta, the fluid stream expands rather slowly through turbulent or laminar interaction with the fluid along its sides. Outside the jet, fluid velocity is comparatively small. Turbulence helps spread out the jet, increases losses, and brings the velocity distribution back to a more uniform profile. Finally, downstream, the velocity profile returns to the fully developed flow of Figure 4. The entrance and exit profiles can profoundly affect the vena contracta and pressure drop (Coleman 2004). This file is licensed to John Murray (). Publication Date: 6/1/2017 Licensed for single user. © 2017 ASHRAE, Inc. Fluid Flow Other geometric separations (Figure 9) occur in conduits at sharp entrances, inclined plates or dampers, or sudden expansions. For these geometries, a vena contracta can be identified; for sudden expansion, its area is that of the upstream contraction. Ideal-fluid theory, using free streamlines, provides insight and predicts contraction coefficients for valves, orifices, and vanes (Robertson 1965). These geometric flow separations produce large losses. To expand a flow efficiently or to have an entrance with minimum losses, design the device with gradual contours, a diffuser, or a rounded entrance. Flow devices with gradual contours are subject to separation that is more difficult to predict, because it involves the dynamics of boundary-layer growth under an adverse pressure gradient rather than flow over a sharp corner. A diffuser is used to reduce the loss in expansion; it is possible to expand the fluid some distance at a gentle angle without difficulty, particularly if the boundary layer is turbulent. Eventually, separation may occur (Figure 10), which is frequently asymmetrical because of irregularities. Downstream flow involves flow reversal (backflow) and excess losses. Such separation is commonly called stall (Kline 1959). Larger expansions may use splitters that divide the diffuser into smaller sections that are less likely to have separations (Moore and Kline 1958). Another technique for controlling separation is to bleed some low-velocity fluid near the wall (Furuya et al. 1976). Alternatively, Heskested (1970) shows that suction at the corner of a sudden expansion has a strong positive effect on geometric separation. Drag Forces on Bodies or Struts Bodies in moving fluid streams are subjected to appreciable fluid forces or drag. Conventionally, the drag force FD on a body can be expressed in terms of a drag coefficient CD: V 2 FD = CD A ----- (20) 2 where A is the projected (normal to flow) area of the body. The drag coefficient CD is a strong function of the body's shape and angularity, and the Reynolds number of the relative flow in terms of the body's characteristic dimension. 3.5 For Reynolds numbers of 103 to 105, the CD of most bodies is constant because of flow separation, but above 105, the CD of rounded bodies drops suddenly as the surface boundary layer undergoes transition to turbulence. Typical CD values are given in Table 1; Hoerner (1965) gives expanded values. Nonisothermal Effects When appreciable temperature variations exist, the primary fluid properties (density and viscosity) may no longer assumed to be constant, but vary across or along the flow. The Bernoulli equation [Equations (9) to (11)] must be used, because volumetric flow is not constant. With gas flows, the thermodynamic process involved must be considered. In general, this is assessed using Equation (9), written as dp V 2 ------ + ------ + gz = B 2 (21) Effects of viscosity variations also appear. In nonisothermal laminar flow, the parabolic velocity profile (see Figure 4) is no longer valid. In general, for gases, viscosity increases with the square root of absolute temperature; for liquids, viscosity decreases with increasing temperature. This results in opposite effects. For fully developed pipe flow, the linear variation in shear stress from the wall value w to zero at the centerline is independent of the temperature gradient. In the section on Laminar Flow, is defined as = (y/b) w , where y is the distance from the centerline and 2b is the wall spacing. For pipe radius R = D/2 and distance from the wall y = R – r (Figure 11), then = w (R – y)/R. Then, solving Equation (2) for the change in velocity yields Table 1 Body Shape Sphere Disk Streamlined strut Circular cylinder Elongated rectangular strut Square strut Drag Coefficients 103 < Re < 2 105 Re > 3 105 0.36 to 0.47 1.12 0.1 to 0.3 1.0 to 1.1 1.0 to 1.2 ~2.0 ~0.1 1.12 < 0.1 0.35 1.0 to 1.2 ~2.0 Fig. 9 Examples of Geometric Separation Encountered in Flows in Conduits Fig. 10 Separation in Flow in Diffuser Fig. 11 Effect of Viscosity Variation on Velocity Profile of Laminar Flow in Pipe This file is licensed to John Murray (). Publication Date: 6/1/2017 3.6 2017 ASHRAE Handbook—Fundamentals (SI) w w R – y dv = --------------------- r dr - dy = – ----- R R (22) When fluid viscosity is lower near the wall than at the center (because of external heating of liquid or cooling of gas by heat transfer through the pipe wall), the velocity gradient is steeper near the wall and flatter near the center, so the profile is generally flattened. When liquid is cooled or gas is heated, the velocity profile is more pointed for laminar flow (Figure 11). Calculations for such flows of gases and liquid metals in pipes are in Deissler (1951). Occurrences in turbulent flow are less apparent than in laminar flow. If enough heating is applied to gaseous flows, the viscosity increase can cause reversion to laminar flow. Buoyancy effects and the gradual approach of the fluid temperature to equilibrium with that outside the pipe can cause considerable variation in the velocity profile along the conduit. Colborne and Drobitch (1966) found the pipe factor for upward vertical flow of hot air at a Re < 2000 reduced to about 0.6 at 40 diameters from the entrance, then increased to about 0.8 at 210 diameters, and finally decreased to the isothermal value of 0.5 at the end of 320 diameters. Licensed for single user. © 2017 ASHRAE, Inc. 4. FLOW ANALYSIS Fluid flow analysis is used to correlate pressure changes with flow rates and the nature of the conduit. For a given pipeline, either the pressure drop for a certain flow rate, or the flow rate for a certain pressure difference between the ends of the conduit, is needed. Flow analysis ultimately involves comparing a pump or blower to a conduit piping system for evaluating the expected flow rate. Generalized Bernoulli Equation Internal energy differences are generally small, and usually the only significant effect of heat transfer is to change the density . For gas or vapor flows, use the generalized Bernoulli equation in the pressure-over-density form of Equation (12), allowing for the thermodynamic process in the pressure-density relation: 1 ------ + 1 -----2- + EM = 2 -----2- + EL 2 – dp 2 V1 2 V2 (23) Elevation changes involving z are often negligible and are dropped. The pressure form of Equation (10) is generally unacceptable when appreciable density variations occur, because the volumetric flow rate differs at the two stations. This is particularly serious in frictionloss evaluations where the density usually varies over considerable lengths of conduit (Benedict and Carlucci 1966). When the flow is essentially incompressible, Equation (20) is satisfactory. Example 1. Specify a blower to produce isothermal airflow of 200 L/s through a ducting system (Figure 12). Accounting for intake and fitting losses, equivalent conduit lengths are 18 and 50 m, and flow is isother- mal. Pressure at the inlet (station 1) and following the discharge (station 4), where velocity is zero, is the same. Frictional losses HL are evaluated as 7.5 m of air between stations 1 and 2, and 72.3 m between stations 3 and 4. Solution: The following form of the generalized Bernoulli relation is used in place of Equation (12), which also could be used: (p1/1 g) + 1(V12/2g) + z1 + HM = (p2 /2 g) + 2(V22/2g) + z2 + HL The term V12/2g (24) can be calculated as follows: D 0.250 A1 = ---- = ------------ = 0.0491 m2 2 2 0.200 V1 = Q/A1 = ---------------- = 4.07 m/s 0.0491 2 2 V12/2g = (4.07)2/2(9.8) = 0.846 m (25) V22/2g The term can be calculated in a similar manner. In Equation (24), HM is evaluated by applying the relation between any two points on opposite sides of the blower. Because conditions at stations 1 and 4 are known, they are used, and the location-specifying subscripts on the right side of Equation (24) are changed to 4. Note that p1 = p4 = p, 1 = 4 = , and V1 = V4 = 0. Thus, (p/g) + 0 + 0.61 + HM = (p/g) + 0 + 3 + (7.5 + 72.3) (26) so HM = 82.2 m of air. For standard air (= 1.20 kg/m3), this corresponds to 970 Pa. The pressure difference measured across the blower (between stations 2 and 3) is often taken as HM. It can be obtained by calculating the static pressure at stations 2 and 3. Applying Equation (24) successively between stations 1 and 2 and between 3 and 4 gives (p1/g) + 0 + 0.61 + 0 = (p2 /g) + (1.06 0.846) + 0 + 7.5 (p3 /g) + (1.03 2.07) + 0 + 0 = (p4 /g) + 0 + 3 + 72.3 (27) where just ahead of the blower is taken as 1.06, and just after the blower as 1.03; the latter value is uncertain because of possible uneven discharge from the blower. Static pressures p1 and p4 may be taken as zero gage. Thus, p2 /g = –7.8 m of air p3 /g = 73.2 m of air (28) The difference between these two numbers is 81 m, which is not the HM calculated after Equation (24) as 82.2 m. The apparent discrepancy results from ignoring velocity at stations 2 and 3. Actually, HM is HM = (p3 /g) + 3(V32 /2g) – [(p2/g) + 2(V22 /2g)] = 73.2 + (1.03 2.07) – [–7.8 + (1.06 0.846)] = 75.3 – (–6.9) = 82.2 m (29) The required blower energy is the same, no matter how it is evaluated. It is the specific energy added to the system by the machine. Only when the conduit size and velocity profiles on both sides of the machine are the same is EM or HM simply found from p = p3 – p2. Conduit Friction Fig. 12 Blower and Duct System for Example 1 The loss term EL or HL of Equation (12) or (13) accounts for friction caused by conduit-wall shearing stresses and losses from conduit-section changes. HL is the head loss (i.e., loss of energy per unit weight). In real-fluid flow, a frictional shear occurs at bounding walls, gradually influencing flow further away from the boundary. A lateral velocity profile is produced and flow energy is converted into heat (fluid internal energy), which is generally unrecoverable (a loss). This loss in fully developed conduit flow is evaluated using the Darcy-Weisbach equation: This file is licensed to John Murray (). Publication Date: 6/1/2017 Licensed for single user. © 2017 ASHRAE, Inc. Fluid Flow 3.7 Fig. 13 Relation Between Friction Factor and Reynolds Number (based on Moody 1944) L V 2 H L = f ---- ------ f D 2g (30) where L is the length of conduit of diameter D and f is the DarcyWeisbach friction factor. Sometimes a numerically different relation is used with the Fanning friction factor (1/4 of the Darcy friction factor f). The value of f is nearly constant for turbulent flow, varying only from about 0.01 to 0.05. For fully developed laminar-viscous flow in a pipe, loss is evaluated from Equation (17) as follows: 2 L 8V 32LV 64 L V H L f = ------ --= = -- 2 g R 2 VD D 2g D g (31) where Re = VD/v and f = 64/Re. Thus, for laminar flow, the friction factor varies inversely with the Reynolds number. The value of 64/ Re varies with channel shape. A good summary of shape factors is provided by Incropera and DeWitt (2002). With turbulent flow, friction loss depends not only on flow conditions, as characterized by the Reynolds number, but also on the roughness height of the conduit wall surface. The variation is complex and is expressed in diagram form (Moody 1944), as shown in Figure 13. Historically, the Moody diagram has been used to determine friction factors, but empirical relations suitable for use in modeling programs have been developed. Most are applicable to limited ranges of Reynolds number and relative roughness. Churchill (1977) developed a relationship that is valid for all ranges of Reynolds numbers, and is more accurate than reading the Moody diagram: 8 f = 8 ----------- Re D h 12 1 + ---------------------1.5 A + B 1 12 1 -- A = 2.457 ln 0.9 7 Re + 0.27 D D h h 37 53016 B = --------------- Re D h (32a) 16 (32b) (32c) Inspection of the Moody diagram indicates that, for high Reynolds numbers and relative roughness, the friction factor becomes independent of the Reynolds number in a fully rough flow or fully turbulent regime. A transition region from laminar to turbulent flow occurs when 2000 < Re < 10 000. Roughness height , which may increase with conduit use, fouling, or aging, is usually tabulated for different types of pipes as shown in Table 2. Noncircular Conduits. Air ducts are often rectangular in cross section. The equivalent circular conduit corresponding to the noncircular conduit must be found before the friction factor can be determined. For turbulent flow, hydraulic diameter Dh is substituted for D in Equation (30) and in the Reynolds number. Noncircular duct friction This file is licensed to John Murray (). Publication Date: 6/1/2017 3.8 2017 ASHRAE Handbook—Fundamentals (SI) Table 2 Table 3 Fitting Loss Coefficients of Turbulent Flow Effective Roughness of Conduit Surfaces Fitting Geometry Entrance Sharp Well-rounded P g K = ---------------2 V 2g 0.5 0.05 Contraction Sharp (D2/D1 = 0.5) 0.38 90° elbow Miter Short radius Long radius Miter with turning vanes 1.3 0.90 0.60 0.2 Globe valve Angle valve Gate valve Open Open Open 75% open 50% open 25% open Closed 10 5 0.19 to 0.22 1.10 3.6 28.8 Straight-through flow Flow through branch 0.5 1.8 , m Material Commercially smooth brass, lead, copper, or plastic pipe Steel and wrought iron Galvanized iron or steel Cast iron 1.52 46 152 259 can be evaluated to within 5% for all except very extreme cross sections (e.g., tubes with deep grooves or ridges). A more refined method for finding the equivalent circular duct diameter is given in Chapter 13. With laminar flow, the loss predictions may be off by a factor as large as two. Licensed for single user. © 2017 ASHRAE, Inc. Valve, Fitting, and Transition Losses Valve and section changes (contractions, expansions and diffusers, elbows, bends, or tees), as well as entrances and exits, distort the fully developed velocity profiles (see Figure 4) and introduce extra flow losses that may dissipate as heat into pipelines or duct systems. Valves, for example, produce such extra losses to control the fluid flow rate. In contractions and expansions, flow separation as shown in Figures 9 and 10 causes the extra loss. The loss at rounded entrances develops as flow accelerates to higher velocities; this higher velocity near the wall leads to wall shear stresses greater than those of fully developed flow (see Figure 6). In flow around bends, velocity increases along the inner wall near the start of the bend. This increased velocity creates a secondary fluid motion in a double helical vortex pattern downstream from the bend. In all these devices, the disturbance produced locally is converted into turbulence and appears as a loss in the downstream region. The return of a disturbed flow pattern into a fully developed velocity profile may be quite slow. Ito (1962) showed that the secondary motion following a bend takes up to 100 diameters of conduit to die out but the pressure gradient settles out after 50 diameters. In a laminar fluid flow following a rounded entrance, the entrance length depends on the Reynolds number: Le /D = 0.06 Re (33) At Re = 2000, Equation (33) shows that a length of 120 diameters is needed to establish the parabolic velocity profile. The pressure gradient reaches the developed value of Equation (30) in fewer flow diameters. The additional loss is 1.2V 2/2g; the change in profile from uniform to parabolic results in a loss of 1.0V 2/2g (because = 2.0), and the remaining loss is caused by the excess friction. In turbulent fluid flow, only 80 to 100 diameters following the rounded entrance are needed for the velocity profile to become fully developed, but the friction loss per unit length reaches a value close to that of the fully developed flow value more quickly. After six diameters, the loss rate at a Reynolds number of 105 is only 14% above that of fully developed flow in the same length, whereas at 107, it is only 10% higher (Robertson 1963). For a sharp entrance, flow separation (see Figure 9) causes a greater disturbance, but fully developed flow is achieved in about half the length required for a rounded entrance. In a sudden expansion, the pressure change settles out in about eight times the diameter change (D2 – D1), whereas the velocity profile may take at least a 50% greater distance to return to fully developed pipe flow (Lipstein 1962). Instead of viewing these losses as occurring over tens or hundreds of pipe diameters, it is possible to treat the entire effect of a disturbance as if it occurs at a single point in the flow direction. By treating these losses as a local phenomenon, they can be related to the velocity by the loss coefficient K: Loss of section = K(V 2/2g) (34) Any valve Tee Chapter 22 and the Pipe Friction Manual (Hydraulic Institute 1990) have information for pipe applications. Chapter 21 gives information for airflow. The same type of fitting in pipes and ducts may yield a different loss, because flow disturbances are controlled by the detailed geometry of the fitting. The elbow of a small threaded pipe fitting differs from a bend in a circular duct. For 90° screw-fitting elbows, K is about 0.8 (Ito 1962), whereas smooth flanged elbows have a K as low as 0.2 at the optimum curvature. Table 3 lists fitting loss coefficients.These values indicate losses, but there is considerable variance. Note that a well-rounded entrance yields a rather small K of 0.05, whereas a gate valve that is only 25% open yields a K of 28.8. Expansion flows, such as from one conduit size to another or at the exit into a room or reservoir, are not included. For such occurrences, the Borda loss prediction (from impulse-momentum considerations) is appropriate: V1 – V2 V 1 A 1 2 Loss at expansion = ------------------------ = ------ 1 – ------ 2g A 2 2g 2 2 (35) Expansion losses may be significantly reduced by avoiding or delaying separation using a gradual diffuser (see Figure 10). For a diffuser of about 7° total angle, the loss is only about one-sixth of the loss predicted by Equation (35). The diffuser loss for total angles above 45 to 60° exceeds that of the sudden expansion, but is moderately influenced by the diameter ratio of the expansion. Optimum diffuser design involves numerous factors; excellent performance can be achieved in short diffusers with splitter vanes or suction. Turning vanes in miter bends produce the least disturbance and loss for elbows; with careful design, the loss coefficient can be reduced to as low as 0.1. For losses in smooth elbows, Ito (1962) found a Reynolds number effect (K slowly decreasing with increasing Re) and a minimum loss at a bend curvature (bend radius to diameter ratio) of 2.5. At this optimum curvature, a 45° turn had 63%, and a 180° turn approximately 120%, of the loss of a 90° bend. The loss does not vary linearly with the turning angle because secondary motion occurs. Note that using K presumes its independence of the Reynolds number. Some investigators have documented a variation in the loss coefficient with the Reynolds number. Assuming that K varies with Re similarly to f, it is convenient to represent fitting losses as adding to the effective length of uniform conduit. The effective length of a fitting is then Leff /D = K/fref (36) This file is licensed to John Murray (). Publication Date: 6/1/2017 Fluid Flow 3.9 Solution: Applying Equation (13) again and inserting the expression for head loss gives fL 8Q z1 – z2 = 10 m + ----- + K + 1 -------------D 2 gD 4 2 Because f depends on Q (unless flow is fully turbulent), iteration is required. The usual procedure is as follows: Licensed for single user. © 2017 ASHRAE, Inc. Fig. 14 Diagram for Example 2 where fref is an appropriate reference value of the friction factor. Deissler (1951) uses 0.028, and the air duct values in Chapter 21 are based on an fref of about 0.02. For rough conduits, appreciable errors can occur if the relative roughness does not correspond to that used when fref was fixed. It is unlikely that fitting losses involving separation are affected by pipe roughness. The effective length method for fitting loss evaluation is still useful. When a conduit contains a number of section changes or fittings, the values of K are added to the fL/D friction loss, or the Leff /D of the fittings are added to the conduit length L/D for evaluating the total loss HL. This assumes that each fitting loss is fully developed and its disturbance fully smoothed out before the next section change. Such an assumption is frequently wrong, and the total loss can be overestimated. For elbow flows, the total loss of adjacent bends may be over- or underestimated. The secondary flow pattern after an elbow is such that when one follows another, perhaps in a different plane, the secondary flow of the second elbow may reinforce or partially cancel that of the first. Moving the second elbow a few diameters can reduce the total loss (from more than twice the amount) to less than the loss from one elbow. Screens or perforated plates can be used for smoothing velocity profiles (Wile 1947) and flow spreading. Their effectiveness and loss coefficients depend on their amount of open area (Baines and Peterson 1951). Example 2. Water at 20°C flows through the piping system shown in Figure 14. Each ell has a very long radius and a loss coefficient of K = 0.31; the entrance at the tank is square-edged with K = 0.5, and the valve is a fully open globe valve with K = 10. The pipe roughness is 250 m. The density = 1000 kg/m3 and kinematic viscosity = 1.01 mm2/s. 1. Assume a value of f, usually the fully rough value for the given values of and D. 2. Use this value of f in the energy calculation and solve for Q. 2 Q= 4 gD z 1 – z 2 --------------------------------------- fL 8 ----- + K + 1 D 3. Use this value of Q to recalculate Re and get a new value of f. 4. Repeat until the new and old values of f agree to two significant figures. Iteration f Q, m/s Re f 0 1 0.0223 0.0230 0.04737 0.04699 3.98 E + 05 3.95 E + 05 0.0230 0.0230 As shown in the table, the result after two iterations is Q 0.047 m3/s = 47 L/s. If the resulting flow is in the fully rough zone and the fully rough value of f is used as first guess, only one iteration is required. c. For H = 22 m, what diameter pipe is needed to allow Q = 55 L/s? Solution: The energy equation in part (b) must now be solved for D with Q known. This is difficult because the energy equation cannot be solved for D, even with an assumed value of f. If Churchill's expression for f is stored as a function in a calculator, program, or spreadsheet with an iterative equation solver, a solution can be generated. In this case, D 0.166 m = 166 mm. Use the smallest available pipe size greater than 166 mm and adjust the valve as required to achieve the desired flow. Alternatively, (1) guess an available pipe size, and (2) calculate Re, f, and H for Q = 55 L/s. If the resulting value of H is greater than the given value of H = 22 m, a larger pipe is required. If the calculated H is less than 22 m, repeat using a smaller available pipe size. a. If pipe diameter D = 150 mm, what is the elevation H in the tank required to produce a flow of Q = 60 L/s? Control Valve Characterization for Liquids Solution: Apply Equation (13) between stations 1 and 2 in the figure. Note that p1 = p2, V1 0. Assume 1. The result is Control valves are characterized by a discharge coefficient Cd . As long as the Reynolds number is greater than 250, the orifice equation holds for liquids: z1 – z2 = H – 12 m = HL + V22 /2g From Equations (30) and (34), total pressure loss is 2 fL 8Q HL = ----- + K -------------2 D gD 4 where L = 102 m, K = 0.5 + (2 0.31) + 10 = 11.1, and V 2/2g = V 22/2g = 8Q2/(2gD4). Then, substituting into Equation (13), 8Q fL H = 12 m + 1 + ----- + K -------------D 2 gD 4 2 To calculate the friction factor, first calculate Reynolds number and relative roughness: Re = VD/v = 4Q/(Dv) = 495 150 /D = 0.0017 From the Moody diagram or Equation (32), f = 0.023. Then HL = 15.7 m and H = 27.7 m. b. For H = 22 m and D = 150 mm, what is the flow? Q = Cd Ao 2 p (37) where Ao is the area of the orifice opening and p is the pressure drop across the valve. The discharge coefficient is about 0.63 for sharp-edged configurations and 0.8 to 0.9 for chamfered or rounded configurations. Incompressible Flow in Systems Flow devices must be evaluated in terms of their interaction with other elements of the system [e.g., the action of valves in modifying flow rate and in matching the flow-producing device (pump or blower) with the system loss]. Analysis is by the general Bernoulli equation and the loss evaluations noted previously. A valve regulates or stops the flow of fluid by throttling. The change in flow is not proportional to the change in area of the valve opening. Figures 15 and 16 indicate the nonlinear action of valves in controlling flow. Figure 15 shows flow in a pipe discharging water from a tank that is controlled by a gate valve. The fitting loss coefficient K values are from Table 3; the friction factor f is 0.027. The degree of control also depends on the conduit L/D ratio. For a This file is licensed to John Murray (). Publication Date: 6/1/2017 3.10 2017 ASHRAE Handbook—Fundamentals (SI) Licensed for single user. © 2017 ASHRAE, Inc. relatively long conduit, the valve must be nearly closed before its high K value becomes a significant portion of the loss. Figure 16 shows a control damper (essentially a butterfly valve) in a duct discharging air from a plenum held at constant pressure. With a long duct, the damper does not affect the flow rate until it is about onequarter closed. Duct length has little effect when the damper is more than half closed. The damper closes the duct totally at the 90° position (K =). Flow in a system (pump or blower and conduit with fittings) involves interaction between the characteristics of the flow-producing device (pump or blower) and the loss characteristics of the pipeline or duct system. Often the devices are centrifugal, in which case the pressure produced decreases as flow increases, except for the lowest flow rates. System pressure required to overcome losses increases roughly as the square of the flow rate. The flow rate of a given system is that where the two curves of pressure versus flow rate intersect (point 1 in Figure 17). When a control valve (or damper) is partially closed, it increases losses and reduces flow (point 2 in Figure 17). For cases of constant pressure, the flow decrease caused by valving is not as great as that indicated in Figures 15 and 16. Flow Measurement The general principles noted (the continuity and Bernoulli equations) are basic to most fluid-metering devices. Chapter 36 has further details. The pressure difference between the stagnation point (total pressure) and the ambient fluid stream (static pressure) is used to give a point velocity measurement. Flow rate in a conduit is measured by placing a pitot device at various locations in the cross section and spatially integrating over the velocity found. A single-point measurement may be used for approximate flow rate evaluation. When flow is fully developed, the pipe-factor information of Figure 5 can be used to estimate the flow rate from a centerline measurement. Measurements can be made in one of two modes. With the pitotstatic tube, the ambient (static) pressure is found from pressure taps along the side of the forward-facing portion of the tube. When this portion is not long and slender, static pressure indication will be low and velocity indication high; as a result, a tube coefficient less than unity must be used. For parallel conduit flow, wall piezometers (taps) may take the ambient pressure, and the pitot tube indicates the impact (total pressure). The venturi meter, flow nozzle, and orifice meter are flow-ratemetering devices based on the pressure change associated with relatively sudden changes in conduit section area (Figure 18). The elbow meter (also shown in Figure 18) is another differential pressure flowmeter. The flow nozzle is similar to the venturi in action, but does not have the downstream diffuser. For all these, the flow rate is proportional to the square root of the pressure difference resulting from fluid flow. With area-change devices (venturi, flow nozzle, and orifice meter), a theoretical flow rate relation is found by applying the Bernoulli and continuity equations in Equations (12) and (3) between stations 1 and 2: 2 d 2gh Qtheoretical = --------- ------------4 1 – 4 (38) where h = h1 – h2 = (p1 – p2)/g and = d/D = ratio of throat (or orifice) diameter to conduit diameter. Fig. 15 Valve Action in Pipeline Fig. 17 Fig. 16 Effect of Duct Length on Damper Action Matching of Pump or Blower to System Fig. 18 Differential Pressure Flowmeters This file is licensed to John Murray (). Publication Date: 6/1/2017 Fluid Flow 3.11 Q = 0.96 0.1527 = 0.147 m3/s Unsteady Flow Conduit flows are not always steady. In a compressible fluid, acoustic velocity is usually high and conduit length is rather short, so the time of signal travel is negligibly small. Even in the incompressible approximation, system response is not instantaneous. If a pressure difference p is applied between the conduit ends, the fluid mass must be accelerated and wall friction overcome, so a finite time passes before the steady flow rate corresponding to the pressure drop is achieved. The time it takes for an incompressible fluid in a horizontal, constant-area conduit of length L to achieve steady flow may be estimated by using the unsteady flow equation of motion with wall friction effects included. On the quasi-steady assumption, friction loss is given by Equation (30); also by continuity, V is constant along the conduit. The occurrences are characterized by the relation 2 dV 1 dp f V ------- + --- ------- + ---------- = 0 d ds 2D Licensed for single user. © 2017 ASHRAE, Inc. Fig. 19 Flowmeter Coefficients The actual flow rate through the device can differ because the approach flow kinetic energy factor deviates from unity and because of small losses. More significantly, jet contraction of orifice flow is neglected in deriving Equation (38), to the extent that it can reduce the effective flow area by a factor of 0.6. The effect of all these factors can be combined into the discharge coefficient Cd: Q = Cd Qtheoretical (39) where is the time and s is the distance in flow direction. Because a certain p is applied over conduit length L, p fV 2 dV ------- = ------- – --------d L 2D (40) Take care to note the definition used by a source of Cd data. The general mode of variation in Cd for orifices and venturis is indicated in Figure 19 as a function of Reynolds number and, to a lesser extent, diameter ratio . For Reynolds numbers less than 10, the coefficient varies as Re . The elbow meter uses the pressure difference inside and outside the bend as the metering signal (Murdock et al. 1964). Momentum analysis gives the flow rate as p 32V dV = A – BV ------- = ------ – -----------L D 2 d = and Example 3. For a venturi with oil (= 800 kg/m3, = 0.01 Pa·s), find Q for P1 – P2 = 28 kPa, D = 300 mm = 0.3 m, d = 150 mm = 0.15 m. 2 d 2 P 1 – P 2 Q = Cd --------- -------------------------4 1 – 4 where = 150 mm/300 mm = 0.5. Inserting numbers, being careful to ensure that the units for Q are m3/s, gives Q = 0.1527Cd. Guessing ReD = 105 and using Figure 19 gives Cd = 0.97 and Q = 0.97 0.962 = 0.1481 m3/s. Checking ReD using this Q gives Re = 5.0 105. At this Re, Cd = 0.96 and - = – – d = --------------A – BV B dV 1 ln(A – BV) – 32 p D2 L V = ------- --------- 1 – ------ exp -------------- L 32 p D2 (45) (46) For long times (), the steady velocity is p R2 p D2 V = ------- --------- = ------ ------ L 8 L 32 (41) where R is the radius of curvature of the bend. Again, a discharge coefficient Cd is needed; as in Figure 19, this drops off for lower Reynolds numbers (below 105). These devices are calibrated in pipes with fully developed velocity profiles, so they must be located far enough downstream of sections that modify the approach velocity. (44) Equation (44) can be rearranged and integrated to yield the time to reach a certain velocity: 2 d R Qtheoretical = --------- ------- 2gh 4 2D (43) For laminar flow, f is given by Equation (31): where Q is actual flow. In some sources, instead of being defined as in Equation (39), Cd is replaced with Cd ------------------4 1– (42) (47) as given by Equation (17). Then, Equation (47) becomes – f V L V = V 1 – ------ exp ------------------ p 2D (48) 64 f = ----------V D (49) where The general nature of velocity development for start-up flow is derived by more complex techniques; however, the temporal variation is as given here. For shutdown flow (steady flow with p = 0 at > 0), flow decays exponentially as e–. Turbulent flow analysis of Equation (42) also must be based on the quasi-steady approximation, with less justification. Daily et al. (1956) indicate that frictional resistance is slightly greater than the This file is licensed to John Murray (). Publication Date: 6/1/2017 3.12 2017 ASHRAE Handbook—Fundamentals (SI) energy into internal energy), the process is reversible (isentropic) and follows the relationship p/k = constant k = cp /cv where k, the ratio of specific heats at constant pressure and volume, is 1.4 for air and diatomic gases. When the elevation term gz is neglected, as it is in most compressible flow analyses, the Bernoulli equation of steady flow, Equation (21), becomes Fig. 20 Temporal Increase in Velocity Following Sudden Application of Pressure (50) Licensed for single user. © 2017 ASHRAE, Inc. and for the accelerating flow, B –1 1 = ------------ tanh V ----- A AB (51) A B tanh AB (52) or V= Because the hyperbolic tangent is zero when the independent variable is zero and unity when the variable is infinity, the initial (V = 0 at = 0) and final conditions are verified. Thus, for long times (), V = AB = p L ----------------- = f 2D p 2D ------ ------- L f (53) which is in accord with Equation (30) when f is constant (the flow regime is the fully rough one of Figure 13). The temporal velocity variation is then V = V tanh (f V /2D) (54) In Figure 20, the turbulent velocity start-up result is compared with the laminar one, where initially the turbulent is steeper but of the same general form, increasing rapidly at the start but reaching V asymptotically. Compressibility All fluids are compressible to some degree; their density depends somewhat on the pressure. Steady liquid flow may ordinarily be treated as incompressible, and incompressible flow analysis is satisfactory for gases and vapors at velocities below about 20 to 40 m/s, except in long conduits. For liquids in pipelines, a severe pressure surge or water hammer may be produced if flow is suddenly stopped. This pressure surge travels along the pipe at the speed of sound in the liquid, alternately compressing and decompressing the liquid. For steady gas flows in long conduits, the pressure drop along the conduit can reduce gas density enough to increase the velocity. If the conduit is long enough, the velocity may reach the speed of sound, and the Mach number (ratio of flow velocity to the speed of sound) must be considered. Some compressible flows occur without heat gain or loss (adiabatically). If there is no friction (conversion of flow mechanical (55) For a frictionless adiabatic process, - ----- – ----- 1 ------ = --------k – 1 2 1 steady-state result for accelerating flows, but appreciably less for decelerating flows. If the friction factor is approximated as constant, p fV 2 dV ------- = ----- – ---------- = A – BV 2 L 2D d dp V 2 ------ + ------ = constant 2 2 dp k p2 p1 (56) Integrating between upstream station 1 and downstream station 2 gives V2 – V1 p1 k p k – 1 k ----- ---------- ----2- + ------------------ = 0 – 1 1 k – 1 p 2 1 2 2 (57) Equation (57) replaces the Bernoulli equation for compressible flows. If station 2 is the stagnation point at the front of a body, V2 = 0, and solving Equation (57) for p2 gives k – 1 1 V1 ps = p2 = p1 1 + ----------- ----------- 2 kp 1 2 k k – 1 (58) where ps is the stagnation pressure. Because the speed of sound of the gas is a = kp/ and Mach number M = V/a, the stagnation pressure in Equation (58) becomes k – 1 2 ps = p1 1 + ---------- M 1 2 k k – 1 (59) For Mach numbers less than one, 1 V1 M1 – k 4 ps = p1 ----------- 1 + ------- + 2---------- M + 2 4 24 1 2 (60) When M = 0, Equation (60) reduces to the incompressible flow result obtained from Equation (9). When the upstream Mach number exceeds 0.2, the difference is significant. Thus, a pitot tube in air is influenced by compressibility at velocities over about 66 m/s. Flow Measurement. For isentropic flow through a converging conduit such as a flow nozzle, venturi, or orifice meter, where velocity at the upstream station 1 is small, Equation (57) gives V2 = p2 k – 1 k 2k p 1 ----------- ----- 1 – ----- k – 1 1 p1 The mass flow rate is (61) This file is licensed to John Murray (). Publication Date: 6/1/2017 Fluid Flow 3.13 m· = V2 A 2 2 = A2 2k k + 1 k p2 p2 2k – ----- ----------- p 1 1 ----- k–1 p1 p1 (62) Compressible Conduit Flow For incompressible frictionless flow, the mass flow rate is m· in = A 2 2 p = A 2 2 p 1 – p 2 (63) The compressibility effect is often accounted for by the expansion factor Y : m· = Y m· in = A 2 Y 2 p 1 – p 2 (64) where = 1, and A2 is the throat cross-sectional area. Y 1, with Y = 1 for the incompressible case. For compressible flow through orifices (ISO Standard 5167), Y = 1 – (0.351 + 0.2564 + 0.938)(1 – (p2/p1)1/k) (65) Licensed for single user. © 2017 ASHRAE, Inc. where = D2/D1. For venturis and nozzles (ISO Standard 5167), k–1 ---------- p 2 k Y = 2 1 – ----- --4 k p 1 p k 1– 2 ----------- ----- ---------------------------2- ---------------------------k – 1 p 1 -- p 2 k 1 – ----- 4 p 2 1 – ----- p 1 p 1 (66) For air (k = 1.4), Y = 0.95 for an orifice with p2/p1 = 0.83 and for a venturi at about 0.90, when these devices are of relatively small diameter (D2/D1 < 0.5). As p2/p1 decreases, flow rate increases, but more slowly than for the incompressible case because of the nearly linear decrease in Y. However, if the downstream velocity reaches the local speed of sound, the mass flow rate becomes the value fixed by upstream pressure and density at the critical pressure ratio: p2 ----p1 c 2 k k – 1 = 0.53 for air = ----------- k + 1 (67) At higher pressure ratios than critical, choking (no increase in flow with decrease in downstream pressure) occurs and is used in some flow control devices to avoid flow dependence on downstream conditions. Using Equations (38) and (39) for the incompressible mass flow rate and adding the compressible expansion factor Y results in 2 d 2p m· = C d Y -------- ------------4 1 – 4 Solution. Using p2 /p1 = 80/100 = 0.8 and = 5/10 = 0.5 in Equation (66) gives Y = 0.879. In Equation (68), d = 0.05 m, p = 20 kPa, and = 1.169 kg/m3 at 25°C, 100 kPa. The result is m· = 0.383 kg/s. (68) When friction loss is included, as it must be except for very short conduits, incompressible flow analysis applies until the pressure drop exceeds about 10% of the initial pressure. The possibility of sonic velocities at the end of relatively long conduits limits the amount of pressure reduction achieved. For an inlet Mach number of 0.2, discharge pressure can be reduced to about 0.2 of the initial pressure; for inflow at M = 0.5, discharge pressure cannot be less than about 0.45p1 (adiabatic) or about 0.6p1 (isothermal). Analysis must treat density change, as evaluated from the continuity relation in Equation (3), with frictional occurrences evaluated from wall roughness and Reynolds number correlations of incompressible flow (Binder 1944). In evaluating valve and fitting losses, consider the reduction in K caused by compressibility (Benedict and Carlucci 1966). Although the analysis differs significantly, isothermal and adiabatic flows involve essentially the same pressure variation along the conduit, up to the limiting conditions. Cavitation Liquid flow with gas- or vapor-filled pockets can occur if the absolute pressure is reduced to vapor pressure or less. In this case, one or more cavities form, because liquids are rarely pure enough to withstand any tensile stressing or pressures less than vapor pressure for any length of time (John and Haberman 1980; Knapp et al. 1970; Robertson and Wislicenus 1969). Robertson and Wislicenus (1969) indicate significant occurrences in various technical fields, chiefly in hydraulic equipment and turbomachines. Initial evidence of cavitation is the collapse noise of many small bubbles that appear initially as they are carried by the flow into higher-pressure regions. The noise is not deleterious and serves as a warning of the occurrence. As flow velocity further increases or pressure decreases, the severity of cavitation increases. More bubbles appear and may join to form large fixed cavities. The space they occupy becomes large enough to modify the flow pattern and alter performance of the flow device. Collapse of cavities on or near solid boundaries becomes so frequent that, in time, the cumulative impact causes cavitational erosion of the surface or excessive vibration. As a result, pumps can lose efficiency or their parts may erode locally. Control valves may be noisy or seriously damaged by cavitation. Cavitation in orifice and valve flow is shown in Figure 21. With high upstream pressure and a low flow rate, no cavitation occurs. As pressure is reduced or flow rate increased, the minimum pressure in the flow (in the shear layer leaving the edge of the orifice) eventually approaches vapor pressure. Turbulence in this layer causes fluctuating pressures below the mean (as in vortex cores) and small bubble-like cavities. These are carried downstream into the region of pressure regain where they collapse, either in the fluid or on the wall (Figure 21A). As pressure reduces, more vapor- or gas-filled bubbles result and coalesce into larger ones. Eventually, a single large cavity results that collapses further downstream (Figure 21B). where d is throat diameter, and Cd is the discharge coefficient introduced in the Flow Measurement section. Note that Cd accounts for the effects of friction in the measuring device, and Y accounts for compressibility. Example 4. For a venturi used to measure air (k = 1.4) flow, inlet pressure and temperature are 100 kPa (absolute) and 25°C. The measured pressure at the throat is 80 kPa (absolute). The inlet diameter is 100 mm, and the throat diameter is 50 mm. What is the mass flow rate? Use Cd = 0.995. Fig. 21 Cavitation in Flows in Orifice or Valve This file is licensed to John Murray (). Publication Date: 6/1/2017 3.14 2017 ASHRAE Handbook—Fundamentals (SI) The region of wall damage is then as many as 20 diameters downstream from the valve or orifice plate. Sensitivity of a device to cavitation is measured by the cavitation index or cavitation number, which is the ratio of the available pressure above vapor pressure to the dynamic pressure of the reference flow: 2 po – pv = ------------------------2 V o (69) Licensed for single user. © 2017 ASHRAE, Inc. where pv is vapor pressure, and the subscript o refers to appropriate reference conditions. Valve analyses use such an index to determine when cavitation will affect the discharge coefficient (Ball 1957). With flow-metering devices such as orifices, venturis, and flow nozzles, there is little cavitation, because it occurs mostly downstream of the flow regions involved in establishing the metering action. The detrimental effects of cavitation can be avoided by operating the liquid-flow device at high enough pressures. When this is not possible, the flow must be changed or the device must be built to withstand cavitation effects. Some materials or surface coatings are more resistant to cavitation erosion than others, but none is immune. Surface contours can be designed to delay onset of cavitation. 5. NOISE IN FLUID FLOW Noise in flowing fluids results from unsteady flow fields and can be at discrete frequencies or broadly distributed over the audible range. With liquid flow, cavitation results in noise through the collapse of vapor bubbles. Noise in pumps or fittings (e.g., valves) can be a rattling or sharp hissing sound, which is easily eliminated by raising the system pressure. With severe cavitation, the resulting unsteady flow can produce indirect noise from induced vibration of adjacent parts. See Chapter 48 of the 2015 ASHRAE Handbook— HVAC Applications for more information on noise control. Disturbed laminar flow behind cylinders can be an oscillating motion. The shedding frequency f of these vortexes is characterized by a Strouhal number St = fd/V of about 0.21 for a circular cylinder of diameter d, over a considerable range of Reynolds numbers. This oscillating flow can be a powerful noise source, particularly when f is close to the natural frequency of the cylinder or some nearby structural member so that resonance occurs. With cylinders of another shape, such as impeller blades of a pump or blower, the characterizing Strouhal number involves the trailing-edge thickness of the member. The strength of the vortex wake, with its resulting vibrations and noise potential, can be reduced by breaking up flow with downstream splitter plates or boundary-layer trip devices (wires) on the cylinder surface. Noises produced in pipes and ducts, especially from valves and fittings, are associated with the loss through such elements. The sound pressure of noise in water pipe flow increases linearly with pressure loss; broadband noise increases, but only in the lowerfrequency range. Fitting-produced noise levels also increase with fitting loss (even without cavitation) and significantly exceed noise levels of the pipe flow. The relation between noise and loss is not surprising because both involve excessive flow perturbations. A valve's pressure-flow characteristics and structural elasticity may be such that for some operating point it oscillates, perhaps in resonance with part of the piping system, to produce excessive noise. A change in the operating point conditions or details of the valve geometry can result in significant noise reduction. Pumps and blowers are strong potential noise sources. Turbomachinery noise is associated with blade-flow occurrences. Broadband noise appears from vortex and turbulence interaction with walls and is primarily a function of the operating point of the machine. For blowers, it has a minimum at the peak efficiency point (Groff et al. 1967). Narrow-band noise also appears at the bladecrossing frequency and its harmonics. Such noise can be very annoying because it stands out from the background. To reduce this noise, increase clearances between impeller and housing, and space impeller blades unevenly around the circumference. 6. A Ao B CD Cd Dh EL EM F = = = = = = = = = f FD fref g gc HL HM K k L Le Leff m· p Pw Q q R Re s St u V v w y Y z = SYMBOLS m2 area, area of orifice opening Bernoulli constant drag coefficient discharge coefficient hydraulic diameter loss during conversion of energy from mechanical to internal external work from fluid machine tangential force per unit area required to slide one of two parallel plates Darcy-Weisbach friction factor, or shedding frequency drag force reference value of friction factor gravitational acceleration, m/s2 gravitational constant = 1 (kg·m)/(N·s2) head lost through friction head added by pump loss coefficient ratio of specific heats at constant pressure and volume length entrance length effective length mass flow rate pressure wetted perimeter volumetric flow rate heat per unit mass absorbed or rejected pipe radius Reynolds number flow direction Strouhal number internal energy velocity fluid velocity normal to differential area dA work per unit mass distance from centerline distance between two parallel plates, m, or expansion factor elevation Greek E p u = = = = = = = = = = w = = = = = kinetic energy factor d/D = ratio of throat (or orifice) diameter to conduit diameter specific mass or density boundary layer thickness change in energy content per unit mass of flowing fluid pressure drop across valve conversion of energy from mechanical to internal roughness height time proportionality factor for absolute or dynamic viscosity of fluid, (mN·s)/m2 kinematic viscosity, mm2/s density, kg/m3 cavitation index or number shear stress, Pa wall shear stress REFERENCES ASHRAE members can access ASHRAE Journal articles and ASHRAE research project final reports at technologyportal .ashrae.org. Articles and reports are also available for purchase by nonmembers in the online ASHRAE Bookstore at www.ashrae.org /bookstore. Baines, W.D., and E.G. Peterson. 1951. An investigation of flow through screens. ASME Transactions 73:467. This file is licensed to John Murray (). Publication Date: 6/1/2017 Licensed for single user. © 2017 ASHRAE, Inc. Fluid Flow 3.15 Ball, J.W. 1957. Cavitation characteristics of gate valves and globe values used as flow regulators under heads up to about 125 ft. ASME Transactions 79:1275. Benedict, R.P., and N.A. Carlucci. 1966. Handbook of specific losses in flow systems. Plenum Press Data Division, New York. Binder, R.C. 1944. Limiting isothermal flow in pipes. ASME Transactions 66:221. Churchill, S.W. 1977. Friction-factor equation spans all fluid flow regimes. Chemical Engineering 84(24):91-92. Colborne, W.G., and A.J. Drobitch. 1966. An experimental study of nonisothermal flow in a vertical circular tube. ASHRAE Transactions 72(4):5. 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Publication Date: 6/1/2017 Related Commercial Resources CHAPTER 4 HEAT TRANSFER Heat Transfer Processes .... 4.1 Thermal Conduction... 4.3 Thermal Radiation .... 4.11 Thermal Convection .... 4.17 Heat Exchangers .... 4.22 Heat Transfer Augmentation.. 4.24 Symbols .... 4.31 EAT transfer is energy transferred because of a temperature difference. Energy moves from a higher-temperature region to a lower-temperature region by one or more of three modes: conduction, radiation, and convection. This chapter presents elementary principles of single-phase heat transfer, with emphasis on HVAC applications. Boiling and condensation are discussed in Chapter 5. More specific information on heat transfer to or from buildings or refrigerated spaces can be found in Chapters 14 to 19, 23, and 27 of this volume and in Chapter 24 of the 2014 ASHRAE Handbook—Refrigeration. Physical properties of substances can be found in Chapters 26, 28, 32, and 33 of this volume and in Chapter 19 of the 2014 ASHRAE Handbook—Refrigeration. Heat transfer equipment, including evaporators, condensers, heating and cooling coils, furnaces, and radiators, is covered in the 2016 ASHRAE Handbook—HVAC Systems and Equipment. For further information on heat transfer, see the Bibliography. Licensed for single user. © 2017 ASHRAE, Inc. H 1. HEAT TRANSFER PROCESSES Conduction Consider a wall that is 10 m long, 3 m tall, and 100 mm thick (Figure 1A). One side of the wall is maintained at ts1 = 25°C, and the other is kept at ts2 = 20°C. Heat transfer occurs at rate q through the wall from the warmer side to the cooler. The heat transfer mode is conduction (the only way energy can be transferred through a solid). • If ts1 is raised from 25 to 30°C while everything else remains the same, q doubles because ts1 – ts2 doubles. • If the wall is twice as tall, thus doubling the area Ac of the wall, q doubles. • If the wall is twice as thick, q is halved. From these relationships, t s1 – t s2 A c q ----------------------------L where means "proportional to" and L = wall thickness. However, this relation does not take wall material into account; if the wall were foam instead of concrete, q would clearly be less. The constant of proportionality is a material property, thermal conductivity k. Thus, t s1 – t s2 A c t s1 – t s2 q = k ----------------------------- = ---------------------L L kA c where k has units of W/(m·K). The denominator L/(kAc) can be considered the conduction resistance associated with the driving potential (ts1 – ts2). This is analogous to current flow through an electrical resistance, I = (V1 – V2)/R, where (V1 – V2) is driving potential, R is electrical resistance, and current I is rate of flow of charge instead of rate of heat transfer q. Thermal resistance has units K/W. A wall with a resistance of 5 K/W requires (ts1 – ts2) = 5 K for heat transfer q of 1 W. The thermal/electrical resistance analogy allows tools used to solve electrical circuits to be used for heat transfer problems. Convection Consider a surface at temperature ts in contact with a fluid at t (Figure 1B). Newton's law of cooling expresses the rate of heat transfer from the surface of area As as ts – t q = hcAs(ts – t) = ---------------------1 hc As Table 1 The preparation of this chapter is assigned to TC 1.3, Heat Transfer and Fluid Flow. 4.1 Copyright © 2017, ASHRAE (2) where hc is the heat transfer coefficient (Table 1) and has units of W/(m2 ·K). The convection resistance 1/(hc As) has units of K/W. If t > ts, heat transfers from the fluid to the surface, and q is written as just q = hc As(t – ts). Resistance is the same, but the sign of the temperature difference is reversed. For heat transfer to be considered convection, fluid in contact with the surface must be in motion; if not, the mode of heat transfer is conduction. If fluid motion is caused by an external force (e.g., fan, pump, wind), it is forced convection. If fluid motion results from buoyant forces caused by the surface being warmer or cooler than the fluid, it is free (or natural) convection. Heat Transfer Coefficients by Convection Type Convection Type Fig. 1 (A) Conduction and (B) Convection (1) Free, gases Free, liquids Forced, gases Forced, liquids Boiling, condensation hc , W/(m2 ·K) 2 to 25 10 to 1000 25 to 250 50 to 20 000 2500 to 100 000 This file is licensed to John Murray (). Publication Date: 6/1/2017 4.2 2017 ASHRAE Handbook—Fundamentals (SI) Radiation Matter emits thermal radiation at its surface when its temperature is above absolute zero. This radiation is in the form of photons of varying frequency. These photons leaving the surface need no medium to transport them, unlike conduction and convection (in which heat transfer occurs through matter). The rate of thermal radiant energy emitted by a surface depends on its absolute temperature and its surface characteristics. A surface that absorbs all radiation incident upon it is called a black surface, and emits energy at the maximum possible rate at a given temperature. The heat emission from a black surface is given by the Stefan-Boltzmann law: qemitted, black = AsWb = AsTs4 where Wb = Ts4 is the blackbody emissive power in W/m2; Ts is absolute surface temperature, K; and = 5.67 10–8 W/(m2 ·K4) is the Stefan-Boltzmann constant. If a surface is not black, the emission per unit time per unit area is Licensed for single user. © 2017 ASHRAE, Inc. W = Wb = Ts4 where W is emissive power, and is emissivity, where 0 1. For a black surface, = 1. Nonblack surfaces do not absorb all incident radiation. The absorbed radiation is qabsorbed = AsG where absorptivity is the fraction of incident radiation absorbed, and irradiation G is the rate of radiant energy incident on a surface per unit area of the receiving surface. For a black surface, = 1. A surface's emissivity and absorptivity are often both functions of the wavelength distribution of photons emitted and absorbed, respectively, by the surface. However, in many cases, it is reasonable to assume that both and are independent of wavelength. If so, = (a gray surface). Two surfaces at different temperatures that can "see" each other can exchange energy through radiation. The net exchange rate depends on the surfaces' (1) relative size, (2) relative orientation and shape, (3) temperatures, and (4) emissivity and absorptivity. However, for a small area As in a large enclosure at constant temperature tsurr , the irradiation on As from the surroundings is the blackbody emissive power of the surroundings Wb,surr. So, if ts > tsurr , net heat loss from gray surface As in the radiation exchange with the surroundings at Tsurr is qnet = qemitted – qabsorbed = As Wbs – As Wb,surr = As(Ts4 – T 4surr) (3) where = for the gray surface. If ts < tsurr , the expression for qnet is the same with the sign reversed, and qnet is the net gain by As. Note that qnet can be written as 2)(T + T where hr = (Ts2 + T surr s surr) is often called a radiation heat transfer coefficient. The disadvantage of this form is that hr depends on ts, which is often the desired result of the calculation. Combined Radiation and Convection When tsurr = t in Equation (4), the total heat transfer from a surface by convection and radiation combined is then q = qrad + qconv = (ts – t)As (hr + hc) The temperature difference ts – t is in either K or °C; the difference is the same. Either can be used; however, absolute temperatures must be used to calculate hr . (Absolute temperatures are K = °C + 273.15.) Note that hc and hr are always positive, and that the direction of q is determined by the sign of (ts – t). Contact or Interface Resistance Heat flow through two layers encounters two conduction resistances L1/k1A and L 2/k 2 A (Figure 2). At the interface between two layers are gaps across which heat is transferred by a combination of conduction at contact points and convection and radiation across gaps. This multimode heat transfer process is usually characterized using a contact resistance coefficient R cont or contact conductance hcont. t q = --------------------- = hcont At Rcont A where t is the temperature drop across the interface. R cont is in (m2 ·K)/W, and hcont is in W/(m2 ·K). The contact or interface resistance is Rcont = R cont /A = 1/hcont A, and the resistance of the two layers combined is the sum of the resistances of the two layers and the contact resistance. Contact resistance can be reduced by lowering surface roughnesses, increasing contact pressure, or using a conductive grease or paste to fill the gaps. Heat Flux The conduction heat transfer can be written as k t s1 – t s2 q q = ----- = ------------------------L Ac where q is heat flux in W/m2. Similarly, for convection the heat flux is q q = ----- = hc(ts – t) As and net heat flux from radiation at the surface is q net 4 4 qnet = --------- = T s – T surr As W bs – W b, surr qnet = ---------------------------------1 A s In this form, Ebs – Eb,surr is analogous to the driving potential in an electric circuit, and 1/(As) is analogous to electrical resistance. This is a convenient analogy when only radiation is being considered, but if convection and radiation both occur at a surface, convection is described by a driving potential based on the difference in the first power of the temperatures, whereas radiation is described by the difference in the fourth power of the temperatures. In cases like this, it is often useful to express net radiation as qnet = hr As(ts – tsurr) = (ts – tsurr)/(1/hr As) (4) Fig. 2 Interface Resistance Across Two Layers This file is licensed to John Murray (). Publication Date: 6/1/2017 Heat Transfer 4.3 Overall Resistance and Heat Transfer Coefficient In Equation (1) for conduction in a slab, Equation (4) for radiative heat transfer rate between two surfaces, and Equation (2) for convective heat transfer rate from a surface, the heat transfer rate is expressed as a temperature difference divided by a thermal resistance. Using the electrical resistance analogy, with temperature difference and heat transfer rate instead of potential difference and current, respectively, tools for solving series electrical resistance circuits can also be applied to heat transfer circuits. For example, consider the heat transfer rate from a liquid to the surrounding gas separated by a constant cross-sectional area solid, as shown in Figure 3. The heat transfer rate from the liquid to the adjacent surface is by convection, then across the solid body by conduction, and finally from the solid surface to the surroundings by both convection and radiation. A circuit using the equations for resistances in each mode is also shown. From the circuit, the heat transfer rate is 2. One-Dimensional Steady-State Conduction Steady-state heat transfer rates and resistances for (1) a slab of constant cross-sectional area, (2) a hollow cylinder with radial heat transfer, and (3) a hollow sphere are given in Table 2. Example 1. Chilled water at 5°C flows in a copper pipe with a thermal conductivity kp of 400 W/(mK), with internal and external diameters of ID = 100 mm and OD = 120 mm. (Figure 4) The tube is covered with insulation 50 mm thick, with ki = 0.20 W/(m·K). The surrounding air is at ta = 25°C, and the heat transfer coefficient at the outer surface ho = 10 W/(m2 ·K). Emissivity of the outer surface is = 0.85. The heat transfer coefficient inside the tube is hi = 1000 W/(m2 ·K). Contact resistance between the insulation and the pipe is assumed to be negligible. Find the rate of heat gain for a given length of pipe and the temperature at the pipe-insulation interface. Solution: The outer diameter of the insulation is Dins = 120 + 2(50) = 220 mm. From Table 2, for L = 1 m, tf 1 – tf 2 q = ------------------------------R1 + R2 + R3 1 R1 = ------------------- = 3.2 10–3 K/W h i IDL Licensed for single user. © 2017 ASHRAE, Inc. where R1 = 1/hA R2 = L/kA ln OD ID R2 = ---------------------------- = 7 10–5 K/W 2k p L 1 hc A 1 hr A R2 = -- 1 hc A + 1 hr A Resistance R3 is the parallel combination of the convection and radiation resistances on the right-hand surface, 1/hc A and 1/hr A. Equivalently, R3 = 1/hrc A, where hrc on the air side is the sum of the convection and radiation heat transfer coefficients (i.e., hrc = hc + hr). The heat transfer rate can also be written as q = UA(tf 1 – tf 2) where U is the overall heat transfer coefficient that accounts for all the resistances involved. Note that tf 1 – tf 2 1 ------------------ = -------- = R1 + R2 +R3 q UA The product UA is overall conductance, the reciprocal of overall resistance. The surface area A on which U is based is not always constant as in this example, and should always be specified when referring to U. Heat transfer rates are equal from the warm liquid to the solid surface, through the solid, and then to the cool gas. Temperature drops across each part of the heat flow path are related to the resistances (as voltage drops are in an electric circuit), so that tf1 – t1 = qR1 t1 – t2 = qR2 THERMAL CONDUCTION t2 – tf2 = qR3 Fig. 3 Thermal Circuit ln D ins OD R3 = ---------------------------------- = 0.482 K/W 2k p L 1 Rc = ------------------------- = 0.144 K/W h o D ins L Assuming insulation surface temperature ts = 21°C (i.e., 294 K) and 2)(T + T 2 Tsurr = Ta = 298.15 K, hr = (Ts2 + T surr s surr) = 5.0 W/(m ·K). Table 2 One-Dimensional Conduction Shape Factors Configuration Constant crosssectional area slab Heat Transfer Rate Thermal Resistance t1 – t2 qx = kAx -------------L L --------kA x Hollow cylinder of length L with negligible heat transfer from end surfaces 2kL t i – t o q r = ----------------------------------ro ln ---- ri ln r o r i R = ------------------------2kL Hollow sphere 4k t i – t o q r = -------------------------1 1 --- + ---ri ro 1 ri – 1 ro R = -------------------------4k This file is licensed to John Murray (). Publication Date: 6/1/2017 4.4 2017 ASHRAE Handbook—Fundamentals (SI) Two- and Three-Dimensional Steady-State Conduction: Shape Factors 1 Rr = ---------------------- = 0.288 K/W h r D ins L Rr Rc R4 = ---------------- = 0.096 K/W Rr + Rc Rtot = R1 + R2 + R3 + R4 = 0.581 KW Finally, the rate of heat gain by the cold water is ta – t qrc = ----------- = 34.4 W R tot Mathematical solutions to a number of two and three-dimensional conduction problems are available in Carslaw and Jaeger (1959). Complex problems can also often be solved by graphical or numerical methods, as described by Adams and Rogers (1973), Croft and Lilley (1977), and Patankar (1980).

There are many two- and threedimensional steady-state cases that can be solved using conduction shape factors. Using the conduction shape factor S, the heat transfer rate is expressed as Temperature at the pipe/insulation interface is ts2 = t + qrc (R1 + R2) = 5.1°C Temperature at the insulation's surface is ts3 = ta – qrc R4 = 21.7°C Licensed for single user. © 2017 ASHRAE, Inc. which is very close to the assumed value of 22°C. Note the importance of the pipe/insulation interface temperature. If this temperature is below the dew point, condensation will occur that will damage the insulation. A more complete version of this kind of example is given in Chapter 6 It covers calculation of interface temperatures and the vapor pressures needed to find the corresponding dew-point temperatures at the interface, as well as methods to prevent condensation. q = Sk(t1 – t2) = (t1 – t2)/(1/Sk) (5) where k is the material's thermal conductivity, t1 and t2 are temperatures of two surfaces, and 1/(Sk) is thermal resistance. Conduction shape factors for some common configurations are given in Table 3. Example 2. The walls and roof of a house are made of 200 mm thick concrete with k = 0.75 W/(m·K). The inner surface is at 20°C, and the outer surface is at 8°C. The roof is 10 × 10 m, and the walls are 6 m high. Find the rate of heat loss from the house through its walls and roof, including edge and corner effects. Solution: The rate of heat transfer excluding the edges and corners is first determined: Atotal = (10 – 0.4)(10 – 0.4) + 4(10 – 0.4)(6 – 0.2) = 314.9 m2 2 kA total 0.75 W/(m·K) 314.9 m qwalls+ceiling = --------------- T = ---0.2 m L = (20 – 8)°C = 14 170 W The shape factors for the corners and edges are in Table 2: Scorners+edges = 4 Scorner + 4 Sedge = 4 0.15L + 4 0.54W = 4 0.15(0.2 m) + 4 0.54(9.6 m) = 20.86 m and the heat transfer rate is qcorners+edges = Scorners+edges kT = (20.86 m)[0.75 W/(m·K)](20 – 8)°C = 188 W which leads to Fig. 4 Thermal Circuit Diagram for Insulated Water Pipe (Example 1) Fig. 5 qtotal = 14 170 W + 188 W = 14 358 W = 14.4 kW Note that the edges and corners are 1.3% of the total. Efficiency of Annular Fins of Constant Thickness This file is licensed to John Murray (). Publication Date: 6/1/2017 Heat Transfer 4.5 Table 3 Configuration Edge of two adjoining walls Corner of three adjoining walls (inner surface at T1 and outer surface at T2) Licensed for single user. © 2017 ASHRAE, Inc. Isothermal rectangular block embedded in semiinfinite body with one face of block parallel to surface of body Thin isothermal rectangular plate buried in semiinfinite medium Multidimensional Conduction Shape Factors Shape Factor S, m 0.54W Restriction W > L/5 0.15L L W 2.756L H ---------------------------------------L >> d, W, H ---- 0.59 d d ln 1 + ----- W W ------------------------ln 4W L d = 0, W > L 2W ------------------------ln 4W L d >> W W>L 2W -------------------------ln 2d L d > 2W W >> L L >> W W > 2R Cylinder centered inside square of length L 2L -----------------------------------ln 0.54W R Isothermal cylinder buried in semi-infinite medium 2L -------------------------------1 cosh d R L >> R 2L ----------------------ln 2d R L >> R d > 3R 2L ---ln L 2d L ln --- 1 – ---------------------ln L R R d >> R L >> d 2L ----------------4d ln ----- R L >> d Horizontal cylinder of length L midway between two infinite, parallel, isothermal surfaces Isothermal sphere in semi-infinite medium Isothermal sphere in infinite medium 4R ----------------------------1 – R 2d 4R This file is licensed to John Murray (). Publication Date: 6/1/2017 4.6 2017 ASHRAE Handbook—Fundamentals (SI) Licensed for single user. © 2017 ASHRAE, Inc. Fig. 6 Efficiency of Annular Fins with Constant Metal Area for Heat Flow Fig. 7 Efficiency of Several Types of Straight Fins Extended Surfaces Heat transfer from a surface can be increased by attaching fins or extended surfaces to increase the area available for heat transfer. A few common fin geometries are shown in Figures 5 to 8. Fins provide a large surface area in a low volume, thus lowering material costs for a given performance. To achieve optimum design, fins are generally located on the side of the heat exchanger with lower heat transfer coefficients (e.g., the air side of an air-to-water coil). Equipment with extended surfaces includes natural- and forcedconvection coils and shell-and-tube evaporators and condensers. Fins are also used inside tubes in condensers and dry expansion evaporators. Fin Efficiency. As heat flows from the root of a fin to its tip, temperature drops because of the fin material's thermal resistance. The temperature difference between the fin and surrounding fluid is therefore greater at the root than at the tip, causing a corresponding variation in heat flux. Therefore, increases in fin length result in proportionately less additional heat transfer. To account for this effect, fin efficiency is defined as the ratio of the actual heat transferred Fig. 8 Efficiency of Four Types of Spines from the fin to the heat that would be transferred if the entire fin were at its root or base temperature: q = --------------------------hA s t r – t e (6) where q is heat transfer rate into/out of the fin's root, te is temperature of the surrounding environment, tr is temperature at fin root, and As is surface area of the fin. Fin efficiency is low for long or thin fins, or fins made of low-thermal-conductivity material. Fin efficiency decreases as the heat transfer coefficient increases because of increased heat flow. For natural convection in air-cooled condensers and evaporators, where the air-side h is low, fins can be fairly large and fabricated from low-conductivity materials such as steel instead of from copper or aluminum. For condensing and boiling, where large heat transfer coefficients are involved, fins must be very short for optimum use of material. Fin efficiencies for a few geometries are shown in Figures 5 to 8. Temperature distribution and fin efficiencies for various fin shapes are derived in most heat transfer texts. This file is licensed to John Murray (). Publication Date: 6/1/2017 Heat Transfer 4.7 Constant-Area Fins and Spines. For fins or spines with constant cross-sectional area [e.g., straight fins (option A in Figure 7), cylindrical spines (option D in Figure 8)], the efficiency can be calculated as tanh mWc = --------------------------mWc (7) = = = = = hP kA c fin perimeter fin cross-sectional area corrected fin/spine length = W + Ac /P d/4 for a cylindrical spine with diameter d = a/4 for an a × a square spine = yb = /2 for a straight fin with thickness A p + A s s = -------------------A Licensed for single user. © 2017 ASHRAE, Inc. Empirical Expressions for Fins on Tubes. Schmidt (1949) presents approximate, but reasonably accurate, analytical expressions (for computer use) for the fin efficiency of circular, rectangular, and hexagonal arrays of fins on round tubes, as shown in Figures 5, 9, and 10, respectively. Rectangular fin arrays are used for an in-line tube arrangement in finned-tube heat exchangers, and hexagonal arrays are used for staggered tubes. Schmidt's empirical solution is given by tanh mr b Z = ----------------------------mr b Z where rb is tube radius, m = given by (8) 2h k , = fin thickness, and Z is Z = [(re /rb) – 1][1 + 0.35 ln(re /rb)] where re is the actual or equivalent fin tip radius. For circular fins, re /rb is the actual ratio of fin tip radius to tube radius. For rectangular fins (Figure 9), re /rb = 1.28 – 0.2 = M/rb q = (hp Ap + hs As)(tr – te) (9) Assuming the heat transfer coefficients for the fin and prime surfaces are equal, a surface efficiency s can be derived as where m P Ac Wc Ac /P Other sources of information on finned surfaces are listed in the References and Bibliography. Surface Efficiency. Heat transfer from a finned surface (e.g., a tube) that includes both fin area As and unfinned or prime area Ap is given by = L/M 1 where M and L are defined by Figure 9 as a/2 or b/2, depending on which is greater. For hexagonal fins (Figure 10), where A = As + Ap is the total surface area, the sum of the fin and prime areas. The heat transfer in Equation (8) can then be written as tr – te q = s hA(tr – te) = ----------------------1 s hA where and are defined as previously, and M and L are defined by Figure 10 as a/2 or b (whichever is less) and 0.5 a 2 2 + b2 , respectively. For constant-thickness square fins on a round tube (L = M in Figure 9), the efficiency of a constant-thickness annular fin of the same area can be used. For more accuracy, particularly with rectangular fins of large aspect ratio, divide the fin into circular sectors as described by Rich (1966). Fig. 9 Rectangular Tube Array (11) where 1/(s hA) is the finned surface resistance. Example 3. An aluminum tube with k = 186 W/(m·K), ID = 45 mm, and OD = 50 mm has circular aluminum fins = 1 mm thick with an outer diameter of Dfin = 100 mm. There are N ' = 250 fins per metre of tube length. Steam condenses inside the tube at ti = 200°C with a large heat transfer coefficient on the inner tube surface. Air at t = 25°C is heated by the steam. The heat transfer coefficient outside the tube is 40 W/(m2 ·K). Find the rate of heat transfer per metre of tube length. Solution: From Figure 5's efficiency curve, the efficiency of these circular fins is = 0.89 2 40 W/(m ·K) h W ---------------- = 0.025 -- = 0.52 186 W m·K 0.0005 m k 2 W = D fin – OD 2 = 0.10 – 0.05 2 = 0.025 m X e X b = 0.10 0.05 = 2.0 The fin area for L = 1 m is 2 re /rb = 1.27 – 0.3 (10) As = 250 × 2(Dfin – OD2)/4 = 2.945 m2 The unfinned area for L = 1 m is Ap = × OD × L(1 – N) = (0.05 m)(1 m)(1 – 250 × 0.001) = 0.118 m2 and the total area A = As + Ap = 3.063 m2. Surface efficiency is Fig. 10 Hexagonal Tube Array This file is licensed to John Murray (). Publication Date: 6/1/2017 4.8 2017 ASHRAE Handbook—Fundamentals (SI) A f + A s s = ------------------- = 0.894 A and resistance of the finned surface is 1 Rs = ------------ = 9.13 10–3 K/W s hA the mass. Examples include a well-stirred fluid in a thin-walled container, or a thin metal plate with high thermal conductivity. In both cases, if the mass is heated or cooled at its surface, the temperature can be assumed to be a function of time only and not location within the body. Such an approximation is valid if h V As Bi = -------------------- 0.1 k Tube wall resistance is ln OD ID ln 5 4.5 R wall = ---------------------------- = --2Lk tube 2 1 m 186 W/(m·K) –5 = 9.02 10 K/W The rate of heat transfer is then ti – t q = ------------------------ = 18 981 W R s + R wall where Bi h V As k = = = = = The temperature is given by dt Mcp ----- = qnet + qgen d Had Schmidt's approach been used for fin efficiency, m= 2h k = 20.74 m–1 rb = OD/2 = 0.025 m Licensed for single user. © 2017 ASHRAE, Inc. Z = [(Dfin/OD) – 1][1 + 0.35 ln(Dfin /OD)] = 1.243 tanh mr b Z = ---------------------------- = 0.88 mr b Z the same as given by Figure 5. Contact Resistance. Fins can be extruded from the prime surface (e.g., short fins on tubes in flooded evaporators or water-cooled condensers) or can be fabricated separately, sometimes of a different material, and bonded to the prime surface. Metallurgical bonds are achieved by furnace-brazing, dip-brazing, or soldering; nonmetallic bonding materials, such as epoxy resin, are also used. Mechanical bonds are obtained by tension-winding fins around tubes (spiral fins) or expanding the tubes into the fins (plate fins). Metallurgical bonding, properly done, leaves negligible thermal resistance at the joint but is not always economical. Contact resistance of a mechanical bond may or may not be negligible, depending on the application, quality of manufacture, materials, and temperatures involved. Tests of plate-fin coils with expanded tubes indicate that substantial losses in performance can occur with fins that have cracked collars, but negligible contact resistance was found in coils with continuous collars and properly expanded tubes (Dart 1959). Contact resistance at an interface between two solids is largely a function of the surface properties and characteristics of the solids, contact pressure, and fluid in the interface, if any. Eckels (1977) modeled the influence of fin density, fin thickness, and tube diameter on contact pressure and compared it to data for wet and dry coils. Shlykov (1964) showed that the range of attainable contact resistances is large. Sonokama (1964) presented data on the effects of contact pressure, surface roughness, hardness, void material, and the pressure of the gas in the voids. Lewis and Sauer (1965) showed the resistance of adhesive bonds, and Clausing (1964) and Kaspareck (1964) gave data on the contact resistance in a vacuum environment. Transient Conduction Often, heat transfer and temperature distribution under transient (i.e., varying with time) conditions must be known. Examples are (1) cold-storage temperature variations on starting or stopping a refrigeration unit, (2) variation of external air temperature and solar irradiation affecting the heat load of a cold-storage room or wall temperatures, (3) time required to freeze a given material under certain conditions in a storage room, (4) quick-freezing objects by direct immersion in brines, and (5) sudden heating or cooling of fluids and solids from one temperature to another. Lumped Mass Analysis. Often, the temperature within a mass of material can be assumed to vary with time but be uniform within Biot number surface heat transfer coefficient material's volume surface area exposed to convective and/or radiative heat transfer material's thermal conductivity (12) where M = body mass cp = specific heat qgen = internal heat generation qnet = net heat transfer rate to substance (into substance is positive, and out of substance is negative) Equation (12) applies to liquids and solids. If the material is a gas being heated or cooled at constant volume, replace cp with the constant-volume specific heat cv . The term qnet may include heat transfer by conduction, convection, or radiation and is the difference between the heat transfer rates into and out of the body. The term qgen may include a chemical reaction (e.g., curing concrete) or heat generation from a current passing through a metal. For a lumped mass M initially at a uniform temperature t0 that is suddenly exposed to an environment at a different temperature t, the time taken for the temperature of the mass to change to tf is given by the solution of Equation (12) with qgen = 0 as tf – t hA s ln ------ --------- = – -----------t0 – t Mc p (13) where M cp As h tf t0 t = = = = = = = = mass of solid specific heat of solid surface area of solid surface heat transfer coefficient time required for temperature change final solid temperature initial uniform solid temperature surrounding fluid temperature Example 4. A copper sphere with diameter d = 1 mm is to be used as a sensing element for a thermostat. It is initially at a uniform temperature of t0 = 21°C. It is then exposed to the surrounding air at t = 20°C. The combined heat transfer coefficient is h = 60 W/(m2 ·K). Determine the time taken for the temperature of the sensing element to reach tf = 20.5°C. The properties of copper are = 8933 kg/m3 cp = 385 J/(kg·K) k = 401 W/(m·K) Solution: Bi = h(d/2)/k = 60.35(0.001/2)/401 = 7.5 10–5, which is much less than 1. Therefore, lumped analysis is valid. M = [4(d/2)3/3] = 4.677 × 10–6 kg As = d 2 = 3.142 × 10–6 m2 Using Equation (13), = 6.6 s. Nonlumped Analysis. When the Biot number is greater than 0.1, variation of temperature with location within the mass is significant. This file is licensed to John Murray (). Publication Date: 6/1/2017 Heat Transfer 4.9 One example is the cooling time of meats in a refrigerated space: the meat's size and conductivity do not allow it to be treated as a lumped mass that cools uniformly. Nonlumped problems require solving multidimensional partial differential equations. Many common cases have been solved and presented in graphical forms (Jakob 1949, 1957; Myers 1971; Schneider 1964). In other cases, numerical methods (Croft and Lilley 1977; Patankar 1980) must be used. Estimating Cooling Times for One-Dimensional Geometries. When a slab of thickness 2L or a solid cylinder or solid sphere with outer radius rm is initially at a uniform temperature t1, and its surface is suddenly heated or cooled by convection with a fluid at t, a mathematical solution is available for the temperature t as a function of location and time . The solution is an infinite series. However, after a short time, the temperature is very well approximated by the first term of the series. The single-term approximations for the three cases are of the form Y = Y0 f (1n) Licensed for single user. © 2017 ASHRAE, Inc. where (14) = temperature at center of slab, cylinder, or sphere 2 = /L c = Fourier number = thermal diffusivity of solid = k/cp = L for slab, ro for cylinder, sphere = x/L for slab, r/rm for cylinder = coefficients that are functions of Bi = Biot number = hLc /k = function of 1n, different for each geometry = distance from midplane of slab of thickness 2L cooled on both sides = density of solid cp = constant pressure specific heat of solid k = thermal conductivity of solid The single term solution is valid for Fo > 0.2. Values of c1 and 1 are given in Table 4 for a few values of Bi, and Couvillion (2004) provides a procedure for calculating them. Expressions for c1 for each case, along with the function f (1n), are as follows: Slab 4 sin 1 c1 = -----------------------------------2 1 + sin 2 1 (15) Long solid cylinder J1 1 2 c1 = ------ --1 J 2 + J 2 0 1 1 (16) 1 Table 4 Values of c1 and 1 in Equations (14) to (17) Slab Solid Cylinder sin 1 n f (1n) = --------------------1 n 4 sin 1 – 1 cos 1 c 1 = ---2 1 – sin 2 1 (17) These solutions are presented graphically (McAdams 1954) by Gurnie-Lurie charts (Figures 11 to 13). The charts are also valid for Fo < 0.2. Example 5. Apples, approximated as 60 mm diameter solid spheres and initially at 30°C, are loaded into a chamber maintained at 0°C. If the surface heat transfer coefficient h = 14 W/(m2 ·K), estimate the time required for the center temperature to reach t = 1°C. Properties of apples are = 830 kg/m3 k = 0.42 W/(m2 ·K) rm = d/2 = 30 mm = 0.03 m Solution: Assuming that it will take a long time for the center temperature to reach 1°C, use the one-term approximation Equation (14). From the values given, t0 Fo Lc n c1, 1 Bi f (1n) x f (1n) = J0(1n) Solid sphere cp = 3600 J/(kg·K) t – t Y = --------------t1 – t t0 – t 2 Y0 = --------------- = c 1 exp – 1 Fo t1 – t f (1n) = cos(1n) where J0 is the Bessel function of the first kind, order zero. It is available in math tables, spreadsheets, and software packages. J0(0) = 1. Solid Sphere Bi c1 1 c1 1 c1 1 0.5 1.0 2.0 4.0 6.0 8.0 10.0 30.0 50.0 1.0701 1.1191 1.1785 1.2287 1.2479 1.2570 1.2620 1.2717 1.2727 0.6533 0.8603 1.0769 1.2646 1.3496 1.3978 1.4289 1.5202 1.5400 1.1143 1.2071 1.3384 1.4698 1.5253 1.5526 1.5677 1.5973 1.6002 0.9408 1.2558 1.5995 1.9081 2.0490 2.1286 2.1795 2.3261 2.3572 1.1441 1.2732 1.4793 1.7202 1.8338 1.8920 1.9249 1.9898 1.9962 1.1656 1.5708 2.0288 2.4556 2.6537 2.7654 2.8363 3.0372 3.0788 t – t 0–1 1 Y = --------------- = --------------- = -----t – t1 0 – 30 30 r 0 n = ----- = ---------- = 0 rm 0.03 hr m 14 0.03 Bi = --------- = ----------------------- = 1 k 0.42 k 0.42 = -------- = -------------------------- = 1.406 10–7 m2/s c p 830 3600 From Equations (14) and (17) with lim(sin 0/0) = 1, Y = Y0 = c1 exp(–21Fo). For Bi = 1, from Table 4, c1= 1.2732 and 1 = 1.5708. Thus, 0.00545 1 1 Y = ------------------------------2 Fo = – -----2 ln ----- = – -------------------2 ln 0.0333 = 1.476 = -----2 c 1 0.1967 2 rm 1.5708 1 = 2.62 h Note that Fo = 0.2 corresponds to an actual time of 1280 s. Multidimensional Cooling Times. One-dimensional transient temperature solutions can be used to find the temperatures with twoand three-dimensional temperatures of solids. For example, consider a solid cylinder of length 2L and radius rm exposed to a fluid at tc on all sides with constant surface heat transfer coefficients h1 on the end surfaces and h2 on the cylindrical surface, as shown in Figure 14. The two-dimensional, dimensionless temperature Y(x1,r1,) can be expressed as the product of two one-dimensional temperatures Y1(x1,) Y2(r1,), where Y1 = dimensionless temperature of constant cross-sectional area slab at (x1,), with surface heat transfer coefficient h1 associated with two parallel surfaces Y2 = dimensionless temperature of solid cylinder at (r1,) with surface heat transfer coefficient h2 associated with cylindrical surface From Figures 11 and 12 or Equations (14) to (16), determine Y1 at (x1/L, /L 2, h1L/k) and Y2 at (r1/rm, /r 2m, h2rm /k). Example 6. A 70 mm diameter by 125 mm high soda can, initially at t1 = 30°C, is cooled in a chamber where the air is at t = 0°C. The heat transfer coefficient on all surfaces is h = 20 W/(m2 ·K). Determine the maximum temperature in the can = 1 h after starting the cooling. Assume the properties of the soda are those of water, and that the soda inside the can behaves as a solid body. This file is licensed to John Murray (). Publication Date: 6/1/2017 Licensed for single user. © 2017 ASHRAE, Inc. 4.10 2017 ASHRAE Handbook—Fundamentals (SI) Fig. 11 Transient Temperatures for Infinite Slab, m = 1/Bi Fig. 12 Transient Temperatures for Infinite Cylinder, m = 1/Bi This file is licensed to John Murray (). Publication Date: 6/1/2017 4.11 Licensed for single user. © 2017 ASHRAE, Inc. Heat Transfer Fig. 13 Transient Temperatures for Sphere, m = 1/Bi Bicyl = hrm /k = 20 0.035/0.5894 = 1.188 Focyl = /rm2 = (1.41 10–7) 3600/0.0352 = 0.4144 Focyl > 0.2, so use the one-term approximation with Equations (14) and (16). Ycyl = c1 exp(–21Focyl)J0(0) Interpolating in Table 4 for Bicyl = 1.188, cyl = 1.3042, J0(0) = 1, ccyl = 1.237, Ycyl = 0.572. 2. Determine Ypl at n = 0. Fig. 14 Solid Cylinder Exposed to Fluid Solution: Because the cylinder is short, the temperature of the soda is affected by the heat transfer rate from the cylindrical surface and end surfaces. The slowest change in temperature, and therefore the maximum temperature, is at the center of the cylinder. Denoting the dimensionless temperature by Y, Y = Ycyl Ypl where Ycyl is the dimensionless temperature of an infinitely long 70 mm diameter cylinder, and Ypl is the dimensionless temperature of a 125 mm thick slab. Each of them is found from the appropriate Biot and Fourier number. For evaluating the properties of water, choose a temperature of 15°C and a pressure of 101.35 kPa. The properties of water are = 999.1 kg/m3 k = 0.5894 W/(m·K) = k/ = 1.41 × 10–7 m2/s 1. Determine Ycyl at n = 0. cp = 4184 J/(kg·K) = 3600 s Bipl = hL/k = 20 0.0625/0.5894 = 2.121 Fopl = 1.41 × 10–7 × 3600/0.06252 = 0.1299 Fopl < 0.2, so the one-term approximation is not valid. Using Figure 11, Ypl = 0.9705. Thus, Y = 0.572 0.9705 = 0.5551 = (t – t)/(t1 – t) t = 16.7°C Note: The solution may not be exact because convective motion of the soda during heat transfer has been neglected. The example illustrates the use of the technique. For well-stirred soda, with uniform temperature within the can, the lumped mass solution should be used. 3. THERMAL RADIATION Radiation, unlike conduction and convection, does not need a solid or fluid to transport energy from a high-temperature surface to a lower-temperature one. (Radiation is in fact impeded by such a material.) The rate of radiant energy emission and its characteristics from a surface depend on the underlying material's nature, microscopic arrangement, and absolute temperature. The rate of emission from a surface is independent of the surfaces surrounding it, but the This file is licensed to John Murray (). Publication Date: 6/1/2017 4.12 2017 ASHRAE Handbook—Fundamentals (SI) rate and characteristics of radiation incident on a surface do depend on the temperatures and spatial relationships of the surrounding surfaces. Blackbody Radiation W= Wb = T 4 (18) where = 5.670 108 W/(m2 ·K4) is the Stefan-Boltzmann constant. Energy is emitted in the form of photons or electromagnetic waves of many different frequencies or wavelengths. Planck showed that the spectral distribution of the energy radiated by a blackbody is C1 Wb = ----------------------------------C 5 2 T e – 1 (19) Licensed for single user. © 2017 ASHRAE, Inc. where = = = = = = W d = Wbd 0 0 or The total energy emitted per unit time per unit area of a black surface is called the blackbody emissive power Wb and is given by the Stefan-Boltzmann law: Wb T C1 C2 T 4 blackbody spectral (monochromatic) emissive power, W/m3 wavelength, m temperature, K first Planck's law constant = 3.742 10–16 W·m2 second Planck's law constant = 0.014 388 m·K The blackbody spectral emissive power Wb is the energy emitted per unit time per unit surface area at wavelength per unit wavelength band d around ; that is, the energy emitted per unit time per unit surface area in the wavelength band d is equal to Wbd . The Stefan-Boltzmann law can be obtained by integrating Equation (19) over all wavelengths: Wb d = T 4 = Wb 0 Wien showed that the wavelength max , at which the monochromatic emissive power is a maximum (not the maximum wavelength), is given by maxT = 2898 mm·K (20) Equation (20) is Wien's displacement law; the maximum spectral emissive power shifts to shorter wavelengths as temperature increases, such that, at very high temperatures, significant emission eventually occurs over the entire visible spectrum as shorter wavelengths become more prominent. For additional details, see Incropera et al. (2007). Actual Radiation The blackbody emissive power Wb and blackbody spectral emissive power Wb are the maxima at a given surface temperature. Actual surfaces emit less and are called nonblack. The emissive power W of a nonblack surface at temperature T radiating to the hemispherical region above it is given by W = T 4 (21) where is the total emissivity. The spectral emissive power W of a nonblack surface at a particular wavelength is given by W = Wb (22) where is the spectral emissivity, and Wb is given by Equation (19). The relationship between and is given by 1 = ---------4T Wbd (23) 0 If does not depend on , then, from Equation (23), = , and the surface is called gray. Gray surface characteristics are often assumed in calculations. Several classes of surfaces approximate this condition in some regions of the spectrum. The simplicity is desirable, but use care, especially if temperatures are high. The gray assumption is often made because of the absence of information relating as a function of . Emissivity is a function of the material, its surface condition, and its surface temperature. Table 5 lists selected values; Modest (2003) and Siegel and Howell (2002) have more extensive lists. When radiant energy reaches a surface, it is absorbed, reflected, or transmitted through the material. Therefore, from the first law of thermodynamics, ++=1 where = absorptivity (fraction of incident radiant energy absorbed) = reflectivity (fraction of incident radiant energy reflected) = transmissivity (fraction of incident radiant energy transmitted) This is also true for spectral values. For an opaque surface, = 0 and + = 1. For a black surface, = 1, = 0, and = 0. Kirchhoff's law relates emissivity and absorptivity of any opaque surface from thermodynamic considerations; it states that, for any surface where incident radiation is independent of angle or where the surface emits diffusely, = . If the surface is gray, or the incident radiation is from a black surface at the same temperature, then = as well, but many surfaces are not gray. For most surfaces listed in Table 5, the total absorptivity for solar radiation is different from the total emissivity for low-temperature radiation, because and vary with wavelength. Much solar radiation is at short wavelengths. Most emissions from surfaces at moderate temperatures are at longer wavelengths. Platinum black and gold black are almost perfectly black and have absorptivities of about 98% in the infrared region. A small opening in a large cavity approaches blackbody behavior because most of the incident energy entering the cavity is absorbed by repeated reflection within it, and very little escapes the cavity. Thus, the absorptivity and therefore the emissivity of the opening are close to unity. Some flat black paints also exhibit emissivities of 98% over a wide range of conditions. They provide a much more durable surface than gold or platinum black, and are frequently used on radiation instruments and as standard reference in emissivity or reflectance measurements. Example 7. In outer space, the solar energy flux on a surface is 1150 W/m2. Two surfaces are being considered for an absorber plate to be used on the surface of a spacecraft: one is black, and the other is specially coated for a solar absorptivity of 0.94 and emissivity of 0.1. Coolant flowing through the tubes attached to the plate maintains the plate at 340 K. The plate surface is normal to the solar flux. For each surface, determine the (1) heat transfer rate to the coolant per unit area of the plate, and (2) temperature of the surface when there is no coolant flow. Solution: For the black surface, = = 1, =0 Absorbed energy flux = 1150 W/m2 This file is licensed to John Murray (). Publication Date: 6/1/2017 Heat Transfer Table 5 Emissivities and Absorptivities of Some Surfaces Surface Licensed for single user. © 2017 ASHRAE, Inc. 4.13 Total Hemispherical Solar Emissivity Absorptivity* Aluminum Foil, bright dipped Alloy: 6061 Roofing Asphalt 0.03 0.04 0.24 0.88 Brass Oxidized Polished Brick Concrete, rough 0.60 0.04 0.90 0.91 Copper Electroplated Black oxidized in Ebanol C Plate, oxidized 0.03 0.16 0.76 Glass Polished Pyrex Smooth Granite Gravel Ice Limestone Marble Polished or white Smooth Mortar, lime Nickel Electroplated Solar absorber, electro-oxidized on copper Paints Black Parsons optical, silicone high heat, epoxy Gloss Enamel, heated 1000 h at 650 K Silver chromatone White Acrylic resin Gloss Epoxy Paper, roofing or white Plaster, rough Refractory Sand Sandstone, red Silver, polished Snow, fresh Soil Water White potassium zirconium silicate 0.10 0.37 Absorbed energy flux = 0.94 1150 = 1081 W/m2 Emitted energy flux = 0.1 × 757.7 = 75.8 W/m2 Heat flux to coolant = 1081 – 75.8 = 1005 W/m2 Without coolant flow, heat flux to the coolant is zero. Therefore, absorbed energy flux = emitted energy flux. For the black surface, 1150 = 5.67 10–8 Ts4 Ts = 377.1 K For the special surface, 0.94 1150 = 0.1 5.67 10–8 Ts4 Ts = 660.8 K Angle Factor 0.60 0.47 0.91 0.87 to 0.92 0.80 0.91 0.44 0.30 0.96 to 0.97 0.92 The foregoing discussion addressed emission from a surface and absorption of radiation leaving surrounding surfaces. Before radiation exchange among a number of surfaces can be addressed, the amount of radiation leaving one surface that is incident on another must be determined. The fraction of all radiant energy leaving a surface i that is directly incident on surface k is the angle factor Fik (also known as view factor, shape factor, and configuration factor). The angle factor from area Ak to area Aj, Fki, is similarly defined, merely by interchanging the roles of i and k. The following relations assume • All surfaces are gray or black • Emission and reflection are diffuse (i.e., not a function of direction) • Properties are uniform over the surfaces • Absorptivity equals emissivity and is independent of temperature of source of incident radiation • Material located between radiating surfaces neither emits nor absorbs radiation 0.89 to 0.92 0.56 0.90 0.03 0.05 to 0.11 0.22 0.85 0.87 to 0.92 0.94 to 0.97 0.90 0.80 0.24 0.20 These assumptions greatly simplify problems, and give good approximate results in many cases. Some of the relations for the angle factor are as follows. Reciprocity relation. Fik Ai = Fki Ak 0.90 0.85 0.85 0.88 to 0.86 0.89 0.90 to 0.94 0.75 0.59 0.02 0.82 0.94 0.90 0.87 0.26 0.25 (24a) Decomposition relation. For three surfaces i, j, and k, with Aij indicating one surface with two parts denoted by Ai and Aj, Ak Fk-ij = Ak Fk-i + Ak Fk-j (24b) Aij Fij-k = Ai Fi-k + Aj Fj-k (24c) Law of corresponding corners. This law is discussed by Love (1968) and Suryanarayana (1995). Its use is shown in Example 8. Summation rule. For an enclosure with n surfaces, some of which may be inside the enclosure, Fik n = 1 (24d) k=1 0.13 0.98 0.13 Source: Mills (1999). *Values are for extraterrestrial conditions, except for concrete, snow, and water. At Ts = 340 K, emitted energy flux = Wb = 5.67 × 10–8 × 3404 = 757.7 W/m2. In space, there is no convection, so an energy balance on the surface gives Heat flux to coolant = Absorbed energy flux – Emitted energy flux = 1150 – 757.7 = 392.3 W/m2 For the special surface, use solar absorptivity to determine the absorbed energy flux, and emissivity to calculate the emitted energy flux. Note that a concave surface may "see itself," and Fii 0 for such a surface. Numerical values of the angle factor for common geometries are given in Figure 15. For equations to compute angle factors for many configurations, refer to Siegel and Howell (2002). Example 8. A picture window, 3 m long and 1.8 m high, is installed in a wall as shown in Figure 16. The bottom edge of the window is on the floor, which is 6 by 10 m. Denoting the window by 1 and the floor by 234, find F234-1. Solution: From decomposition rule, A234 F234-1 = A2 F2-1 + A3 F3-1 + A4 F4-1 By symmetry, A2 F2-1 = A4 F4-1 and A234-1 = A3 F3-1 + 2A2 F2-1. A23 F23-15 = A2 F2-1+ A2F2-5 This file is licensed to John Murray (). Publication Date: 6/1/2017 2017 ASHRAE Handbook—Fundamentals (SI) Licensed for single user. © 2017 ASHRAE, Inc. 4.14 Fig. 15 Radiation Angle Factors for Various Geometries + A3F3-1 + A3F3-5 From the law of corresponding corners, A2 F2-1 = A3 F3-5; therefore A23 F23-5 = A2 F2-5 + A3 F3-1 + 2A2 F2-1. Thus, A234 F234-1 = A3 F3-1 + A23 F23-15 – A2 F2-5 – A3 F3-1 = A23 F23-15 – A2 F2-5 A234 = 60 m2 A23 = 45 m2 A2 = 15 m2 From Figure 15A with Y/X = 10/6 = 1.67 and Z/X = 1.8/4.5 = 0.4, F23-15 = 0.061. With Y/X = 10/1.5 = 6.66 and Z/X = 1.8/1.5 = 1.2, F25 = 0.041. Substituting the values, F234-1 = 1/60(45 0.061 – 15 0.041) = 0.036. Radiant Exchange Between Opaque Surfaces A surface Ai radiates energy at a rate independent of its surroundings. It absorbs and reflects incident radiation from surrounding surfaces at a rate dependent on its absorptivity. The net heat transfer rate qi is the difference between the rate radiant energy leaves the surface and the rate of incident radiant energy; it is the rate at which energy must be supplied from an external source to maintain the Fig. 16 Diagram for Example 8 This file is licensed to John Murray (). Publication Date: 6/1/2017 Heat Transfer 4.15 surface at a constant temperature. The net radiant heat flux from a surface Ai is denoted by qi . Several methods have been developed to solve specific radiant exchange problems. The radiosity method and thermal circuit method are presented here. Consider the heat transfer rate from a surface of an n-surface enclosure with an intervening medium that does not participate in radiation. All surfaces are assumed gray and opaque. The radiosity Ji is the total rate of radiant energy leaving surface i per unit area (i.e., the sum of energy flux emitted and energy flux reflected): Ji = iWb + iGi (25) where Gi is the total rate of radiant energy incident on surface i per unit area. For opaque gray surfaces, the reflectivity is i = 1 – i = 1 – i Thus, Ji = iWb + (1– i)Gi (26) Licensed for single user. © 2017 ASHRAE, Inc. Note that for a black surface, = 1, = 0, and J = Wb . The net radiant energy transfer qi is the difference between the total energy leaving the surface and the total incident energy: qi = Ai (Ji – Gi) (27) Eliminating Gi between Equations (26) and (27), (28) Radiosity Method. Consider an enclosure of n isothermal surfaces with areas of A1, A2, ..., An, and emissivities of 1, 2, ..., n, respectively. Some may be at uniform but different known temperatures, and the remaining surfaces have uniform but different and known heat fluxes. The radiant energy flux incident on a surface Gi is the sum of the radiant energy reaching it from each of the n surfaces: Fki Jk Ak = Fik Jk Ai n n k=1 k=1 or G i = Fik Jk n (29) k=1 Substituting Equation (29) into Equation (26), J i = i W bi + 1 – i F ik J k n (30) k=1 Combining Equations (30) and (28), qi J i = ----- + Ai Fik Jk n (31) k=1 Note that in Equations (30) and (31), the summation includes surface i. Equation (30) is for surfaces with known temperatures, and Equation (31) for those with known heat fluxes. An opening in the enclosure is treated as a black surface at the temperature of the surroundings. The resulting set of simultaneous, linear equations can be solved for the unknown Ji s. Once the radiosities (Ji s) are known, the net radiant energy transfer to or from each surface or the emissive power, whichever is unknown is determined. For surfaces where Ebi is known and qi is to be determined, use Equation (28) for a nonblack surface. For a black surface, Ji = Wbi and Equation (31) can be rearranged to give Fik Jk n (32) k=1 At surfaces where qi is known and Ebi is to be determined, rearrange Equation (28): 1 – i E bi = J i + q i ------------ Ai i (33) The temperature of the surface is then W bi1 4 T i = --------- (34) A surface in radiant balance is one for which radiant emission is balanced by radiant absorption (i.e., heat is neither removed from nor supplied to the surface). These are called reradiating, insulated, or refractory surfaces. For these surfaces, qi = 0 in Equation (31). After solving for the radiosities, Wbi can be found by noting that qi = 0 in Equation (33) gives Wbi = Ji. Thermal Circuit Method. Another method to determine the heat transfer rate is using thermal circuits for radiative heat transfer rates. Heat transfer rates from surface i to surface k and surface k to surface i, respectively, are given by qi-k = Ai Fi-k (Ji – Jk) W bi – J i qi = ----------------------------- 1 – i i Ai Gi Ai = qi ----- = W bi – Ai and qk-i = Ak Fk-i (Jk – Ji) Using the reciprocity relation Ai Fi-k = Ak Fk-i, the net heat transfer rate from surface i to surface k is Ji – Jk qik = qi-k – qk-i = Ai Fi-k (Ji – Jk) = --------------------1 A i F i-k (35) Equations (28) and (35) are analogous to the current in a resistance, with the numerators representing a potential difference and the denominator representing a thermal resistance. This analogy can be used to solve radiative heat transfer rates among surfaces, as illustrated in Example 9. Using angle factors and radiation properties as defined assumes that the surfaces are diffuse radiators, which is a good assumption for most nonmetals in the infrared region, but poor for highly polished metals. Subdividing the surfaces and considering the variation of radiation properties with angle of incidence improves the approximation but increases the work required for a solution. Also note that radiation properties, such as absorptivity, have significant uncertainties, for which the final solutions should account. Example 9. Consider a 4 m wide, 5 m long, 2.5 m high room as shown in Figure 17. Heating pipes, embedded in the ceiling (1), keep its temperature at 40°C. The floor (2) is at 30°C, and the four side walls (3) are at 18°C. The emissivity of each surface is 0.8. Determine the net radiative heat transfer rate to/from each surface. Solution: Consider the room as a three-surface enclosure. The corresponding thermal circuit is also shown. The heat transfer rates are found after finding the radiosity of each surface by solving the thermal circuit. From Figure 15A, F1-2 = F2-1 = 0.376 From the summation rule, F1-1 + F1-2 + F1-3 = 1. With F1-1 = 0, F1-3 = 1 – F1-2 = 0.624 = F2-3 1 – 1 1 – 0.8 R1 = ------------- = ------------------- = 0.0125 m–2 = R2 A1 1 20 0.8 This file is licensed to John Murray (). Publication Date: 6/1/2017 4.16 2017 ASHRAE Handbook—Fundamentals (SI) Table 6 Emissivity of CO2 and Water Vapor in Air at 24°C Path Length, m 3 30 300 CO2, % by Volume Relative Humidity, % 0.1 0.3 1.0 10 50 100 0.03 0.09 0.16 0.06 0.12 0.19 0.09 0.16 0.23 0.06 0.22 0.47 0.17 0.39 0.64 0.22 0.47 0.70 Table 7 Emissivity of Moist Air and CO2 in Typical Room Fig. 17 Diagrams for Example 9 Licensed for single user. © 2017 ASHRAE, Inc. 1 1 R13 = -------- -------- = ------------------------- = 0.080 13 m–2 = R23 A 1 F 1-3 20 0.624 Performing a balance on each of the three Ji nodes gives Surface 1: W b1 – J 1 J 2 – J 1 J 3 – J 1 --------------------- + ---------------- + ---------------- = 0 R 13 R 12 R1 Surface 2: W b2 – J 2 J 1 – J 2 J 3 – J 2 --------------------- + ---------------- + ---------------- = 0 R 23 R 12 R2 Surface 3: W b3 – J 3 J 1 – J 3 J 2 – J 3 --------------------- + ---------------- + ---------------- = 0 R 23 R 13 R3 W b2 = 479.2 W/m 4 313.2 = 545.6 W/m 2 W b3 = 407.7 W/m 2 Substituting the values and solving for J1, J2, and J3, J1 = 524.5 W/m2 J2 = 475.1 W/m2 J3 = 418.9 W/m2 W b1 – J 1 545.6 – 524.5 q 1 = --------------------- = ---------------------------------- = 1688 W R1 0.0125 q 2 = 328 W 0.10 0.19 0.22 L = 1 – e – L 1 1 R12 = ---------------- = -------------------------- = 0.133 m–2 A 1 F 1-2 20 0.376 –8 g 10 50 75 Beer's law states that the attenuation of radiant energy in a gas is a function of the product pg L of the partial pressure of the gas and the path length. The monochromatic absorptivity of a body of gas of thickness L is then 1 – 3 1 – 0.8 R3 = ------- -------- = ------------------- = 0.005 556 m–2 A3 3 45 0.8 W b1 = 5.67 10 Relative Humidity, % q 3 = – 2016 W Note that floor and ceiling must both be heated because of heat loss from the walls. Radiation in Gases Monatomic and diatomic gases such as oxygen, nitrogen, hydrogen, and helium are essentially transparent to thermal radiation. Their absorption and emission bands are confined mainly to the ultraviolet region of the spectrum. The gaseous vapors of most compounds, however, have absorption bands in the infrared region. Carbon monoxide, carbon dioxide, water vapor, sulfur dioxide, ammonia, acid vapors, and organic vapors absorb and emit significant amounts of energy. Radiation exchange by opaque solids may be considered a surface phenomenon unless the material is transparent or translucent, though radiant energy does penetrate into the material. However, the penetration depths are small. Penetration into gases is very significant. (36) Because absorption occurs in discrete wavelength bands, the absorptivities of all the absorption bands must be summed over the spectral region corresponding to the temperature of the blackbody radiation passing through the gas. The monochromatic absorption coefficient is also a function of temperature and pressure of the gas; therefore, detailed treatment of gas radiation is quite complex. Estimated emissivity for carbon dioxide and water vapor in air at 24°C is a function of concentration and path length (Table 6). Values are for an isothermal hemispherically shaped body of gas radiating at its surface. Among others, Hottel and Sarofim (1967), Modest (2003), and Siegel and Howell (2002) describe geometrical calculations in their texts on radiation heat transfer. Generally, at low values of pg L, the mean path length L (or equivalent hemispherical radius for a gas body radiating to its surrounding surfaces) is four times the mean hydraulic radius of the enclosure. A room with a dimensional ratio of 1:1:4 has a mean path length of 0.89 times the shortest dimension when considering radiation to all walls. For a room with a dimensional ratio of 1:2:6, the mean path length for the gas radiating to all surfaces is 1.2 times the shortest dimension. The mean path length for radiation to the 2 by 6 face is 1.18 times the shortest dimension. These values are for cases where the partial pressure of the gas times the mean path length approaches zero (pg L 0). The factor decreases with increasing values of pg L. For average rooms with approximately 2.4 m ceilings and relative humidity ranging from 10 to 75% at 24°C, the effective path length for carbon dioxide radiation is about 85% of the ceiling height, or 2 m. The effective path length for water vapor is about 93% of the ceiling height, or 2.3 m. The effective emissivity of the water vapor and carbon dioxide radiating to the walls, ceiling, and floor of a room 4.9 by 14.6 m with 2.4 m ceilings is in Table 7. Radiation heat transfer from the gas to the walls is then q = Awg T g4 – T w4 (37) The preceding discussion indicates the importance of gas radiation in environmental heat transfer problems. In large furnaces, gas radiation is the dominant mode of heat transfer, and many additional factors must be considered. Increased pressure broadens the spectral bands, and interaction of different radiating species prohibits simple summation of emissivity factors for the individual species. Nonblackbody conditions require separate calculations of emissivity and absorptivity. Hottel and Sarofim (1967) and McAdams (1954) discuss gas radiation more fully. This file is licensed to John Murray (). Publication Date: 6/1/2017 Heat Transfer 4. 4.17 THERMAL CONVECTION Convective heat transfer coefficients introduced previously can be estimated using correlations presented in this section. Forced Convection Forced-air coolers and heaters, forced-air- or water-cooled condensers and evaporators, and liquid suction heat exchangers are examples of equipment that transfer heat primarily by forced convection. Although some generalized heat transfer coefficient correlations have been mathematically derived from fundamentals, they are usually obtained from correlations of experimental data. Most correlations for forced convection are of the form hL c Nu = -------- = f (ReLc, Pr) k Licensed for single user. © 2017 ASHRAE, Inc. where Nu h Lc ReLc V Pr cp k = = = = = = = = = = = Nusselt number convection heat transfer coefficient characteristic length VLc /=VLc / fluid velocity Prandtl number = cp /k fluid specific heat fluid dynamic viscosity fluid density kinematic viscosity = / fluid conductivity front edge or other irregularities, it can start at much smaller Reynolds numbers. Internal Flow. For tubes, channels, or ducts of small diameter at sufficiently low velocity, the laminar boundary layers on each wall grow until they meet. This happens when the Reynolds number based on tube diameter, Re = Vavg D/, is less than 2000 to 2300. Beyond this point, the velocity distribution does not change, and no transition to turbulent flow occurs. This is called fully developed laminar flow. When the Reynolds number is greater than 10 000, the boundary layers become turbulent before they meet, and fully developed turbulent flow is established (Figure 19). If flow is turbulent, three different flow regions exist. Immediately next to the wall is a laminar sublayer, where heat transfer occurs by thermal conduction; next is a transition region called the buffer layer, where both eddy mixing and conduction effects are significant; the final layer, extending to the pipe's axis, is the turbulent region, where the dominant mechanism of transfer is eddy mixing. In most equipment, flow is turbulent. For low-velocity flow in small tubes, or highly viscous liquids such as glycol, the flow may be laminar. The characteristic length for internal flow in pipes and tubes is the inside diameter. For noncircular tubes or ducts, the hydraulic diameter Dh is used to compute the Reynolds and Nusselt numbers. It is defined as Cross-sectional area for flow D h = 4 ---Total wetted perimeter (38) Inserting expressions for cross-sectional area and wetted perimeter of common cross sections shows that the hydraulic diameter is equal to Fluid velocity and characteristic length depend on the geometry. External Flow. When fluid flows over a flat plate, a boundary layer forms adjacent to the plate. The velocity of fluid at the plate surface is zero and increases to its maximum free-stream value at the edge of the boundary layer (Figure 18). Boundary layer formation is important because the temperature change from plate to fluid occurs across this layer. Where the boundary layer is thick, thermal resistance is great and the heat transfer coefficient is small. Flow within the boundary layer immediately downstream from the leading edge is laminar. As flow proceeds along the plate, the laminar boundary layer increases in thickness to a critical value. Then, turbulent eddies develop in the boundary layer, except in a thin laminar sublayer adjacent to the plate. The boundary layer beyond this point is turbulent. The region between the breakdown of the laminar boundary layer and establishment of the turbulent boundary layer is the transition region. Because turbulent eddies greatly enhance heat transport into the main stream, the heat transfer coefficient begins to increase rapidly through the transition region. For a flat plate with a smooth leading edge, the turbulent boundary layer starts at distance xc from the leading edge where the Reynolds number Re = Vxc / is in the range 300 000 to 500 000 (in some cases, higher). In a plate with a blunt Table 8 lists various forced-convection correlations. In general, the Nusselt number is determined by the flow geometry, Reynolds number, and Prandtl number. One often useful form for turbulent internal flow is known as Colburn's analogy: Fig. 18 External Flow Boundary Layer Build-up (Vertical Scale Magnified) Fig. 19 Boundary Layer Build-up in Entrance Region of Tube or Channel • • • • The diameter of a round pipe Twice the gap between two parallel plates The difference in diameters for an annulus The length of the side for square tubes or ducts fF Nu j = ------------------ = ---13 2 RePr where fF is the Fanning friction factor (1/4 of the Darcy-Weisbach friction factor in Chapter 3) and j is the Colburn j-factor. It is related to the friction factor by the interrelationship of the transport of momentum and energy in turbulent flow. These factors are plotted in Figure 20. This file is licensed to John Murray (). Publication Date: 6/1/2017 4.18 2017 ASHRAE Handbook—Fundamentals (SI) Table 8 Forced-Convection Correlations Nu = f (Re, Pr) I. General Correlation II. Internal Flows for Pipes and Ducts: Characteristic length = D, pipe diameter, or Dh, hydraulic diameter. m· D h QD h V avg D h 4Q 4m· Re = --------------------- = ----------- = ----------- = ------------- = ------------where m· = mass flow rate, Q = volume flow rate, Pwet = wetted perimeter, P wet Ac Ac P wet A = cross-sectional area, and = kinematic viscosity (/). c f Nu Colburn's analogy (turbulent) = -------------------1/3 2 Re Pr 1/3 0.14 L Re Pr 0.42 Re Pr ----Nu = 1.86 ---------------- ---Laminar : Re < 2300 ---- ---------------- L D s D 8 s 0.065 D L Re Pr Developing Nu = 3.66 + --2/3 1 + 0.04 D L Re Pr Fully developed, round Nu = 3.66 Uniform surface temperature Licensed for single user. © 2017 ASHRAE, Inc. Turbulent: (T8.1) a (T8.2) (T8.3) (T8.4a) Nu = 4.36 Uniform heat flux (T8.4b) Nu = 0.023 Re4/5 Pr 0.4 Heating fluid Re 10 000 Cooling fluid Re 10 000 (T8.5a)b Fully developed Nu = 0.023 Re4/5 Pr 0.3 Evaluate properties at bulk temperature tb except s and ts at surface temperature f s 2 Re – 1000 Pr D 2/3 Nu = -- 1 + ---- 1/2 2/3 L 1 + 12.7 f s 2 Pr – 1 For fully developed flows, set D/L = 0. Nu = 0.027 Re 4/5 Pr (T8.5b)b 1 f s = ---2 1.58 ln Re – 3.28 Multiply Nu by (T/Ts)0.45 for gases and by (Pr/Prs)0.11 for liquids 1/3 0.14 ----- s c (T8.6) a For viscous fluids (T8.7) For noncircular tubes, use hydraulic mean diameter Dh in the equations for Nu for an approximate value of h. III. External Flows for Flat Plate: Characteristic length = L = length of plate. Re = VL/. All properties at arithmetic mean of surface and fluid temperatures. Laminar boundary layer: Nu = 0.332 Re1/2 Pr 1/3 Re < 5 × 105 Nu = 0.664 Re1/2 Pr 1/3 Nu = 0.0296 Re4/5 Pr 1/3 Turbulent boundary layer: Re > 5 × 105 Turbulent boundary layer Nu = 0.037 Re4/5 Pr 1/3 beginning at leading edge: All Re Nu = (0.037 Re4/5 – 871)Pr 1/3 Laminar-turbulent boundary layer: Re > 5 × 105 Local value of h IV. External Flows for Cross Flow over Cylinder: Characteristic length = D = diameter. Re = VD/. All properties at arithmetic mean of surface and fluid temperatures. 5/8 0.62 Re Pr Re Nu = 0.3 + --- 1 + ------------------ 282 000 2/3 1/4 1 + 0.4 Pr 1/2 1/3 (T8.8) Average value of h Local value of h (T8.9) (T8.10) Average value of h (T8.11) Average value Rec = 5 × 105 (T8.12) Average value of h 4/5 (T8.14) d V. Simplified Approximate Equations: h is in W/(m2 ·K), V is in m/s, D is in m, and t is in °C. Atmospheric air (0 to 200°C): h = (3.76 – 0.00497t)V 0.8/D 0.2 Water (3 to 200°C): h = (1206 + 23.9t)V 0.8/D 0.2 Water (4 to 104°C): h = (1431 + 20.9t)V 0.8/D 0.2 (McAdams 1954) Atmospheric air: 0°C < t < 200°C, where t = arithmetic mean of air and surface temperature. 35 < Re < 5000 h = 2.755V 0.471/D 0.529 5000 < Re < 50 000 h = (4.22 – 0.002 57t)V 0.633/D 0.367 Water: 5°C < t < 90°C, where t = arithmetic mean of water and surface temperature. h = (461.8 + 2.01t)V 0.471/D 0.529 35 < Re < 5000 5000 < Re < 50 000 h = (1012 + 9.19t)V 0.633/D 0.367 Flows in pipes Re > 10 000 Flow over cylinders (T8.15a) e (T8.15b)e (T8.15c)g (T8.16a) (T8.16b) (T8.17a) (T8.17b)f VI. External Flow over Spheres: Characteristic length = D = diameter. Re = VD/. All properties at arithmetic mean of surface and fluid bulk temperature, except s at surface temperature. 12 Nu D = 2 + (0.4 Re D Sources: aSieder gMcAdams and Tate (1936), (1954). bDittus and Boelter (1930), cGnielinski 23 + 0.06 Re D)Pr0.4(/s)1/4 (1990), dChurchill and Bernstein (1977), (T8.18) eBased on Nu = 0.023 Re 4/5Pr 1/3, fBased on Morgan (1975). This file is licensed to John Murray (). Publication Date: 6/1/2017 Heat Transfer 4.19 Note that for gases, the (/s)1/4 property correction is often negligible. For liquids, it is often significant. 2 k 0.02588 W/ m K h = ---- Nu = -- (1056) = 9.05 W/(m2 ·K) 3.02 m D The rate of heat transfer to the iced water is q = hAs(Ts – T) = h(D2)(Ts – T) = [9.05 W/(m2 ·K)][(3.02 m)2](30 – 0)K = 7780 W With a uniform tube surface temperature and heat transfer coefficient, the exit temperature can be calculated using ts – te hA ln ------------ = – --------ts – ti m· c p Licensed for single user. © 2017 ASHRAE, Inc. Fig. 20 Typical Dimensionless Representation of ForcedConvection Heat Transfer (39) where ti and te are the inlet and exit bulk temperatures of the fluid, ts is the pipe/duct surface temperature, and A is the surface area inside the pipe/duct. The convective heat transfer coefficient varies in the direction of flow because of the temperature dependence of the fluid properties. In such cases, it is common to use an average value of h in Equation (39) computed either as the average of h evaluated at the inlet and exit fluid temperatures or evaluated at the average of the inlet and exit temperatures. With uniform surface heat flux q, the temperature of fluid t at any section can be found by applying the first law of thermodynamics: (40) m· c (t – t) = qA p i The surface temperature can be found using q = h(ts – t) Fig. 21 Heat Transfer Coefficient for Turbulent Flow of Water Inside Tubes Simplified correlations for atmospheric air are also given in Table 8. Figure 21 gives graphical solutions for water. Example 10. An uninsulated 3 m ID spherical tank with 0.01 m walls is used to store iced water. Winds at V = 25 km/h and T = 30°C blow over the outside of the tank. What is the rate of heat gain to the iced water? Neglect radiation, conduction resistance of tank wall, and convection resistance between ice water and tank; the outer surface temperature of the tank is 0°C. Because the geometry is for forced convection over a sphere, use Equation (T8.18). Use the properties of air at 101.325 kPa pressure and the freestream temperature of T = 30°C: k = 0.0259 W/(mK), = 1.608 × 10–5 m2/s, = 1.872 × 10–5 kg/(m·s), Pr = 0.728, and s = 1.729 × 10–5 kg/(m·s) at 0°C. VD 25 1000 3600 m/s 3.02 m - = 1.304 106 Re = -------- = --5 2 1.608 10 m /s hD 0.5 23 0.4 14 Nu = ------- = 2 + 0.4Re + 0.06Re Pr s k 6 0.5 = 2 + 0.4 1.304 10 0.7282 0.4 1.872 6 23 + 0.06 1.304 10 –5 10 - ------------------------------ 1.729 10 – 5 14 = 1056 (41) With uniform surface heat flux, surface temperature increases in the direction of flow along with the fluid. Natural Convection. Heat transfer with fluid motion resulting solely from temperature differences (i.e., from temperaturedependent density and gravity) is natural (free) convection. Naturalconvection heat transfer coefficients for gases are generally much lower than those for forced convection, and it is therefore important not to ignore radiation in calculating the total heat loss or gain. Radiant transfer may be of the same order of magnitude as natural convection, even at room temperatures; therefore, both modes must be considered when computing heat transfer rates from people, furniture, and so on in buildings (see Chapter 9). Natural convection is important in a variety of heating and refrigeration equipment, such as (1) gravity coils used in high-humidity cold-storage rooms and in roof-mounted refrigerant condensers, (2) the evaporator and condenser of household refrigerators, (3) baseboard radiators and convectors for space heating, and (4) cooling panels for air conditioning. Natural convection is also involved in heat loss or gain to equipment casings and interconnecting ducts and pipes. Consider heat transfer by natural convection between a cold fluid and a hot vertical surface. Fluid in immediate contact with the surface is heated by conduction, becomes lighter, and rises because of the difference in density of the adjacent fluid. The fluid's viscosity resists this motion. The heat transfer rate is influenced by fluid properties, temperature difference between the surface at ts and environment at t , and characteristic dimension Lc . Some generalized heat transfer coefficient correlations have been mathematically derived from fundamentals, but they are usually obtained from correlations of experimental data. Most correlations for natural convection are of the form hL c Nu = -------- = f Ra Lc Pr k where Nu = Nusselt number H = convection heat transfer coefficient This file is licensed to John Murray (). Publication Date: 6/1/2017 4.20 Lc K RaLc t g Pr 2017 ASHRAE Handbook—Fundamentals (SI) = = = = = = = = = Correlations for a number of geometries are given in Table 9. Other information on natural convection is available in the Bibliography under Heat Transfer, General. Comparison of experimental and numerical results with existing correlations for natural convective heat transfer coefficients indicates that caution should be used when applying coefficients for (isolated) vertical plates to vertical surfaces in enclosed spaces (buildings). Altmayer et al. (1983) and Bauman et al. (1983) developed improved characteristic length fluid thermal conductivity Rayleigh number = g tL3c / ts – t | gravitational acceleration coefficient of thermal expansion fluid kinematic viscosity = / fluid thermal diffusivity = k/cp Prandtl number = / Table 9 Natural Convection Correlations Nu = f (Ra, Pr) or f (Ra) I. General relationships Characteristic length depends on geometry (T9.1) 2 3 cp Pr = --------- t = t s – t k g t L Gr = ------------------------------2 Ra = Gr Pr II. Vertical plate 1/4 0.67Ra Nu = 0.68 + ---9/16 4/9 1 + 0.492 Pr ts = constant 10–1 < Ra < 109 (T9.2)a 1/6 0.387Ra Nu = 0.825 + --- 9/16 8/27 1 + 0.492 Pr 109 < Ra < 1012 (T9.3)a 1/6 0.387Ra Nu = 0.825 + -- 9/16 8/27 1 + 0.437 Pr 10–1 < Ra < 1012 (T9.4)a 1 < Ra < 200 200 < Ra < 104 2.2 × 104 < Ra < 8 × 106 8 × 106 < Ra < 1.5 × 109 105 < Ra < 1010 (T9.5)b (T9.6)b (T9.7)b (T9.8)b (T9.9)b 2 Characteristic dimension: L = height Licensed for single user. © 2017 ASHRAE, Inc. Properties at (ts + t)/2 except at t qs = constant 2 Characteristic dimension: L = height Properties at ts, L/2 – t except at t Equations (T9.2) and (T9.3) can be used for vertical cylinders if D/L > 35/Gr1/4 where D is diameter and L is axial length of cylinder III. Horizontal plate Characteristic dimension = L = A/P, where A is plate area and P is perimeter Properties of fluid at (ts + t)/2 Downward-facing cooled plate and upward-facing heated plate Nu = 0.96 Ra1/6 Nu = 0.59 Ra1/4 Nu = 0.54 Ra1/4 Nu = 0.15 Ra1/3 Downward-facing heated plate and upward-facing cooled plate Nu = 0.27 Ra1/4 IV. Horizontal cylinder 1/6 0.387 Ra Nu = 0.6 + -- 9/16 8/27 1 + 0.559 Pr 2 Characteristic length = d = diameter Properties of fluid at (ts + t)/2 except at t V. Sphere 109 < Ra < 1013 (T9.10)c Ra < 1011 (T9.11)d 10–8 < Ra < 106 (T9.12)e 1/4 0.589 Ra Nu = 2 + ---9/16 4/9 1 + 0.469 Pr Characteristic length = D = diameter Properties at (ts + t)/2 except at t VI. Horizontal wire Characteristic dimension = D = diameter Properties at (ts + t)/2 3.3 2 ------- = ln 1 + -----------n- Nu cRa VII. Vertical wire Characteristic dimension = D = diameter; L = length of wire Nu = c (Ra D/L)0.25 + 0.763c (1/6)(Ra D/L)(1/24) c (Ra D/L)0.25 > 2 × 10–3 (T9.13)e 0.671 In both Equations (T9.12) and (T9.13), c = -- and 9/16 4/9 1 + 0.492 Pr 1 n = 0.25 + --------------------------------------0.175 10 + 5 Ra Properties at (ts + t)/2 VIII. Simplified equations with air at mean temperature of 21°C: h is in W/(m2 ·K), L and D are in m, and t is in °C. h = 1.33 -----t L Vertical surface h = 1.26 t h = 1.23 t Sources: and Chu (1975a), 1/3 T h = 1.04 ------- D Horizontal cylinder a Churchill 1/4 bLloyd and Moran (1974), Goldstein et al. (1973), 1/4 1/3 cChurchill and Chu (1975b), dChurchill 105 < Ra < 109 (T9.14) Ra > 109 (T9.15) 105 < Ra < 109 (T9.16) Ra > 109 (T9.17) (1990), eFujii et al. (1986). This file is licensed to John Murray (). Publication Date: 6/1/2017 Heat Transfer 4.21 correlations for calculating natural convective heat transfer from vertical surfaces in rooms under certain temperature boundary conditions. Natural convection can affect the heat transfer coefficient in the presence of weak forced convection. As the forced-convection effect (i.e., the Reynolds number) increases, "mixed convection" (superimposed forced-on-free convection) gives way to pure forced convection. In these cases, consult other sources [e.g., Grigull et al. (1982); Metais and Eckert (1964)] describing combined free and forced convection, because the heat transfer coefficient in the mixed-convection region is often larger than that calculated based on the natural- or forced-convection calculation alone. Metais and Eckert (1964) summarize natural-, mixed-, and forced-convection regimes for vertical and horizontal tubes. Figure 22 shows the approximate limits for horizontal tubes. Other studies are described by Grigull et al. (1982). Licensed for single user. © 2017 ASHRAE, Inc. Example 11. A horizontal electric immersion heater of 10 mm diameter D and 400 mm length L is rated at q = 733 W. Estimate the surface temperature ts if immersed in 20°C t water. Solution: Because the geometry is for natural convection from a horizontal cylinder, use Equation (T9.10). If the allowable surface temperature of the heater were given, and the need were to calculate the maximum allowable power rating, this would be a straightforward sequence of calculations: 1. Calculate the film temperature tf = ts + t 2. Look up the needed water properties k, , Pr, , and at tf. 3. Calculate Rayleigh number Ra. 4. Use Equation (T9.10) to calculate Nusselt number Nu. 5. Calculate h = Nuk/D. 6. q = hDL(ts – t) However, in this case, given q and needing ts, the solution is iterative. First, guess the solution ts, execute items 1 to 5 in the preceding sequence, and use the resulting h to calculate ts = t + q/(hDL). If necessary, use this value as a new guess for ts and repeat the process. Assume ts = 64°C. Then tf = (64 + 20)/2 = 42°C = 315 K. Use k = 0.634 W/(m·K), = 6.25 × 10–7 m2/s, = 1.53 × 10–7 m2/s, Pr = 4.08, and = 0.4004 × 10–3/K. 3 gTD Ra = -------------------- = 1.804 106 2 16 0.387Ra Nu = 0.6 + -- = 20.5 8 27 9 16 1 + 0.559 Pr h = Nuk/D = 1301 W/(m2 ·K) ts = t + q/(hDL) = 64.8°C The initial guess was close enough; another iteration is unnecessary. Example 12. Chilled water at 5°C flows inside a freely suspended horizontal 20 mm OD pipe at a velocity of 2.5 m/s. Surrounding air is at 30°C, 70% rh. The pipe is to be insulated with cellular glass having a thermal conductivity of 0.045 W/(m·K). Determine the radial thickness of the insulation to prevent condensation of water on the outer surface. Solution: In Figure 23, tfo = 30°C tf i = 5°C From the problem statement, the outer surface temperature of the insulation should not be less than the dew-point temperature of air. The dew-point temperature of air at 30°C, 70% rh = 23.93°C. To determine the outer diameter of the insulation, equate the heat transfer rate per unit length of pipe (from the outer surface of the pipe to the water) to the heat transfer rate per unit length from the air to the outer surface: t o – t fi t fo – t o --- = -------------1 d 1 1 o --------------------- + ------- ln ----h ot d o h i d i 2k i di (42) Heat transfer from the outer surface is by natural convection to air, so the surface heat transfer coefficient hot is the sum of the convective heat transfer coefficient ho and the radiative heat transfer coefficient hr . With an assumed emissivity of 0.7 and using Equation (4), hr = 4.3 W/(m2 ·K). To determine the value of do, the values of the heat transfer coefficients associated with the inner and outer surfaces (hi and ho, respectively) are needed. Compute the value of hi using Equation (T8.5a). Properties of water at an assumed temperature of 5°C are w = 1000 kg/m3 w = 0.001 518 (N·s)/m2 kw = 0.5708 W/(m·K) cpw = 4197 W/(m·K) Prw = 11.16 vd Red = --------- = 32 944 Nud = 248.3 kw = 7087 W/(m2 ·K) To compute ho using Equation (T9.10a), the outer diameter of the insulation material must be found. Determine it by iteration by assuming a value of do, computing the value of ho, and determining the value of do from Equation (42). If the assumed and computed values of do are close to each other, the correct solution has been obtained. Otherwise, recompute ho using the newly computed value of do and repeat the process. Assume do = 0.05 m. Properties of air at tf = 27°C and 101.325 kPa are = 1.176 kg/m3 Fig. 22 Regimes of Free, Forced, and Mixed Convection— Flow in Horizontal Tubes do = OD of tube = 0.02 m ki = thermal conductivity of insulation material = 0.045 W/(m·K) k = 0.025 66 W/(m·K) = 1.858 10–5 (N·s)/m2 Fig. 23 Diagram for Example 12 This file is licensed to John Murray (). Publication Date: 6/1/2017 4.22 2017 ASHRAE Handbook—Fundamentals (SI) Pr = 0.729 Ra = 74 574 For a given exchanger type, heat transfer effectiveness can generally be expressed as a function of the number of transfer units (NTU) and the capacity rate ratio cr : = 0.003 299 (at 273.15 + 30 = 293.15 K) Nu = 7.223 ho = 3.65 W/(m2 ·K) hot =3.65 + 4.3 = 7.95 W/(m2 ·K) Solving for do makes the left side of Equation (42) equal to the right side, and gives do = 0.040 03 m. Now, using the new value of 0.040 03 m for the outer diameter, the new values of ho and hot are 3.86 W/(m2 ·K) and 8.20 W/(m2 ·K), respectively. The updated value of do is 0.044 03 m. Repeating the process several times results in a final value of do = 0.047 17 m. Thus, an outer diameter of 0.045 m (corresponding to an insulation radial thickness of 12.5 mm) keeps the outer surface temperature at 24.1°C, higher than the dew point. [Another method is to use Equation (42) to solve for to for values of do corresponding to available insulation thicknesses and using the insulation thickness that keeps to above the dew-point temperature.] 5. = f (NTU, cr , Flow arrangement) where NTU = UA/Cmin cr = Cmin /Cmax Effectiveness is independent of exchanger inlet temperatures. For any exchanger in which cr is zero (where one fluid undergoing a phase change, as in a condenser or evaporator, has an effective cp =), the effectiveness is = 1 – exp(–NTU) With heat transfer from one fluid to another (separated by a solid surface) flowing through a heat exchanger, the local temperature difference t varies along the flow path. Heat transfer rate may be calculated using Licensed for single user. © 2017 ASHRAE, Inc. (48) t hi – t ci t m = ------------------------NTU Mean Temperature Difference Analysis (44) where t1 and t2 are temperature differences between the fluids at each end of the heat exchanger; tm is the logarithmic mean temperature difference (LMTD). For the special case of t1 = t2 (possible only with a counterflow heat exchanger with equal capacities), which leads to an indeterminate form of Equation (44), tm = t1 = t2. Equation (44) for tm is true only if the overall coefficient and the specific heat of the fluids are constant through the heat exchanger, and no heat losses occur (often well-approximated in practice). Parker et al. (1969) give a procedure for cases with variable overall coefficient U. For heat exchangers other than parallel and counterflow, a correction factor [see Incropera et al. (2007)] is needed for Equation (44) to obtain the correct mean temperature difference. (49) After finding the heat transfer rate q, exit temperatures for constant-density fluids are found from (43) where U is the overall uniform heat transfer coefficient, A is the area associated with the coefficient U, and tm is the appropriate mean temperature difference. For a parallel or counterflow heat exchanger, the mean temperature difference is given by tm = (t1 – t2)/ln(t1/t2) cr = 0 The mean temperature difference in Equation (44) is then given by HEAT EXCHANGERS q = UA tm (47) q t e – t i = --------- m cp (50) Effectiveness for selected flow arrangements are given in Table 10. Afgan and Schlunder (1974), Incropera, et al. (2007), and Kays and London (1984) present graphical representations for convenience. NTUs as a function of expressions are available in Incropera et al. (2007). Example 13. Flue gases from a gas-fired furnace are used to heat water in a 5 m long counterflow, double-pipe heat exchanger. Water enters the inner, thin-walled 40 mm diameter pipe at 40°C with a velocity of 0.5 m/s. Flue gases enter the annular space with a mass flow rate of 0.12 kg/s at 200°C. To increase the heat transfer rate to the gases, 16 rectangular axial copper fins are attached to the outer surface of the inner pipe. Each fin is 60 mm high (radial height) and 1 mm thick, as shown in Figure 24. The gas-side surface heat transfer coefficient is 115 W/(m2 ·K). Find the heat transfer rate and the exit temperatures of the gases and water. The heat exchanger has the following properties: Water in the pipe tci = 40°C Gases thi = 200°C Length of heat exchanger Ltube = 5 m vc = 0.5 m/s m· = 0.12 kg/s h d = 0.04 m L = 0.06 m NTU-Effectiveness () Analysis t = 0.001 m Calculations using Equations (43) and (44) for tm are convenient when inlet and outlet temperatures are known for both fluids. Often, however, the temperatures of fluids leaving the exchanger are unknown. To avoid trial-and-error calculations, the NTU- method uses three dimensionless parameters: effectiveness , number of transfer units (NTU), and capacity rate ratio cr; the mean temperature difference in Equation (44) is not needed. Heat exchanger effectiveness is the ratio of actual heat transfer rate to maximum possible heat transfer rate in a counterflow heat exchanger of infinite surface area with the same mass flow rates and inlet temperatures. The maximum possible heat transfer rate for hot fluid entering at thi and cold fluid entering at tci is Solution: The heat transfer rate is computed using Equations (45) and (46), and exit temperatures from Equation (50). To find the heat transfer rates, UA and are needed. qmax = Cmin(thi – tci) (45) · where Cmin is the smaller of the hot [Ch = (m cp)h] and cold [Cc = (m· cp)h] fluid capacity rates, W/K; Cmax is the larger. The actual heat transfer rate is q = qmax (46) N = number of fins = 16 1 1 1 -------- = -------------------- + -----------UA s hA o hA i where hi = convective heat transfer coefficient on water side ho = gas-side heat transfer coefficient s = surface effectiveness = (Auf + Af)/Ao = fin efficiency Auf = surface area of unfinned surface = L tube (d – Nt) = 0.548 m2 Af = fin surface area = 2LNL tube = 9.6 m2 Ao = Auf + Af = 10.15 m2 Ai = dL tube = 0.628 m2 Step 1. Find hi using Equation (T8.6). Properties of water at an assumed mean temperature of 45°C are = 990.4 kg/m3 cpc = 4181 J/(kg·K) k = 0.6376 W/(m·K) = 5.964 × 10–4 (N·s)/m2 Pr = 3.91 This file is licensed to John Murray (). Publication Date: 6/1/2017 Heat Transfer 4.23 Table 10 Equations for Computing Heat Exchanger Effectiveness, N = NTU Flow Configuration Effectiveness Parallel flow 1 – exp – N 1 + c r ---1 + cr Counterflow 1 – exp – N 1 – c r --1 – c r exp – N 1 – c r cr 1 (T10.2) N ------------1+N cr = 1 (T10.3) 2 -- aN –aN 1 + cr + a 1 + e 1 – e a = Shell-and-tube (one-shell pass, 2, 4, etc. tube passes) Both fluids unmixed Licensed for single user. © 2017 ASHRAE, Inc. Cmax (mixed), Cmin (unmixed) (T10.1) 1 – 1 c r n 1 – 1 cr n ------------------ – 1 ------------------ – c r 1 – 1 1 – 1 Shell-and-tube (n-shell pass, 2n, 4n, etc. tube passes) Cross-flow (single phase) Comments N 0.22 1 – exp -------------- cr 1 – exp – c r ----------------------------------cr Cmax (unmixed), Cmin (mixed) 1 – exp(–/cr) Both fluids mixed N -- Nc –N N 1 – e + cr N 1 – e r – 1 All exchangers with cr = 0 1 – exp(–N) –1 2 1 + cr (T10.4) 1 = effectiveness of one-shell pass shell-and-tube heat exchanger (T10.5) = exp(–cr N 0.78) – 1 (T10.6) = 1 – exp(–N) (T10.7) = 1 – exp(–N cr) (T10.8) (T10.9) (T10.10) Step 2.

Compute fin efficiency and surface effectiveness s . For a rectangular fin with the end of the fin not exposed, tanh mL = ------------------------mL For copper, k = 401 W/(m·K). mL = (2h/kt)1/2L = [(2 115)/(401 0.001)]1/2(0.06) = 1.44 tanh1.44 = ------------------------- = 0.62 1.44 s = (Auf + Af)/A0 = (0.548 + 0.62 × 9.6)/10.15 = 0.64 Step 3. Find heat exchanger effectiveness. For air at an assumed mean temperature of 175°C, cph = 1018 J/(kg·K). C = cm· c = 0.12 1018 = 122.2 W/K h h ph m· c = vc d 2/4 = (990.4 0.5 0.042)/4 = 0.6223 kg/s Cc = m· c cpc = 0.6223 4181 = 2602 W/K cr = Cmin /Cmax = 122.2/2602 = 0.0469 UA = [1/(0.64 × 115 × 10.15) + 1/(2876 0.628)]–1 = 528.5 W/K Fig. 24 Cross Section of Double-Pipe Heat Exchanger in Example 13 v c d 990.4 0.5 0.04 Re = ------------------ = -------------------------------------- = 33 213 –3 4.964 10 fs /2 = [1.58 ln(Re) – 3.28]–2/2 = (1.58 33 213 – 3.28)–2/2 = 0.002 88 0.002 88 33 213 – 1000 3.91 Nu d = --- = 180.4 1/2 2/3 1 + 12.7 0.002 88 3.91 – 1 180.4 0.6376 2 h i = -------------------------------- = 2876 W/(m · K) 0.04 NTU = UA/Cmin = 528.5/122.2 = 4.32 From Equation (T10.2), 1 – exp – N 1 – c r = ---1 – c r exp – N 1 – c r 1 – exp – 4.24 1 – 0.046 96 = --- 0.983 1 – 0.046 96 exp – 4.24 1 – 0.046 96 Step 4. Find heat transfer rate: qmax = Cmin (thi – tci) = 122.2 (200 – 40) = 19 552 W q = qmax = 0.985 19 552 = 19 255 W Step 5. Find exit temperatures: This file is licensed to John Murray (). Publication Date: 6/1/2017 4.24 2017 ASHRAE Handbook—Fundamentals (SI) q 19 255 the = thi – ------ = 200 – ---------------- = 42.4°C Ch 122.2 q 19 255 tce = tci + ------ = 40 + ---------------- = 47.4°C Cc 2602 The mean temperature of hot air now is 43.7°C. The properties of water at this temperature do not vary much from those at the assumed value of 45°C. The only property of air that needs to be updated is cph. For air, at the updated mean temperature of 121°C is 1011 J/(kg·K), which is not very different from the assumed value of 1018 J/(kg·K). Therefore, no further iteration is necessary. Plate Heat Exchangers Licensed for single user. © 2017 ASHRAE, Inc. Plate heat exchangers (PHEs) are used regularly in HVAC&R. The three main types of plate exchangers are plate-and-frame (gasket or semiwelded), compact brazed (CBE), and shell-andplate. The basic plate geometry is shown in Figure 25. Plate Geometry. Different geometric parameters of a plate are defined as follows (Figure 25): • Chevron angle varies between 22 and 65°. This angle also defines the thermal hydraulic softness (low thermal efficiency and pressure drop) and hardness (high thermal efficiency and pressure drop). • Enlargement factor is the ratio of developed length to protracted length. • Mean flow channel gap b is the actual gap available for the flow: b = p – t. • Channel flow area Ax is the actual flow area: Ax = bw. • Channel equivalent diameter de is defined as de = 4Ax /P, where P = 2(b + w) = 2w, because b Bccr , 10 < Bccr < 400, water, 10–500)2] Muley and Manglik (1999) Nu = [0.2668 – 0.006 967(90 –) + 7.244 × (20–78 – 50.94 + 41.162 – 10.513) Re 0.728 – 0.0543 sin[(90 –)/45 + 3.7] Pr 1/3(/w)0.14 f = [2.917 – 0.127 2(90 –) + 0.028 (1.5)] Re –[0.2 + 0.05 77 sin[(90 –)/45] + 2.1} Hayes and Jokar (2009) Nu = CRe P Pr 1/3 (/w)0.14 Re 103, 30 60, 1 1.5. f = ARe –b Water to water C P A b Licensed for single user. © 2017 ASHRAE, Inc. Khan et al. (2010) 60/60 0.134 0.712 1.183 0.095 Dynalene to water 27/60 0.214 0.698 1.559 0.079 Nu = C(*)Re P(*)Pr 0.35(/w)0.14 C(*) = 0.016* + 0.13 P(*) = 0.2* + 0.6 * = chevron angle ratio, (/min) 27/27 0.240 0.724 3.089 0.060 60/60 0.177 0.744 0.570 0 27/60 0.278 0.745 21.405 0.458 27/27 0.561 0.726 3.149 0.078 500 < Re < 2500 and 3.5 < Pr < 6.0 f = ARe b Water to water A b Heavner et al. (1993) Wanniarachchi et al. (1995) 60/60 1.56 –0.24 60/30 1.84 –0.25 30/30 34.43 –0.5 Nu = C11–m Re m Pr 0.5(w)0.17 f = C2 p +1 Re –p C1, C2, m, and p are constants and given as 400 < Re < 10 000, 3.3 < Pr < 5.9, water chevron plate (0° 67°): avg C1 m C2 67/67 67/45 67/0 45/45 45/0 67 56 33.5 22.5 0.089 0.118 0.308 0.195 0.278 0.718 0.720 0.667 0.692 0.683 0.490 0.545 1.441 0.687 1.458 Nu = (Nu13+ Nut 3)1/3 Pr1/3 (w)0.17 Nu1 = 3.65–0.455 0.661 Re 0.339 Nut = 12.6 –1.142 1–m Re m m = 0.646 + 0.0011 f = (f13 + ft3)1/3 f13 = 1774 –1.026 2 Re –1 ft = 46.6 –1.08 1+p Re –p p = 0.004 23 + 0.000 022 32 p 0.1814 0.1555 0.1353 0.1405 0.0838 1 Re 10A, herringbone plates (20° 62, 62° = 62°). Source: Ayub (2003). Passive Techniques Finned-Tube Coils. Heat transfer coefficients for finned coils follow the basic equations of convection, condensation, and evaporation. The fin arrangement affects the values of constants and exponential powers in the equations. It is generally necessary to refer to test data for the exact coefficients. For natural-convection finned coils (gravity coils), approximate coefficients can be obtained by considering the coil to be made of tubular and vertical fins at different temperatures and then applying the natural-convection equation to each. This is difficult because the natural-convection coefficient depends on the temperature difference, which varies at different points on the fin. Fin efficiency should be high (80 to 90%) for optimum naturalconvection heat transfer. A low fin efficiency reduces temperatures near the tip. This reduces t near the tip and also the coefficient h, which in natural convection depends on t. The coefficient of heat transfer also decreases as fin spacing decreases because of interfering convection currents from adjacent fins and reduced free-flow passage; 50 to 100 mm spacing is common. Generally, high coefficients result from large temperature differences and small flow restriction. Edwards and Chaddock (1963) give coefficients for several circular fin-on-tube arrangements, using fin spacing as the characteristic length and in the form Nu = f (Ra, D/), where Do is the fin diameter. Forced-convection finned coils are used extensively in a wide variety of equipment. Fin efficiency for optimum performance is smaller than that for gravity coils because the forced-convection coefficient is almost independent of the temperature difference between surface and fluid. Very low fin efficiencies should be avoided because an inefficient surface gives a high (uneconomical) pressure drop. An efficiency of 70 to 90% is often used. As fin spacing is decreased to obtain a large surface area for heat transfer, the coefficient generally increases because of higher air velocity between fins at the same face velocity and reduced equivalent This file is licensed to John Murray (). Publication Date: 6/1/2017 Licensed for single user. © 2017 ASHRAE, Inc. 4.26 2017 ASHRAE Handbook—Fundamentals (SI) diameter. The limit is reached when the boundary layer formed on one fin surface (see Figure 19) begins to interfere with the boundary layer formed on the adjacent fin surface, resulting in a decrease of the heat transfer coefficient, which may offset the advantage of larger surface area. Selection of fin spacing for forced-convection finned coils usually depends on economic and practical considerations, such as fouling, frost formation, condensate drainage, cost, weight, and volume. Fins for conventional coils generally are spaced 1.8 to 4.2 mm apart, except where factors such as frost formation necessitate wider spacing. There are several ways to obtain higher coefficients with a given air velocity and surface, usually by creating air turbulence, generally with a higher pressure drop: (1) staggered tubes instead of inline tubes for multiple-row coils; (2) artificial additional tubes, or collars or fingers made by forming the fin materials; (3) corrugated fins instead of plane fins; and (4) louvered or interrupted fins. Figure 26 shows data for one-row coils. Thermal resistances plotted include the temperature drop through the fins, based on one square metre of total external surface area. Internal Enhancement. Several examples of tubes with internal roughness or fins are shown in Figure 27. Rough surfaces of the spiral repeated rib variety are widely used to improve in-tube heat transfer with water, as in flooded chillers. Roughness may be produced by spirally indenting the outer wall, forming the inner wall, or inserting coils. Longitudinal or spiral internal fins in tubes can be produced by extrusion or forming and substantially increase surface area. Efficiency of extruded fins can usually be taken as unity (see the section on Fin Efficiency). Twisted strips (vortex flow devices) can be inserted as original equipment or as a retrofit (Manglik and Bergles 2002). From a practical point of view, the twisted tape width should be such that the tape can be easily inserted or removed. Ayub and Al-Fahed (1993) discuss clearance between the twisted tape and tube inside dimension. Microfin tubes (internally finned tubes with about 60 short fins around the circumference) are widely used in refrigerant evaporators (Fig. 26) and condensers. Because gas entering the condenser in vaporcompression refrigeration is superheated, a portion of the condenser that desuperheats the flow is single phase. Some data on single-phase performance of microfin tubes, showing considerably higher heat transfer coefficients than for plain tubes, are available [e.g., Al-Fahed et al. (1993); Khanpara et al. (1986)], but the upper Reynolds numbers of about 10 000 are lower than those found in practice. ASHRAE research [e.g., Eckels (2003)] is addressing this deficiency. The increased friction factor in microfin tubes may not require increased pumping power if the flow rate can be adjusted or the length of the heat exchanger reduced. Nelson and Bergles (1986) discuss performance evaluation criteria, especially for HVAC applications. In fouled chilled-water systems, fouling may, in some cases, seriously reduce the overall heat transfer coefficient U. In general, fouled enhanced tubes perform better than fouled plain tubes, as shown in studies of scaling caused by cooling tower water (Knudsen and Roy 1983) and particulate fouling (Somerscales et al. 1991). A comprehensive review of fouling with enhanced surfaces is presented by Somerscales and Bergles (1997). Fire-tube boilers are frequently fitted with turbulators to improve the turbulent convective heat transfer coefficient (addressing the dominant thermal resistance). Also, because of high gas temperatures, radiation from the convectively heated insert to the tube wall can represent as much as 50% of the total heat transfer. (Note, however, that the magnitude of convective contribution decreases as the radiative contribution increases because of the reduced temperature difference.) Two commercial bent-strip inserts, a twisted-strip insert, and a simple bent-tab insert are depicted in Figure 28. Design equations for convection only are included in Table 12. Beckermann and Goldschmidt (1986) present procedures to include radiation, and Junkhan et al. (1985, 1988) give friction factor data and performance evaluations. Enhanced Surfaces for Gases. Several such surfaces are depicted in Figure 29. The offset strip fin is an example of an interrupted fin that is often found in compact plate fin heat exchangers used for heat Overall Air-Side Thermal Resistance and Pressure Drop for One-Row Coils (Shepherd 1946) Fig. 27 Typical Tube-Side Enhancements This file is licensed to John Murray (). Publication Date: 6/1/2017 Heat Transfer 4.27 Licensed for single user. © 2017 ASHRAE, Inc. recovery from exhaust air. Design equations in Table 12 apply to laminar and transitional flow as well as to turbulent flow, which is a necessary feature because the small hydraulic diameter of these surfaces drives the Reynolds number down. Data for other surfaces (wavy, spine, louvered, etc.) are available in the References. Microchannel Heat Exchangers. Microchannels for heat transfer enhancement are widely used, particularly for compact heat exchangers in automotive, aerospace, fuel cell, and high-flux electronic cooling applications. Bergles (1964) demonstrated the potential of narrow passages for heat transfer enhancement; more recent experimental and numerical work includes Adams et al. (1998), Costa et al. (1985), Kandlikar (2002), Ohadi et al. (2008), Pei et al. (2001), and Rin et al. (2006). Fig. 28 Turbulators for Fire-Tube Boilers Fig. 29 Compared with channels of normal size, microchannels have many advantages. When properly designed, they can offer substantially higher heat transfer rates (because of their greater heat transfer surface area per unit volume and a large surface-to-volume ratio) and reduced pressure drops and pumping power requirements when compared to conventional mini- and macrochannels. Optimum flow delivery to the channels and proper heat transfer surface/channel design is critical to optimum operation of microchannels (Ohadi et al. 2012). This feature allows heat exchangers to be compact and lightweight. Despite their thin walls, microchannels can withstand high operating pressures: for example, a microchannel with a hydraulic diameter of 0.8 mm and a wall thickness of 0.3 mm can easily withstand operating pressures of up to 14 MPa. This feature makes microchannels particularly suitable for use with highpressure refrigerants such as carbon dioxide (CO2). For high-flux electronics (with heat flux at 1 kW/cm2 or higher), microchannels can provide cooling with small temperature gradients (Ohadi et al. 2008). Microchannels have been used for both single-phase and phase-change heat transfer applications. Drawbacks of microchannels include large pressure drop, high cost of manufacture, dirt clogging, and flow maldistribution, especially for two-phase flows. Most of these weaknesses, however, may be solved by optimizing design of the surface and the heat exchanger manifold and feed system. Microchannels are fabricated by a variety of processes, depending on the dimensions and plate material (e.g., metals, plastics, silicon). Conventional machining and electrical discharge machining are two typical options; semiconductor fabrication processes are appropriate for microchannel fabrication in chip-cooling applications. Using microfabrication techniques developed by the electronics industry, three-dimensional structures as small as 0.1 m long are manufactured. Fluid flow and heat transfer in microchannels may be substantially different from those encountered in the conventional tubes. Early research indicates that deviations might be particularly important for microchannels with hydraulic diameters less than 100 m. Recent Progress. The automotive, aerospace, and cryogenic industries have made major progress in compact evaporator development. Thermal duty and energy efficiency have substantially increased, and space constraints have become more important, encouraging greater heat transfer rates per unit volume. The hot side of the evaporators in these applications is generally air, gas, or a condensing vapor. Air-side fin geometry improvements derive from Enhanced Surfaces for Gases This file is licensed to John Murray (). Publication Date: 6/1/2017 4.28 2017 ASHRAE Handbook—Fundamentals (SI) Table 12 Equations for Augmented Forced Convection (Single Phase) Description Equation I. Turbulent in-tube flow of liquids Spiral repeated ribe Comments = 0.21 0.29 – 0.024 7 ha 0.036 + 0.212 p ----- ---Pr ----- = 1 + 2.64 Re --- 90 d d hs 1/7 fa 2.94 w e x p y z --- = 1 + 29.1 Re ------ sin d d 90 fs n w e 0.67 – 0.06(p/d) – 0.49(/90) · Re = GD/, where G = m A x 16/15 15/16 x = 1.37 – 0.157(p/d) y = –1.66 × 10–6 Re – 0.33/90 z = 4.59 + 4.11 × 10–6 Re – 0.15(p/d) k D f s 2 Re Pr h s = --1/2 2/3 1 + 12.7 f s 2 Pr – 1 fs = (1.58 ln Re – 3.28)–2 Licensed for single user. © 2017 ASHRAE, Inc. Finsb hD h 0.4 GD h ------------ = 0.023 Pr ------------ k 0.8 AF ------------ AF i 0.1 A i ----- A 0.5 sec 3 Note that, in computing Re for fins and twisted-strip inserts, there is allowance for reduced cross-sectional area. GD h – 0.2 A F 0.5 0.75 ------- sec f h = 0.046 ------------- ---------- AF i Twisted-strip insertsc hd k ------------ = 1 + 0.769 y hd k y GD hd ------ = 0.023 ------ k y 0.8 Pr 0.4 ------------------- 4 d 0.8 + 2 – 2 d ------------------ 4 d 0.2 – 2 d f = (b / w)n n = 0.18 for liquid heating, 0.30 for liquid cooling 1.75 + 2 – 2 d 1.25 0.0791 1 + 2.752 ---------- f = -- 0.25 – 4 d 1.29 – 4 d GD y II. Turbulent in-tube flow of gases Bent-strip insertsd Twisted-strip insertsd Bent-tab insertsd hD T w --------- ------ k Tb hD T w -------- ------- k Tb hD T w -------- ------- k Tb 0.45 GD 0.6 = 0.258 --------- 0.45 GD 0.65 = 0.122 --------- 0.45 GD 0.54 = 0.406 --------- or hD T w ------ ----- k Tb 0.45 GD 0.63 = 0.208 --------- Respectively, for computing Re and for 4 Figure 28. Note that, in computing Re, there is no allowance for flow blockage of the insert. III. Offset strip fins for plate-fin heat exchangere 1.340 GD –0.5403 –0.1541 0.1499 –0.0678 h = 5 GD 0.504 0.456 – 1.055 1 + 5.269 10 ---------- hb --------- = 0.6522 -----------h- cp G 4.429 GD –0.7422 –0.1856 –0.3053 – 0.2659 – 8 GD 0.920 3.767 0.236 f h = 9.6243 -----------h- 1 + 7.669 10 ---------h- 0.1 0.1 h/cpG, fh, and GDh / are based on the hydraulic mean diameter given by Dh = 4shl/[2(sl + hl + th) + ts] Sources: aRavigururajan and Bergles (1985), bCarnavos (1979), cManglik and Bergles (1993), dJunkhan et al. (1985), eManglik and Bergles (1990). increased heat transfer coefficients and greater surface area densities. To decrease the air-side heat transfer resistance, more aggressive fin designs have been used on the evaporating side, resulting in narrower flow passages. The narrow refrigerant channels with large aspect ratios are brazed in small cross-ribbed sections to improve flow distribution along the width of the channels. Major recent changes in designs involve individual, small-hydraulic-diameter flow passages arranged in multichannel configuration for the evaporating fluid. Figure 30 shows a plate-fin evaporator geometry widely used in compact refrigerant evaporators. The refrigerant-side passages are made from two plates brazed together, and air-side fins are placed between two refrigerant microchannel flow passages. Figure 31 depicts two representative microchannel geometries widely used in the compact heat exchanger industry, with corresponding nominal dimensions provided in Table 13 (Zhao et al. 2000). This file is licensed to John Murray (). Publication Date: 6/1/2017 Heat Transfer 4.29 Plastic heat exchangers have been suggested for HVAC applications (Pescod 1980) and are being manufactured for refrigerated sea water (RSW) applications. These are made of materials impervious to corrosion [e.g., by acidic condensate when cooling a gaseous stream (flue gas heat recovery)], and are easily manufactured with enhanced surfaces. Several companies now offer heat exchangers in plastic, including various enhancements. Licensed for single user. © 2017 ASHRAE, Inc. Active Techniques Unlike passive techniques, active techniques require external power to sustain the enhancement mechanism. Table 14 lists the more common active heat transfer augmentation techniques and the corresponding heat transfer mode believed most applicable to the particular technique. Various active techniques and their world-wide status are listed in Table 15. Except for mechanical aids, which are universally used for selected applications, most other active techniques have found limited commercial applications and are still in development. However, with increasing demand for smart and miniaturized thermal management systems, actively controlled heat transfer augmentation techniques will soon become a necessary requirement for some advanced thermal management systems. All-electric ships, airplanes, and cars use electronics for propulsion, auxiliary systems, sensors, countermeasures, and other system needs. Advances in power electronics and control systems will allow optimized and tactical allocation of total installed power among system components. Table 13 Microchannel Dimensions Triangular 0.86 25 300 1.9 27.12 0.3 Heat Transfer Mode Forced Convection Boil- Evapo- Conden- Mass (Gases) (Liquids) ing ration sation Transfer Technique Mechanical aids Surface vibration Fluid vibration Electrostatic/electrohydrodynamic Suction/injection Jet impingement Rotation Induced flow Table 15 Technique Mechanical aids Fig. 31 Microchannel II Rectangular 0.7 28 300 1.5 28 0.4 Table 14 Active Heat Transfer Augmentation Techniques and Most Relevant Heat Transfer Modes *** = Highly significant — = Not significant Fig. 30 Typical Refrigerant and Air-Side Flow Passages in Compact Automotive Microchannel Heat Exchanger Microchannel I Channel geometry Hydraulic diameter Dh, mm Number of channels Length L, mm Height H, mm Width W, mm Wall thickness, mm NA ** ** *** ** * *** ** * ** ** *** NA ** — *** *** *** ** * NA NA *** NA NA NA *** NA ** *** NA ** *** ** * ** = Significant * = Somewhat significant NA = Not believed to be applicable Worldwide Status of Active Techniques Country or Countries Universally used in selected applications (e.g., fluid mixers, liquid injection jets) Surface vibration Most recent work in United States; not significant Fluid vibration Recent, mostly used for sonic cleaning Electrostatic/electroJapan, United States, United Kingdom; hydrodynamic successful prototypes demonstrated Other electrical methods United Kingdom, France, United States Suction/injection No recent significant developments Jet impingement France, United States; high-temperature units and aerospace applications Rotation United States (industry), United Kingdom (R&D) Induced flow United States; particularly combustion Microchannel Dimensions This file is licensed to John Murray (). Publication Date: 6/1/2017 Licensed for single user. © 2017 ASHRAE, Inc. 4.30 2017 ASHRAE Handbook—Fundamentals (SI) This in turn will require smart (online/on-demand), compact heat exchangers and thermal management systems that can communicate and respond to transient system needs. This section briefly overviews active techniques and recent progress; for additional details, see Ohadi et al. (1996). Mechanical Aids. Augmentation by mechanical aids involves stirring the fluid mechanically. Heat exchangers that use mechanical enhancements are often called mechanically assisted heat exchangers. Stirrers and mixers that scrape the surface are extensively used in chemical processing of highly viscous fluids, such as the blending of a flow of highly viscous plastic with air. Surface scraping can also be applied to duct flow of gases. Hagge and Junkhan (1974) reported tenfold improvement in the heat transfer coefficient for laminar airflow over a flat plate. Table 16 lists selected works on mechanical aids, suction, and injection. Injection. This method involves supplying a gas to a flowing liquid through a porous heat transfer surface or injecting a fluid of a similar type upstream of the heat transfer test section. Injected bubbles produce an agitation similar to that of nucleate boiling. Gose et al. (1957) bubbled gas through sintered or drilled heated surfaces and found that the heat transfer coefficient increased 500% in laminar flow and about 50% in turbulent flow. Tauscher et al. (1970) demonstrated up to a fivefold increase in local heat transfer coefficients by injecting a similar fluid into a turbulent tube flow, but the effect dies out at a length-to-diameter ratio of 10. Practical application of injection appears to be rather limited because of difficulty in cost-effectively supplying and removing the injection fluid. Suction. The suction method involves removing fluid through a porous heated surface, thus reducing heat/mass transfer resistance at the surface. Kinney (1968) and Kinney and Sparrow (1970) reported that applying suction at the surface increased heat transfer coefficients for laminar film and turbulent flows, respectively. Jeng et al. (1995) conducted experiments on a vertical plate channel with asymmetric, isothermal walls. A porous wall segment was embedded in a segment of the test section wall, and enhancement occurred as hot air was sucked from the channel. The local heat transfer coefficient increased with increasing porosity. The maximum heat transfer enhancement obtained was 140%. Table 16 Fluid or Surface Vibration. Fluid or surface vibrations occur naturally in most heat exchangers; however, naturally occurring vibration is rarely factored into thermal design. Vibration equipment is expensive, and power consumption is high. Depending on frequency and amplitude of vibration, forced convection from a wire to air is enhanced by up to 300% (Nesis et al. 1994). Using standing waves in a fluid reduced input power by 75% compared with a fan that provided the same heat transfer rate (Woods 1992). Lower frequencies are preferable because they consume less power and are less harmful to users' hearing. Vibration has not found industrial applications at this stage of development. Rotation. Rotation heat transfer enhancement occurs naturally in rotating electrical machinery, gas turbine blades, and some other equipment. The rotating evaporator, rotating heat pipe, high-performance distillation column, and Rotex absorption cycle heat pump are typical examples of previous work in this area. In rotating evaporators, the rotation effectively distributes liquid on the outer part of the rotating surface. Rotating the heat transfer surface also seems promising for effectively removing condensate and decreasing liquid film thickness. Heat transfer coefficients have been substantially increased by using centrifugal force, which may be several times greater than the gravity force. As shown in Table 17, heat transfer enhancement varies from slight improvement up to 450%, depending on the system and rotation speed. The rotation technique is of particular interest for use in two-phase flows, particularly in boiling and condensation. This technique is not effective in gas-to-gas heat recovery mode in laminar flow, but its application is more likely in turbulent flow. High power consumption, sealing and vibration problems, moving parts, and the expensive equipment required for rotation are some of this technique's drawbacks. Electrohydrodynamics. Electrohydrodynamic (EHD) enhancement of single-phase heat transfer refers to coupling an electric field with the fluid field in a dielectric fluid medium. The net effect is production of secondary motions that destabilize the thermal boundary layer near the heat transfer surface, leading to heat transfer coefficients that are often an order of magnitude higher than those achievable by most conventional enhancement techniques. EHD heat Selected Studies on Mechanical Aids, Suction, and Injection Source Process Heat Transfer Surface Fluid max Valencia et al. (1996) Jeng et al. (1995) Inagaki and Komori (1993) Dhir et al. (1992) Duignan et al. (1993) Son and Dhir (1993) Malhotra and Majumdar (1991) Aksan and Borak (1987) Hagge and Junkhan (1974) Hu and Shen (1996) Natural convection Suction Turbulent natural convection/suction Forced convection/ injection Forced convection/film boiling Forced convection/ injection Water to bed/stirring Pool of water/stirring Forced convection/ scraping Turbulent natural convection Finned tube Asymmetric isothermal wall Vertical plate Tube Horizontal plate Annuli Granular bed Tube coils Cylindrical wall Converging ribbed tube Air Air Air Air Air Water Air Air 0.5 1.4 1 8 1.45 2.0 1.85 3.0 1.7 1.1 0 1.0 ‡ = Enhancement factor (ratio of enhanced to unenhanced heat transfer coefficient) Table 17 Selected Studies on Rotation Source Process Heat Transfer Surface Fluid Prakash and Zerle (1995) Mochizuki et al. (1994) Lan (1991) McElhiney and Preckshot (1977) Nichol and Gacesa (1970) Astarf evi and Balakoter (1970) Tang and McDonald (1971) Marto and Gray (1971) Natural convection Natural convection Solidification External condensation External condensation Boil-ing Nucleate boiling In-tube boiling Ribbed duct Serpentine duct Vertical tube Horizontal tube Vertical cylinder Circular disk Horizontal heated circular cylinder Vertical heated circular cylinder Air Air Water Steam Steam Steam R-113 Water = Enhancement factor (ratio of enhanced to unenhanced heat transfer coefficient) Rotational Speed, rpm Given as a function Given as a function 400 40 2700 2500 1400 2660 max 1.3 3.0 NA 1.7 4.5 3.4 10 gigabit/s and increasing Less per connection, more per data rate 30 000 m Even higher More per connection, less per data rate gaining recognition as building infrastructure, and the standard is being applied to BAS networks as well. ANSI/TIA/EIA Standard 568-B specifies star topology (each device individually cabled to a hub) because connectivity is more robust and management is simpler than for busses and rings. If the wires in a leg are shorted, only that leg fails, making fault isolation easier; with a bus, all drops would fail. The basic structure specified is a backbone, which typically runs from floor to floor in a building and possibly between buildings. Horizontal cabling runs between the distribution frames on each floor and the information outlets in the work areas. Wireless Networks. The rapid maturity of everyday wireless technologies, now widely used for mobile phones, Internet access, and even barcode replacement, has tremendously increased the ability to collect information from the physical world. Wireless technologies offer significant opportunities in sensors and controls for building operation, especially in reducing the cost of installing data acquisition and control devices. Installation costs typically represent 20 to 80% of the total cost of a sensor and control point in any HVAC system, so reducing or eliminating the cost of installation can have a dramatic effect on the overall installed system cost. Low-cost wireless sensors and control systems also make it economical to use more sensors, thereby establishing highly energy-efficient building operations and demand responsiveness that enhance the electric grid reliability. Wireless sensors and control devices that are connected to a network using radio-frequency (RF) or optical (infrared) signals. Devices can communicate bidirectionally (i.e., transmitting and receiving) or one way (transmitting only). Most RF products transmit in the industrial, scientific, or medical frequency bands, which are set aside by the Federal Communication Commission (FCC) for use without an FCC license. Wireless sensor networks have different network topologies, and several types of communication protocols have evolved for them. The simplest is the point-to-point topology, in which two nodes communicate directly with each other. The point-to-multipoint or star topology is an extension of the point-to-point configuration in which many nodes communicate with a central receiving or gateway node. In either topology, sensor nodes might have pure transmitters, which provide one-way communication only, or transceivers, which allow two-way communication and verification of the receipt of messages. Gateways provide a means to convert and pass data between protocols (e.g., from a wireless sensor network protocol to the wired Ethernet protocol). The communication range of the point-to-point and star topologies is limited by the maximum communication range between the sensor node (from which the measured data originates) and the receiver node. This range can be extended by using repeaters, which receive transmissions from sensor nodes and then retransmit them, usually at higher power than the original transmissions. In the mesh network topology, each sensor node includes a transceiver that can communicate directly with any other node within its communication range. These networks connect many devices to many other This file is licensed to John Murray (). Publication Date: 6/1/2017 7.18 2017 ASHRAE Handbook—Fundamentals (SI) devices, thus forming a mesh of nodes in which signals are transmitted between distant nodes via multiple hops. This approach decreases the distance over which each node must communicate and reduces each node's power use substantially, making them more compatible with onboard power sources such as batteries (Capehardt 2005). Table 2 Some Standard Communication Protocols Applicable to BAS 3.4 SPECIFYING BUILDING AUTOMATION SYSTEM NETWORKS Specifying a building automation system includes specifying a platform comprising the following components: field device (e.g., sensors, actuators), controllers (e.g., equipment and/or supervisory), and information management and network communication (e.g., security, diagnostics, maintenance). Many technologies can deliver many performance levels at many different prices. Building automation system design requires assessing the owner's risk tolerance against the proposed project budget. In some cases, new equipment must interface with existing devices, which may limit networking options. ASHRAE Guideline 13-2015 provides detailed information on how to specify a building automation system. Licensed for single user. © 2017 ASHRAE, Inc. Communication Tasks Determining network performance requirements means identifying and quantifying the communication functions required to establish network requirements. Data Exchange. What data passes between which devices? What control and optimization data passes between controllers? What update rates are required? What data does an operator need to reach? How much delay is acceptable in retrieving values? What update rates are required on "live" data displays? (Within one system, answers may vary according to data use.) Which set points and control parameters do operators need to adjust over the network? Alarms and Events. Where do alarms originate? Where are they logged and displayed? How are they acknowledged? What information must be delivered along with the alarm? (Depending on system design, alarm messages may be passed over the network along with the alarms.) Where are alarm summary reports required? How and where do operators need to adjust alarm limits, etc.? Schedules. For HVAC equipment that runs on schedules, where are the schedules be read? Where can they be modified? Trends. Where does trend data originate? Where is it stored? How much will be transmitted? Where is it displayed and processed? Which user interfaces can set and modify trend collection parameters? Network Management. What network diagnostic and maintenance functions are required at which user interfaces? Data access and security functions may be handled as network management tasks. Bushby et al. (1999) refer to the same five communication tasks as interoperability areas and list many more specific considerations in each area. ASHRAE Guideline 13 also provides more detailed information that is helpful. 3.5 APPROACHES TO INTEROPERABILITY Many approaches to interoperability have been proposed and applied, each with varying degrees of success under various circumstances. The field changes quickly as product lines emerge and standards develop and gain acceptance. The building automation world continues to evaluate options project by project. Typically, an interoperable system uses one of two approaches: standard protocols or special-purpose gateways. With a standard, the supplier is responsible for compliance with the standard; the system specifier or integrator is responsible for interoperation. With a Protocol Definition BACnet® ANSI/ASHRAE Standard 135-2012, EN/ ISO Standard 16484-5:2013 ANSI/CEA Standard 709.1 EN 50170:2000 Volume 2 EN 50090 Modbus Application Protocol Specification V1.1 ZigBee® Commercial Building Automation Profile Specification LonTalk PROFIBUS FMS Konnex MODBUS ZigBee® gateway, the supplier takes responsibility for interoperation. Where the job requires interoperation with existing equipment, gateways may be the only solution available. Bushby (1998) addressed this issue and some of the limitations associated with gateways. To date, interoperability by any method requires solid field engineering and careful system integration; the issues extend well beyond the selection of a communication protocol. Standard Protocols Table 2 lists some applicable standard protocols that have been used in BAS. Their different characteristics make some more suited to particular tasks than others. PROFIBUS (www.profibus.com) and MODBUS (www.modbus.org) were designed for low-cost industrial process control and automated manufacturing applications, but they have been applied in BAS. LonTalk defines a LAN technology but not messages that are to be exchanged for BAS applications. BACnet® implementers' agreements, such as those made by members of LonMark International, are necessary to achieve interoperability with LonTalk devices. Konnex evolved from the European Installation Bus (EIB) and several other European protocols developed for residential applications, including multifamily housing. ZigBee® is an open communications standard for wireless devices developed by the ZigBee Alliance. Annex O of the BACnet standard (ANSI/ASHRAE Standard 135-2013) specifies using BACnet messaging with devices described in the ZigBee specification. Martocci (2008) describes how a wireless ZigBee network can be integrated into a BACnet network. BACnet is the only standard protocol developed specifically for building automation applications. BACnet has been adopted as a national standard in the United States, Korea, and Japan, as a European standard, and as a world standard (EN/ISO Standard 16484-5). BACnet was designed to be used with non-BACnet networks. Principles of mapping are discussed in Annex H of ANSI/ASHRAE Standard 135-2012. Gateways and Interfaces Rather than conforming to a published standard, a supplier can design a specific device to exchange data with another specific device. This typically requires cooperation between two manufacturers. In some cases, it can be simpler and more cost-effective than for both manufacturers to conform to an agreed-upon standard. The device can be either custom-designed or off the shelf. In either case, the communication tasks must be carefully specified to ensure that the gateway performs as needed. This is particularly true for a gateway that supports a variety of gateways may be a way to maintain a flexible position as products and standards continue to develop. 4. SPECIFYING BUILDING AUTOMATION SYSTEMS Successful building automation system (BAS) installation depends in part on a clear description (specification) of what is This file is licensed to John Murray (). Publication Date: 6/1/2017 Licensed for single user. © 2017 ASHRAE, Inc. Fundamentals of Control required to meet the customer's needs. The specification should include descriptions of the products desired, or of the performance and features expected. Needed points or data objects should be listed. A control schematic shows the layout of each system to be controlled, including instrumentation and input/output objects and any hard-wired interlocks. Writing a descriptive network specification requires knowledge of the details of network technology. To succeed with any specification, the designer must articulate the end user's needs. Typically, performance-based specification is the best value for the customer (Ehrlich and Pittel 1999). The sequences of operation describe how the system should function and are the designer's primary method of communication to the system programmer. A sequence should be written for each system to be controlled. In writing a sequence, be sure to describe all operational modes and to ensure that all input/output (I/O) devices needed to implement the sequence are shown on the object list and drawings. Annex A of ASHRAE Guideline 13-2015 shows a sample specification outline for a building automation system. Information on specifying building automation systems is in MasterFormat (CSI 2004): Division 23, Section 23 09 00, or in Division 25. Additional information on specifying BAS controls and sample sequences of control for air-handling systems can be found in ASHRAE Guideline 13. 5. COMMISSIONING Commissioning controls can refer either to the proper operation and tuning of a controller or, more broadly, a standard process of quality assurance to ensure that owner's requirements are met, design intent is achieved, and staff is well prepared for operation and maintenance. Because individual pieces of equipment are often tied together into larger systems, and sequences of operation on these systems (affecting safety, indoor air quality, comfort, and energy efficiency) are implemented through controls configuration and programming, building performance is highly dependent on the quality of controls design and implementation. A successful control system requires proper start-up, testing, and documentation, not merely adjustment of a few parameters (set points and throttling ranges) and a few quick checks. Because of the impact on building performance, controls have become a significant focus of the building commissioning process. The typical BAS system should be commissioned directly using an experienced, unbiased third party; this is an effective way to test and document HVAC system performance. The commissioning process requires coordination between the owner, designers, and contractors, and is most effective when it begins before the start of design and continues for the life of the building. Issues are tracked and results are documented throughout the process. Design and construction specifications should include specific commissioning procedures. Review submittals for conformance to design. Check each control device to ensure that it is installed and connected according to approved drawings. Each connection should be verified, and all safeties and sequences tested. Performance assessment should continue after occupancy, especially for large equipment, to identify and address degradation over time. Chapter 43 of the 2015 ASHRAE Handbook—HVAC Applications and ASHRAE Guideline 1 explain more about commissioning. Package controls of high-cost equipment (e.g., chillers, preassembled plants) are not typically checked in field tests of all controls and is particularly important for digital control. First, the controlled process should be controlled manually between various set points to evaluate the following questions: • • • Is the process noisy (rapid fluctuations in controlled variable)? Is there appreciable hysteresis (backlash) in the actuator? How easy (or difficult) is it to maintain and change set point? In which operating region is the process most sensitive (highest gain)? If the process cannot be controlled manually, the reason should be identified and corrected before the controller is tuned. Tuning optimizes control parameters that determine steady-state and transient characteristics of the control system. HVAC processes are nonlinear, and characteristics change seasonally. Controllers tuned under one operating condition may become unstable as conditions change. A well-tuned controller (1) minimizes steady-state error for set point, (2) responds with appropriate timing to disturbances, and (3) remains stable under all operating conditions. Tuning proportional controllers is a compromise between minimizing steady-state error and maintaining margins of stability. Proportional plus integral (PI) control minimizes this compromise because the integral action reduces steady-state error, and the proportional term determines the controller's response to disturbances. As performance requirements have become more stringent, sequences of operation have become increasingly complex, and the task of tuning has also become more challenging. Some manufacturers now provide self-tuning routines to avoid the need for manual adjustment and help maintain performance with changing conditions. Tuning Proportional, PI, and PID Controllers Popular methods of determining proportional, PI, and PID controller tuning parameters include closed- and open-loop process identification methods and trial-and-error methods. For each method, carefully consider the resulting tuning of system responses to avoid compromising safety or reducing the expected life of equipment. Two of the most widely used techniques for tuning these controllers are ultimate oscillation and first-order-plus-dead-time. There are many optimization calculations for these two techniques. The Ziegler-Nichols, which is given here, is well established. Ultimate Oscillation (Closed-Loop) Method. The closed-loop method increases controller gain in proportional-only mode until the equipment continuously cycles after a set-point change (Figure 21, where Kp = 40). Proportional and integral terms are then computed from the cycle's period of oscillation and the Kp value that caused cycling. The ultimate oscillation method is as follows: 1. Adjust control parameters so that all are essentially off. This corresponds to a proportional band (gain) at its maximum (minimum), the integral (repeats per minute) or integral gain to maximum (minimum), and derivative to its minimum. 2. Put controller in automatic. 4. Gradually increase proportional feedback (this corresponds to reducing the proportional band or increasing the proportional gain) until observed oscillations neither grow nor diminish in amplitude. If reduce its minimum, 3. Adjust manual output of the controller to give a measurement as close to midscale as possible. 3. Put controller in automatic. If oscillations are observed, change the set point to either extreme, start over at Step 2 to obtain a stable response. If no oscillations are observed, put the controller back in automatic, and change the set point and try again. This file is licensed to John Murray (). Publication Date: 6/1/2017 7.20 2017 ASHRAE Handbook—Fundamentals (SI) Licensed for single user. © 2017 ASHRAE, Inc. Fig. 21 Response of Discharge Air Temperature to Step Change in Set Points at Various Proportional Constants with No Integral Action Fig. 23 Response of Discharge Air Temperature to Step Change in Set Points at Various Integral Constants with Fixed Proportional Constant 1. Adjust controller manual output to give a midscale measurement. 2. Arrange to record the process variable over time. 3. Move the manual output of the controller by 10% as rapidly as possible to approximate a step change. 4. Record the value of the process variable until it reaches a new steady-state value. 5. Determine dead time and time constant. 6. Use dead time (TD) and time constant (TC) values to calculate PID values as follows: % change in controlled variable Gain = --% change in control signal (10) Proportional-only: PB = Gain/(TC/TD) (12) Ti = 3.33TD minute per repeat (14) Proportional plus integral (PI): PB = 1.8(PBu) percent (4) Ti = 0.83Tu (6) minute per repeat Proportional plus integral plus derivative (PID): PB = 1.67(PBu) percent (7) Ti = 0.50Tu minute per repeat (8) Td = 0.125Tu minute (9) First-Order-plus-Dead-Time (Open-Loop) Method. The openloop method introduces a step change in input into the opened control loop. A graphical technique is used to estimate the process transfer function parameters. Proportional and integral terms are calculated from the process parameters using a series of equations. The value of the process variable must be recorded over time, and the dead time and time constant must be determined from it. This can be accomplished graphically, as is seen in Figure 22. The firstorder-plus-dead-time method is as follows: PB = 9(Gain)(TC/TD) (12) Ti = 3.33(TD) (13) Proportional-integral-derivative (PID): PB = 1.2Gain/(TC/TD) (14) Ti = 2(TD) (15) Td = 0.5(TD) (16) Trial and Error. This method involves adjusting the gain of the proportion-only controller until the desired response to a set point change is observed. Conservative tuning dictates that this response should have a small initial overshoot and quickly damp to steady-state conditions. Set-point changes should be made in the range where controller saturation, or output limit, is avoided. The integral term is then increased until changes in set point produce the same dynamic response as the controller under proportional control, but with the response now centered about the set point (Figure 23). Tuning Digital Controllers In tuning digital controllers, additional parameters must be considered. The controller sampling interval is crucial because it determines how often the system is adjusted. A controller sampling interval of about one-tenth of the controlled-process time constant usually provides adequate control. Many digital control algorithms include an error dead band to eliminate unnecessary control actions when the process is near set This file is licensed to John Murray (). Publication Date: 6/1/2017 Fundamentals of Control point. Hysteresis compensation is possible with digital controllers, but it must be carefully applied because overcompensation can cause continuous cycling of the control loop. Computer Modeling of Control Systems Each component of a control system can be represented by a transfer function, which is an idealized mathematical representation of the relationship between the input and output variables of the component. The transfer function must be sufficiently detailed to cover both the dynamic and static characteristics of the device. The dynamics are represented in the time domain by a differential equation. In environmental control, the transfer function of many of the components can be adequately described by a first-order differential equation, implying that the dynamic behavior is dominated by a single capacitance factor. For a solution, the differential equation is converted to its Laplace or z-transform. For more information on computer modeling programs, see Chapter 40 of the 2015 ASHRAE Handbook—HVAC Applications. Licensed for single user. © 2017 ASHRAE, Inc. 5.2 CODES AND STANDARDS AMCA. 2012. Laboratory methods of testing dampers for rating. ANSI/ AMCA Standard 500-D-12. Air Movement and Control Association, Arlington Heights, IL. ANSI/CEA Standard 709.1-C-2014. Consumer Electronics Association, Arlington, VA. ANSI/CTA. 2015. Free-topology twisted-pair channel specification. ANSI/ CTA Standard 709.3-R2015. Consumer Technology Association, Arlington, VA. ANSI/CTA. 2014. 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Available at opcfoundation.org/about/what-is-opc/. Related Commercial Resources This file is licensed to John Murray (). Publication Date: 6/1/2017 Related Commercial Resources CHAPTER 8 SOUND AND VIBRATION Acoustical Design Objective .... 8.1 Characteristics of Sound.. 8.1 Measuring Sound .... 8.4 Determining Sound Power.. 8.7 Converting from Sound Power to Sound Pressure .... 8.8 Sound Transmission Paths... 8.9 Typical Sources of Sound.. 8.10 Controlling Sound .... System Effects.. Human Response to Sound.. Sound Rating Systems and Acoustical Design Goals.............. Fundamentals of Vibration .... Vibration Measurement Basics .... Symbols .... F FUNDAMENTAL principles of sound and vibration control are applied in the design, installation, and use of HVAC&R systems, suitable levels of noise and vibration can be achieved with a high probability of user acceptance. This chapter introduces these fundamental principles, including basic definitions and terminology, sound and vibration fundamentals, Chapter 48 of the 2015 ASHRAE Handbook—HVAC Applications and the references at the end of this chapter contain technical discussions, tables, and design examples helpful to HVAC designers. Sound Pressure and Sound Pressure Level Licensed for single user. © 2017 ASHRAE, Inc. f 1. ACOUSTICAL DESIGN OBJECTIVE A primary objective in the design of HVAC systems and equipment is to evaluate noise and vibration to ensure that the acoustical environment in a given space is acceptable for various occupant activities. Sound and vibration are created by a source, are transmitted along one or more paths, and reach a receiver. Treatments and modifications can be applied to any or all of these elements to reduce unwanted noise and vibration, although it is usually most effective and least expensive to reduce noise at the source. 2. CHARACTERISTICS OF SOUND Sound is a propagating disturbance in a fluid (gas or liquid) or in a solid. In fluid media, the disturbance travels as a longitudinal compression wave. Sound in air is called airborne sound or just sound. It is generated by a vibrating surface or turbulent fluid stream. In solids, sound can travel as bending, compressional, torsional, shear, or other waves, which, in turn, are sources of airborne sound. Sound in solids is generally called structureborne sound. In HVAC system design, both airborne and structureborne sound propagation are important. Levels The magnitudes of sound and vibration physical properties are almost always expressed in levels. As shown in the following equation, the level L is based on the common (base 10) logarithm of a ratio of the magnitude of a physical property (or its square) to a reference magnitude of the same type: (1) where A L = 10 log -------- A ref (1) where A is the magnitude of the physical property of interest and Aref is the reference value. Note that the ratio is dimensionless. In this equation, a factor of 10 is included to convert bels to decibels (dB). The preparation of this chapter is assigned to TC 2.6, Sound and Vibration. 8.1 Copyright © 2017, ASHRAE 8.11 8.13 8.14 8.15 8.17 8.19 8.19 Waves in air are variations in pressure above and below atmospheric pressure. Sound pressure is measured in pascals (Pa). The human ear responds across a broad range of sound pressures; the threshold of hearing to the threshold of pain covers a range of approximately 1014:1. Table 1 gives approximate values of sound pressure by various sources at specified distances from the source. The range of sound pressure in Table 1 is so large that it is more convenient to use a scale proportional to the logarithm of this quantity. Therefore, the decibel (dB) scale is the preferred method to present sound quantities, not only because it collapses a large range of numbers to a more manageable range, but also because its levels correlate better with human responses to the magnitude of sound than do sound pressures. Equation (1) describes levels of power, intensity, and energy, which are proportional to the square of other physical properties, such as sound pressure and vibration acceleration. Thus, the sound pressure level Lp corresponding to a sound pressure is given by p Lp = 10 log ------- p ref 2 p = 20 log -------- p ref (2) where p is the root mean square (RMS) value of acoustic pressure in pascals. The root mean square is the square root of the time average of the square of the acoustic pressure ratio. The ratio p/pref is squared to give quantities proportional to intensity or energy. A Table 1 Typical Sound Pressures and Sound Pressure Levels Source Military jet takeoff at 30 m Artillery fire at 3 m Passenger jet takeoff at 15 m Loud rock band Automobile horn at 3 m Unmuffled large diesel engine at 40 m Accelerating diesel truck at 15 m Freight train at 30 m Conversational speech at 1 m Window air conditioner at 3 m Quiet residential area Whispered conversation at 2 m Buzzing insect at 1 m Threshold of good hearing Threshold of excellent youthful hearing Sound Pressure, Pressure, Level, dB re 20 Pa 200 63.2 20 6.3 2 0.6 3 140 130 120 110 2 0.6 100 90 0.2 0.06 0.02 0.006 0.002 0.0006 0.0002 0.00006 0.00002 80 70 60 50 40 30 20 10 0 Subjective Reaction Extreme danger Threshold of pain Threshold of discomfort Very loud Loud Moderate Quiet Perceptible Faint Threshold of hearing This file is licensed to John Murray (). Publication Date: 6/1/2017 8.2 2017 ASHRAE Handbook—Fundamentals (SI) reference quantity is needed so that the quantity in parentheses is nondimensional. For sound pressure levels in air, the reference pressure pref is 20 Pa, which corresponds to the approximate threshold of hearing for a young person with good hearing exposed to a pure tone with a frequency of 1000 Hz. The decibel scale is used for many different descriptors relating to sound: source strength, sound level at a specified location, and attenuation along propagation paths; each has a different reference quantity. For this reason, it is important to be aware of the context in which the term decibel or level is used. For most acoustical quantities, there is an internationally accepted reference value. A reference value should always be stated whenever a value of a level is given. Licensed for single user. © 2017 ASHRAE, Inc. Frequency Frequency is the number of oscillations (or cycles) completed per second by a vibrating object. The international unit for frequency is hertz (Hz) with dimension s–1. When the motion of vibrating air particles is simple harmonic, the sound is said to be a pure tone and the sound pressure p as a function of time and frequency can be described by p(t, f) = p0 sin(2ft) (3) where f is frequency in hertz, p0 is the maximum amplitude of oscillating (or acoustic) pressure, and t is time in seconds. The audible frequency range for humans with unimpaired hearing extends from about 20 Hz to 20 kHz. In some cases, infrasound (20 kHz) are important, but methods and instrumentation for these frequency regions are specialized and are not considered here. Speed The speed of a longitudinal wave in a fluid is a function of the fluid's density and bulk modulus of elasticity. In air, at room temperature, the speed of sound is about 340 m/s; in water, about 1500 m/s. In solids, there are several different types of waves, each with a different speed. The speeds of compressional, torsional, and shear waves do not vary with frequency, and are often greater than the speed of sound in air. However, these types of waves are not the primary source of radiated noise because small (compared to the internal displacements. Bending waves, however, are significant sources of radiation, and their speed changes with frequency. At lower frequencies, bending waves are slower than sound in air, but can exceed this value at higher frequencies (e.g., above approximately 1000 Hz). Wavelength The wavelength of sound in a medium is the distance between successive maxima or minima of a simple harmonic disturbance propagating in that medium at a single instant in time. Wavelength, speed, and frequency are related by = c/f (4) where = wavelength, m c = speed of sound, m/s f = frequency, Hz Sound Power and Sound Power Level The sound power of a source is its rate of emission of acoustical energy and is expressed in watts. Sound power depends on operating conditions but not distance of observation location from the Table 2 Examples of Sound Power Outputs and Sound Power Levels Sound Source Sound Power, Level, W dB re 10–12 W Source Large rocket launch (e.g., space shuttle) Jet aircraft at takeoff Large pipe organ Small aircraft engine Large HVAC fan Heavy truck at highway speed Voice, shouting Garbage disposal unit Voice, conversation level Electronic equipment ventilation fan Office air diffuser Small electric clock Voice, soft whisper Rustling leaves Human breath 108 200 104 10 1 0.1 0.01 0.001 10–4 10–5 10–6 10–7 10–8 10–9 10–10 10–11 160 150 130 120 110 100 90 80 70 60 50 40 30 20 10 Note: Sound power levels are computed from exact values of sound power and rounded to nearest dB. source. Sound power levels Lw can often be compared to common sounds that are more familiar. Approximate sound power outputs for common sources are shown in Table 2 with corresponding sound power levels. For sound power level Lw , the power reference is 10–12 W or 1 picowatt. The definition of sound power level is therefore Lw = 10 log(w/10–12) (5) where w is the sound power emitted by the source in watts. (Sound power emitted by a source is not the same as the power consumed by the source. Only a small fraction of the consumed power is converted into sound. For example, a loudspeaker rated at 100 W may be only 1 to 5% efficient, generating only 1 to 5 W of sound power.) Note that sound power level is 10 times the logarithm of the ratio of the power to the reference power, and sound pressure level is 20 times the logarithm of the ratio of the pressure to the reference pressure. Most mechanical equipment is rated in terms of sound power levels so that comparisons can be made using a common reference independent of distance and acoustical conditions in the room. AHRI Standard 370-2011 is a common source for rating large air-cooled outdoor equipment. AMCA Publication 303-79 provides guidelines for using sound power level ratings. Also, AMCA Standards 301-90 and 311-05 provide methods for developing fan sound ratings from laboratory test data. Note, however, some HVAC equipment has sound data available only in terms of sound pressure levels; for example, AHRI Standard 575-2008 is used for watercooled chiller sound rating for indoor applications. In such cases, special care must be taken in predicting the sound pressure level in a specific room (e.g., manufacturer's sound pressure data may be obtained in large spaces nearly free of sound reflection, whereas an HVAC equipment room can often be small and very reverberant). Sound Intensity and Sound Intensity Level The sound intensity I at a point in a specified direction is the rate of flow of sound energy (i.e., power) through unit area at that point. The unit area is perpendicular to the specified direction, and the units of intensity are watts per square metre. Sound intensity level LI is expressed in dB with a reference quantity of 10–12 W/m2; thus, LI = 10 log(I/10–12) (6) The instantaneous intensity I is the product of the pressure and velocity of air motion (e.g., particle velocity), as shown here: I = pv (7) This file is licensed to John Murray (). Publication Date: 6/1/2017 Sound and Vibration 8.3 Both pressure and particle velocity are oscillating, with a magnitude and time variation. Usually, the time-averaged intensity Iave (i.e., the net power flowing through a unit area, often simply called "the intensity") is of interest. Taking the time average of Equation (7) over one period yields Iave = Real{pv} (8) where Real is the real part of the complex (with amplitude and phase) quantity. At locations far from the source and reflecting surfaces, Licensed for single user. © 2017 ASHRAE, Inc. Iave p2/0c (9) where p is the RMS sound pressure, 0 is the density of air (1.2 kg/m3), and c is the acoustic phase speed in air (335 m/s). Equation (9) implies that the relationship between sound intensity and sound pressure varies with air temperature and density. Conveniently, the sound intensity level differs from the sound pressure level by less than 0.5 dB for temperatures and densities normally encountered in HVAC environments. Therefore, sound pressure level at locations far from sources and reflecting surfaces in this chapter that relate sound power level to sound pressure level can be based on the assumption that sound pressure level is equal to sound intensity level. Combining Sound Levels To estimate the levels from multiple sources from the levels from each source, the intensities (not the levels) must be added. Thus, the levels must first be converted to find intensities, the intensities summed, and then converted to a level again, so the combination of multiple levels L1, L2, etc., produces a level Lsum given by L i 10 Lsum = 10 log 10 i (10) where, for sound pressure level, 10 Li /10 is pi 2/p 2ref, and Li is the sound pressure level for the ith source. A simpler and slightly less accurate method is outlined in Table 3. This method, although not exact, results in errors of 1 dB or less. The process with a series of levels may be shortened by combining the largest with the next largest, then combining this sum with the third largest, and so on until the combination of the remaining levels is 10 dB lower than the combined level. The process may then be stopped. The procedures in Table 3 and Equation (10) are valid if the individual sound levels are not highly correlated, which is true for most sounds encountered in HVAC systems. One notable exception is the pure tone. If two or more sound signals contain pure tones at the same frequency, the pressures (amplitude and phase) should be added and the level (20 log) taken of the sum to find the sound pressure level of the two combined tones. The combined sound level is a function of not only the level of each tone (i.e., amplitude of the pressure), but also the phase difference between the tones. Combined sound levels from two tones of equal amplitude and

frequency can range from zero (if the tones are 180° out of phase) up to 6 dB greater than the level of either tone (if the tones are exactly in phase). When two tones of similar amplitude are very close in frequency but not exactly the same, the combined sound level oscillates as the tones move in and out of phase. This effect creates Table 3 Combining Two Sound Levels Difference between levels to be combined, dB Number of decibels to add to highest level to obtain combined level an audible "beating" with a period equal to the inverse of the difference in frequency between the two tones. Measurements of sound levels generated by individual sources are made in the presence of background noise (i.e., noise from sources other than the ones of interest). Thus, the measurement includes noise from the source and background noise. To remove background noise, the levels are unlogged and the square of the background sound pressure subtracted from the square of the sound pressure for the combination of the source and background noise [see Equation (2)]: 0 to 1 2 to 4 10 and 5 to 9 More Lp(source) = 10 log (10 L(comb)/10 – 10 L(bkgd)/10) where L(bkgd) is the sound pressure level of the background noise, measured with the source of interest turned off. If the difference between the levels with the source on and off is greater than 10 dB, then background noise levels are low enough that the effect of background noise on the levels measured with the source on can be ignored. Resonances Acoustic resonances occur in enclosures, such as a room or HVAC plenum, and mechanical resonances occur in structures, such as the natural frequency of vibration of a duct wall. Resonances occur at discrete frequencies where system response to excitation is high. To prevent this, the frequencies at which resonances occur must be known and avoided, particularly by sources of discrete-frequency tones. Avoid aligning the frequency of tonal noise with any frequencies of resonance of the space into which the noise is radiated. At resonance, multiple reflections inside the space form a standing wave pattern (called a mode shape) with nodes at minimum pressure and antinodes at maximum pressure. Spacing between nodes (minimum acoustic pressure) and antinodes (maximum acoustic pressure) is one-quarter of an acoustic wavelength for the frequency of resonance. Absorption and Reflection of Sound Sound incident on a surface, such as a ceiling, is either absorbed, reflected, or transmitted. Absorbed sound is the part of incident sound that is transmitted through the surface and either dissipated (as in acoustic tiles) or transmitted into the adjoining space (as through an intervening partition). The fraction of acoustic intensity incident on the surface that is absorbed is called the absorption coefficient , as defined by the following equation: = Iabs /Iinc 2 1 0 (12) where Iabs is the intensity of absorbed sound and Iinc is the intensity of sound incident on the surface. The absorption coefficient depends on the frequency and angle of incident sound. In frequency bands, the absorption coefficient of nearly randomly incident sound is measured in large reverberant rooms. The difference in the rates at which sound decays after the source is turned off is measured before and after the sample is placed in the reverberant room. The rate at which sound decays is related to the total absorption in the room via the Sabine equation: T60 = 0.161(V/A) (13) where T60 = reverberation time (time required for average sound pressure level in room to decay by 60 dB), s V = volume of room, m3 A = total absorption in room, given by A= Si i i 3 (11) Si = surface area for ith surface, m2 i = absorption coefficient for ith surface This file is licensed to John Murray (). Publication Date: 6/1/2017 8.4 2017 ASHRAE Handbook—Fundamentals (SI) Just as for absorption coefficients, reverberation time varies with frequency. For sound to be incident on surfaces from all directions during absorption measurement, the room must be reverberant so that most of the sound incident on surfaces is reflected and bounced around the room in all directions. In a diffuse sound field, sound is incident on the absorbing sample equally from all directions. The Sabine equation applies only in a diffuse field. Reflected sound superimposes on the incident sound, which increases the level of sound at and near the surfaces (i.e., the sound level near a surface is higher than those away from the surface in the free field). Because the energy in the room is related to the free-field sound pressure levels (see the section on Determining Sound Power for a discussion of free fields) and is often used to relate the sound power emitted into the room and the room's total absorption, it is important that sound pressure level measurements not be made close to reflecting surfaces, where the levels will be higher than in the free field. Measurements should be made at least one-quarter of a wavelength from the nearest reflecting surface (i.e., at a distance of d /4 85/f, where d is in metres and f is frequency in Hz). Licensed for single user. © 2017 ASHRAE, Inc. Room Acoustics The characteristics of sound radiated into a room are affected by surfaces in the room that might absorb, reflect, or transmit sound. The changes of primary concern are the increase in sound levels from those that would exist without the room (i.e., in the open) and reverberation. Lower absorption leads to higher sound pressure levels away from the sources of noise (see the section on Sound Transmission Paths). With lower absorption, reverberation times may be longer. Reverberation can affect perception of music (e.g., in a concert hall) and speech intelligibility (e.g., in a lecture hall). Thus, when adding absorption to reduce a room's background HVACgenerated noise levels, it is important to be aware of the added absorption's effect on reverberation in the room. Acoustic Impedance Acoustic impedance za is the ratio of acoustic pressure p to particle velocity v: za = p/v (14) For a wave propagating in free space far (more than ~1 m) from a source, the acoustic impedance is za 0c (15) where 0 is the density of air (1.2 kg/m3) and c is the sound speed in air (335 m/s). Where acoustic impedance changes abruptly, some of the sound incident at the location of the impedance change is reflected. For example, inside an HVAC duct, the acoustic impedance is different from the free field acoustic impedance, so at the duct termination there is an abrupt change in the acoustic impedance from inside the duct to outside into the room, particularly at low frequencies. Thus, some sound inside the duct is reflected back into the duct (end reflection). Losses from end reflection are discussed in Chapter 48 of the 2015 ASHRAE Handbook—HVAC Applications. 3. MEASURING SOUND Instrumentation The basic instrument for measuring sound is a sound level meter, which comprises a microphone, electronic circuitry, and a display device. The microphone converts sound pressure at a point to an electronic signal, which is then processed and the sound pressure level displayed using analog or digital circuitry. Sound level meters are usually battery-operated, light, handheld units with outputs that vary in complexity depending on cost and level of technology. Time Averaging Most sounds are not constant; pressure fluctuates from moment to moment and the level can vary quickly or slowly. Sound level meters can show time fluctuations of the sound pressure level using specified time constants (slow, fast, impulse), or can hold the maximum or minimum level recorded during some specified interval. All sound level meters perform some kind of time averaging. Some integrating sound level meters take an average of the sound pressure level over a user-definable time, then hold and display the result. The advantage of an integrating meter is that it is easier to read and more repeatable (especially if the measurement period is long). The quantity measured by the integrating sound level meter is the equivalent continuous sound pressure level Leq , which is the level of the time average of the squared pressure: 1 Leq = 10 log --T - dt 2 0 ---------p T 2 p t (16) ref where 1/T T0 dt is the time average (i.e., the sum 0T dt divided by the time over which the sum is taken). Spectra and Analysis Bandwidths Real sounds are much more complex than simple pure tones, where all the energy is at a single frequency. Broadband sound contains energy that usually covers most of the audible frequency range. Sometimes there are multiple, harmonically related tones. All sounds, however, can be represented as levels as a function of frequency using frequency or spectral analysis. A constant-bandwidth analysis expresses a sound's energy content as a spectrum where each data point represents the same spectral width in frequency (e.g., 1 Hz). This is useful when an objectionable sound contains strong tones and the tones' frequencies must be accurately identified before remedial action is taken. A constant-bandwidth spectrum usually contains too much information for typical noise control work or for specifications of acceptable noise levels. Measurements for most HVAC noise control work are usually made with filters that extract the energy in either octave or onethird octave bands. An octave band is a frequency band with an upper frequency limit twice that of its lower frequency limit. Octave and 1/3 octave bands are identified by their respective center frequencies, which are the geometric means of the upper and lower band limits (ANSI Standards S1.6 and S1.11): fc = f upper f lower Three 1/3 octave bands make up an octave band. Table 4 lists the upper, lower, and center frequencies for the preferred series of octave and 1/3 octave bands. For most HVAC sound measurements, filters for the range 20 to 5000 Hz are usually adequate. Although octave band analysis is usually acceptable for rating acoustical environments in rooms, 1/3 octave band analysis is often useful in product development, in assessing transmission losses through partitions, and for remedial investigations. Some sound level meters have standard broadband filters that simulate the frequency response to sound of the average human ear. The A-weighting filter, which simulates the response of the human ear to low levels of sound, is the most common (Figure 1 and Table 5). It deemphasizes the low-frequency portions of a sound spectrum, automatically compensating for the lower sensitivity of the human ear to low-frequency sounds. The C-weighting filter weights the sound less as a function of frequency than the A-weighting, as shown in Figure 1. Because sound levels at low frequencies are attenuated by A-weighting but not by This file is licensed to John Murray (). Publication Date: 6/1/2017 Sound and Vibration 8.5 Table 5 A-Weighting for 1/3 Octave and Octave Bands Licensed for single user. © 2017 ASHRAE, Inc. Fig. 1 Curves Showing A- and C-Weighting Responses for Sound Level Meters Table 4 Midband and Approximate Upper and Lower Cutoff Frequencies for Octave and 1/3 Octave Band Filters Octave Bands, Hz Lower Midband 1/3 Octave Bands, Hz Upper 11.2 16 22.4 22.4 31.5 45 45 63 90 90 125 180 180 250 355 355 500 710 710 1 000 1 400 1 400 2 000 2 800 2 800 4 000 5 600 5 600 8 000 11 200 11 200 16 000 22 400 Lower Midband Upper 11.2 14 18 22.4 28 35.5 45 56 71 90 112 140 180 224 280 355 450 560 710 900 1 120 1 400 1 800 2 240 2 800 3 550 4 500 5 600 7 100 9 000 11 200 14 000 18 000 12.5 16 20 25 31.5 40 50 63 80 100 125 160 200 250 315 400 500 630 800 1 000 1 250 1 600 2 000 2 500 3 150 4 000 5 000 6 300 8 000 10 000 12 500 16 000 20 000 14 18 22.4 28 35.5 45 56 71 90 112 140 180 224 280 355 450 560 710 900 1 200 1 400 1 800 2 240 2 800 3 550 4 500 5 600 7 100 9 000 11 200 14 000 18 000 22 400 C-weighting, these weightings can be used to estimate whether a particular sound has excessive low-frequency energy when a spectrum analyzer is not available. If the difference between C- and Aweighted levels for the sound exceeds about 20 dB, then the sound is likely to be annoying because of excessive low-frequency noise. 1/3 Octave Band Center Frequency, Hz A-Weighting, dB 16 20 25 31.5 40 50 63 80 100 125 160 200 250 315 400 500 630 800 1000 1250 1600 2000 2500 3150 4000 5000 6300 8000 10 000 –56.7 –50.5 –44.7 –39.4 –34.6 –30.2 –26.2 –22.5 –19.1 –16.1 –13.4 –10.9 –8.6 –6.6 –4.8 –3.2 –1.9 –0.8 0 +0.6 +1.0 +1.2 +1.3 +1.2 +1.0 +0.5 –0.1 –1.1 –2.5 Octave Band Center Frequency, Hz A-Weighting, dB 16 –56.7 31.5 –39.4 63 –26.2 125 –16.1 250 –8.6 500 –3.2 1000 0 2000 +1.2 4000 +1.0 8000 –1.1 Table 6 Combining Decibels to Determine Overall Sound Pressure Level Octave Band Frequency, Hz 63 125 250 500 1000 2000 4000 8000 Octave Band Level Lp, dB 85 90 92 87 82 78 65 54 10 L p /10 108 3.2 × 1.0 × 109 1.6 × 109 5.0 × 108 1.6 × 108 6.3 × 107 3.2 × 106 2.5 × 105 = = = = = = = = 0.32 × 109 1.0 × 109 1.6 × 109 0.5 × 109 0.16 × 109 0.06 × 109 0.003 × 109 0.0002 × 109 3.6432 × 109 10 log (3.6 × 109) = 96 dB Note that C-weighting provides some attenuation at very low and very high frequencies: C-weighting is not the same as no weighting (i.e., flat weighting). Sound level meters are available in several accuracy grades specified by ANSI Standard S1.4. A type 1 meter has an accuracy of about ±1.0 dB from 50 to 4000 Hz. The general-purpose type 2 meter, which is less expensive, has a tolerance of about ±1.5 dB from 100 to 1000 Hz, and is adequate for most HVAC sound measurements. Manually selecting filters sequentially to cover the frequency range from 20 to 5000 Hz is time consuming. An instrument that gives all filtered levels simultaneously is called a real-time analyzer (RTA). It speeds up measurement significantly, and most models can save information to an internal or external digital storage device. The process described in Equation (10) for adding a series of levels can be applied to a set of octave or 1/3 octave bands to calculate the overall broadband level (see Table 6 for an example). The This file is licensed to John Murray (). Publication Date: 6/1/2017 8.6 A-weighted sound level may be estimated using octave or 1/3 octave band levels by adding A-weightings given in Table 5 to octave or 1/3 octave band levels before combining the levels. Licensed for single user. © 2017 ASHRAE, Inc. Sound Measurement Basics The sound pressure level in an occupied space can be measured directly with a sound level meter, or estimated from published sound power data after accounting for room volume, distance from the source, and other acoustical factors (see the section on Sound Transmission Paths). Sound level meters measure sound pressure at the microphone location. Estimation techniques calculate sound pressure at a specified point in an occupied space. Measured or estimated sound pressure levels in frequency bands can then be plotted, analyzed, and compared with established criteria for acceptance. Sound measurements must be done carefully to ensure repeatable and accurate results. Note that equipment noise varies significantly with the operation conditions. To make proper comparisons, HVAC unit conditions must be controlled under a reference condition (e.g., full load). Even so, sound levels may not be steady, particularly at low frequencies (250 Hz and lower), and can vary significantly with time. In these cases, both maximum (as measured on a meter with slow response) and average levels (over intervals established by various standards) should be recorded. Other important considerations for sound measurement procedures include • Ambient sound pressure level with HVAC equipment off, with correction factors when HVAC levels are not significantly above ambient • Number of locations for measurements, based on room volume, occupancy, etc. • Duration of time-averaged measurements, statistical meter settings, etc. Sophisticated sound measurements and their procedures should be carried out by individuals experienced in acoustic measurements. At present, there are only a few noise standards that can be used to measure interior sound levels from mechanical equipment (e.g., ASTM Standards E1573 and E1574). Most manuals for sound level meters include sections on how to measure sound, but basic methods that can help obtain acceptable measurements are included here. Determining the sound spectrum in a room or investigating a noise complaint usually requires measuring sound pressure levels in the octave bands from 16 to 8000 Hz. In cases where tonal noise or rumble is the complaint, narrow-band or 1/3 octave band measurements are recommended because of their greater frequency resolution. Whatever the measurement method, remember that sound pressure levels can vary significantly from point to point in a room. In a room, each measurement point often provides a different value for sound pressure level, so the actual location of measurement is very important and must be detailed in the report. A survey could record the location and level of the loudest position, or could establish a few representative locations where occupants are normally situated. In general, the most appropriate height is 1.2 to 1.8 m above the floor. Avoid the exact geometric center of the room and any location within 1 m of a wall, floor, or ceiling. Wherever the location, it must be defined and recorded. If the meter has an integrating-averaging function, use a rotating boom to sample a large area, or slowly walk around the room, and the meter will determine the average sound pressure level for that path. However, take care that no extraneous sounds are generated by microphone movement or by walking; using a windscreen reduces extraneous noise generated by airflow over the moving microphone. Locations with noticeably higher-than-average sound levels should be recorded. See the section on Measurement of Room Sound Pressure Level for more details. When measuring HVAC noise, background noise from other sources (occupants, wind, nearby traffic, elevators, etc.) must be determined. Sometimes the sound from a particular piece of HVAC 2017 ASHRAE Handbook—Fundamentals (SI) Table 7 Guidelines for Determining Equipment Sound Levels in the Presence of Contaminating Background Sound Measurement A minus Measurement B Correction to Measurement A to Obtain Equipment Sound Level 10 dB or more 6 to 9 dB 4 to 5 dB Less than 4 dB 0 dB –1 dB –2 dB Equipment sound level is more than 2 dB below Measurement A Measurement A = Tested equipment plus background sound Measurement B = Background sound alone equipment must be measured in the presence of background sound from sources that cannot be turned off, such as automobile traffic or certain office equipment. Determining the sound level of just the selected equipment requires making two sets of measurements: one with both the HVAC equipment sound and background sound, and another with only the background sound (with HVAC equipment turned off). This situation might also occur, for example, when determining whether noise exposure at the property line from a cooling tower meets a local noise ordinance. The guidelines in Table 7 help determine the sound level of a particular machine in the presence of background sound. Equation (11) in the section on Combining Sound Levels may be used. The uncertainty associated with correcting for background sound depends on the uncertainty of the measuring instrument and the steadiness of the sounds being measured. In favorable circumstances, it might be possible to extend Table 7. In particularly unfavorable circumstances, even values obtained from the table could be substantially in error. Measuring sound emissions from a particular piece of equipment or group of equipment requires a measurement plan specific to the situation. The Air-Conditioning, Heating, and Refrigeration Institute (AHRI); Air Movement and Control Association International (AMCA); American Society of Testing and Materials (ASTM); American National Standards Institute (ANSI); and Acoustical Society of America (ASA) all publish sound level measurement procedures for various laboratory and field sound measurement situations. Outdoor measurements are somewhat easier to make than indoor because there are typically few or no boundary surfaces to affect sound build-up or absorption. Nevertheless, important issues such as the effect of large, nearby sound-reflecting surfaces and weather conditions such as wind, temperature, and precipitation must be addressed. Where measurements are made close to extended surfaces (i.e., flat or nearly flat surfaces with dimensions more than four times the wavelength of the sound of interest), sound pressure levels can be significantly increased. These effects can be estimated through guidelines in many sources such as Harris (1991). Measurement of Room Sound Pressure Level In commissioning building HVAC systems, often a specified room noise criterion must demonstratively be met. Measurement procedures for obtaining the data to demonstrate compliance should also be specified to avoid confusion when different parties make measurements using different procedures. The problem is that most rooms exhibit significant point-to-point variation in sound pressure level. When a noise has no audible tonal components, differences in measured sound pressure level at several locations in a room may be as high as 3 to 5 dB. However, when audible tonal components are present, especially at low frequencies, variations caused by standing waves that occur at frequencies of resonance may exceed 10 dB. These are generally noticeable to the average listener when moving through the room. This file is licensed to John Murray (). Publication Date: 6/1/2017 Sound and Vibration 8.7 Although commissioning procedures usually set precise limits for demonstrating compliance, the outcome can unfortunately be controversial unless the measurement procedure has been specified in detail. In the absence of firm agreement in the industry on an acoustical measurement procedure for commissioning HVAC systems, possibilities include the new ANSI Standard S12.72-2015 on measuring ambient noise levels in a room, as well as AHRI Standard 885, which incorporates a "suggested procedure for field verification of NC/RC levels." Measurement of Acoustic Intensity Equation (8) for the time-averaged intensity (often called simply intensity) requires both the pressure and particle velocity. Pressure is easily measured with a microphone, but there is no simple transducer that converts particle velocity to a measurable electronic signal. Fortunately, particle velocity can be estimated from sound pressures measured at closely spaced (less than ~1/10 of an acoustic wavelength) locations, using Euler's equation: Licensed for single user. © 2017 ASHRAE, Inc. p2 – p1 1 p 1 v = – ---------------- ------ – ---------------- ----------------i2f 0 x i2f 0 x 2 – x 1 (17) where x2 and x1 are the locations of measurements of pressures p2 and p1, f is frequency in Hz, and 0 is density of air. The spatial derivative of pressure (p/x) is approximated with (p/x) = [(p2 – p1)/(x2 – x1)]. Thus, intensity probes typically contain two closely spaced microphones that have nearly identical responses (i.e., are phase matched). Because intensity is a vector, it shows the direction of sound propagation along the line between the microphones, in addition to the magnitude of the sound. Also, because intensity is power/area, it is not sensitive to the acoustic nearfield (see the section on Typical Sources of Sound) or to standing waves where the intensity is zero. Therefore, unlike pressure measurement, intensity measurements can be made in the acoustic nearfield of a source or in the reverberant field in a room to determine the power radiated from the source. However, intensity measurements cannot be used in a diffuse field to determine the acoustic energy in the field, such as used for determining sound power using the reverberation room method. 4. DETERMINING SOUND POWER The sound power of a source cannot be measured directly. Rather, it is calculated from several measurements of sound pressure or sound intensity created by a source in one of several test environments. The following four methods are commonly used. Free-Field Method A free field is a sound field where the effects of any boundaries are negligible over the frequency range of interest. In ideal conditions, there are no boundaries. Free-field conditions can be approximated in rooms with highly sound-absorbing walls, floor, and ceiling (anechoic rooms). In a free field, the sound power of a sound source can be determined from measurements of sound pressure level on an imaginary spherical surface centered on and surrounding the source. This method is based on the fact that, because sound absorption in air can be practically neglected at small distances from the sound source, all of the sound power generated by a source must flow through an imagined sphere with the source at its center. The intensity I of the sound (conventionally expressed in W/m2) is estimated from measured sound pressure levels using the following equation: I = (1 10–12)10Lp /10 (18) where Lp is sound pressure level. The intensity at each point around the source is multiplied by that portion of the area of the imagined sphere associated with the measuring points. Total sound power W is the sum of these products for each point: W= Ii Ai (19) i where Ai is the surface area (in m2) associated with the ith measurement location. ANSI Standard S12.55 describes various methods used to calculate sound power level under free-field conditions. Measurement accuracy is limited at lower frequencies by the difficulty of obtaining room surface treatments with high sound absorption coefficients at low frequencies. For example, a glass fiber wedge structure that gives significant absorption at 70 Hz must be at least 1.2 m long. The relationship between sound power level Lw and sound pressure level Lp for a nondirectional sound source in a free field at distance r in metres can be written as Lw = Lp + 20 log r + 11 (20) For directional sources, use Equation (19) to compute sound power. Often, a completely free field is not available, and measurements must be made in a free field over a reflecting plane. This means that the sound source is placed on a hard floor (in an otherwise soundabsorbing room) or on smooth, flat pavement outdoors. Because the sound is then radiated into a hemisphere rather than a full sphere, the relationship for Lw and Lp for a nondirectional sound source becomes Lw = Lp + 20 log r + 8 (21) A sound source may radiate different amounts of sound power in different directions. A directivity pattern can be established by measuring sound pressure under free-field conditions, either in an anechoic room or over a reflecting plane in a hemianechoic space at several points around the source. The directivity factor Q is the ratio of the squared sound pressure at a given angle from the sound source to the squared sound pressure that would be produced by the same source radiating uniformly in all directions. Q is a function of frequency and direction. The section on Typical Sources of Sound in this chapter and Chapter 48 of the 2015 ASHRAE Handbook— HVAC Applications provide more detailed information on sound source directivity. Reverberation Room Method Another method to determine sound power places the sound source in a reverberation room. AHRI Standard 220 and ANSI Standard S12.58 give standardized methods for determining the sound power of HVAC equipment in reverberation rooms when the sound source contains mostly broadband sound or when tonal sound is prominent. These standards provide a method of qualifying the room to verify that sound power levels for both broadband and tonal noise sources can be accurately determined. Some sound sources that can be measured by these methods are room air conditioners, refrigeration compressors, components of central HVAC systems, and air terminal devices. For ducted equipment, AHRI Standard 261 provides a method of test and AHRI Standards 270 and 370 provide test methods for measuring outdoor equipment. Compressors should be tested according to AHRI Standard 530. ANSI/ASHRAE Standard 130, and ANSI/AHRI Standard 881-2011 establish special measuring procedures for some of these units. AMCA Standard 300 is appropriate for testing fans that are not incorporated into equipment. Two measurement methods may be used in reverberation rooms: direct and substitution. In direct reverberation room measurement, the sound pressure level is measured with the source in the reverberation room at several locations at a distance of at least 1 m from the source and at least one-quarter of a wavelength from the This file is licensed to John Murray (). Publication Date: 6/1/2017 8.8 2017 ASHRAE Handbook—Fundamentals (SI) surfaces of the room. The sound power level is calculated from the average of the sound pressure levels, using the reverberation time and the volume of the reverberation room. The relationship between sound power level and sound pressure level in a reverberation room is given by Lw = Lp + 10 logV – 10 log T60 – 14 (22) where Licensed for single user. © 2017 ASHRAE, Inc. Lp = sound pressure level averaged over room, dB re 20 Pa V = volume of room, m3 T60 = room reverberation time (time required for a 60 dB decay), s The substitution procedure implemented in most ASHRAE, AHRI, and AMCA test standards uses a calibrated reference sound source (RSS). The sound power levels of noise radiated by an RSS are known by calibration using the free-field methodmethod or, in the case of AHRI Standard 250-2013, a hemi-anechoic room method. The most common RSS is a small, vertically shafted direct-drive fan impeller that has no volute housing or scroll. The forwardcurved impeller has a choke plate on its inlet face, causing the fan to operate in a rotating-stall condition that is very noisy. The reference source is designed to have a stable sound power level output from 63 to 8000 Hz and a relatively uniform frequency spectrum in each octave band. Sound pressure level measurements are first made in the reverberant field (far from the RSS or source in question) with only the reference sound source operating in the test room. Then the reference source is turned off and the measurements are repeated with the given source in operation. Because the acoustical environment and measurement locations are the same for both sources, the differences in sound pressure levels measured represent differences in sound power level between the two sources. Using this method, the relationship between sound power level and sound pressure level for the two sources is given by Lw = Lp + (Lw – Lp)ref (23) radiation can be localized using intensity measurements. This procedure can be particularly useful in diagnosing sources of noise during product development. International and U.S. standards that prescribe methods for making sound power measurements with sound intensity probes consisting of two closely spaced microphones include ANSI Standard S12.12 and ISO Standards 9614-1 and 9614-2. In some situations, the sound fields may be so complex that measurements become impractical. A particular concern is that small test rooms or those with somewhat flexible boundaries (e.g., sheet metal or thin drywall) can increase the radiation impedance for the source, which could affect the source's sound power output. Measurement Bandwidths for Sound Power Sound power is normally determined in octave or 1/3 octave bands. Occasionally, more detailed determination of the sound source spectrum is required: narrowband analysis, using either constant fractional bandwidth (1/12 or 1/24 octave) or constant absolute bandwidth (e.g., 1 Hz). The most frequently used analyzer types are digital filter analyzers for constant-percent bandwidth measurements and fast Fourier transform (FFT) analyzers for constant-bandwidth measurements. Narrowband analyses are used to determine the frequencies of pure tones and their harmonics in a sound spectrum. 5. CONVERTING FROM SOUND POWER TO SOUND PRESSURE Designers are often required to use sound power level information of a source to predict the sound pressure level at a given location. Sound pressure at a given location in a room from a source of known sound power level depends on (1) room volume, (2) room furnishings and surface treatments, (3) magnitude of sound source(s), (4) distance from sound source(s) to point of observation, and (5) directivity of source. The classic relationship between a single-point source sound power level and room sound pressure level at some frequency is Lp = Lw + 10 log(Q/4r 2 where Lp = sound pressure level averaged over room, dB re 20 Pa (Lw – Lp)ref = difference between sound power level and sound pressure level of reference sound source Progressive Wave (In-Duct) Method By attaching a fan to one end of a duct, sound energy is confined to a progressive wave field in the duct. Fan sound power can then be determined by measuring the sound pressure level inside the duct. Intensity is then estimated from the sound pressure levels (see the section on the Free-Field Method) and multiplied by the crosssectional area of the duct to find the sound power. The method is described in detail in ASHRAE Standard 68 (AMCA Standard 330) for in-duct testing of fans. This method is not commonly used because of difficulties in constructing the required duct termination and in discriminating between fan noise and flow noise caused by the presence of the microphone in the duct. Sound Intensity Method The average sound power radiated by the source can be determined by measuring the sound intensity over the sphere or hemisphere surrounding a sound source (see the sections on Measurement of Acoustic Intensity and on the Free-Field Method). One advantage of this method is that, with certain limitations, sound intensity (and therefore sound power) measurements can be made in the presence of steady background noise in semireverberant environments and in the acoustic nearfield of sources. Another advantage is that by measuring sound intensity over surfaces that enclose a sound source, sound directivity can be determined. Also, for large sources, areas of + 4/R) (24) where Lp Lw Q r R S = = = = = = = sound pressure level, dB re 20 Pa sound power level, dB re 10–12 W directivity of sound source (dimensionless) distance from source, m room constant, S /(1 –) sum of all surface areas, m2 average absorption coefficient of room surfaces at given frequency, given by Si i Si i i where Si is area of ith surface and i is absorption coefficient for ith surface. If the source is outdoors, far from reflecting surfaces, this relationship simplifies to Lp = Lw + 10 log(Q/4r 2) (25) This relationship does not account for atmospheric absorption, weather effects, and barriers. Note that r 2 is present because the sound pressure in a free field decreases with 1/r 2 (the inverse-square law; see the section on Sound Transmission Paths). Each time the distance from the source is doubled, the sound pressure level decreases by 6 dB. For a simple source centered in a large, flat, reflecting surface, Q may be taken as 2. At the junction of two large flat surfaces, Q is 4; in a corner, Q is 8. In most typical rooms, the presence of acoustically absorbent surfaces and sound-scattering elements (e.g., furniture) creates a This file is licensed to John Murray (). Publication Date: 6/1/2017 Sound and Vibration 8.9 relationship between sound power and sound pressure level that is difficult to predict. For example, hospital rooms, which have only a small amount of absorption, and executive offices, which have substantial absorption, are similar when the comparison is based on the same room volume and distance between the source and point of observation. Using a series of measurements taken in typical rooms, Equation (26) was developed to estimate the sound pressure level at a chosen observation point in a normally furnished room. The estimate is accurate to ±2 dB (Schultz 1985). Lp = Lw – 5 log V – 3 log f – 10 log r + 12 (26) Equation (26) applies to a single sound source in the room itself, not to sources above the ceiling. With more than one source, total sound pressure level at the observation point is obtained by adding the contribution from each source in energy or power-like units, not decibels, and then converting back to sound pressure level [see Equation (10)]. Studies (Warnock 1997, 1998a, 1998b) indicate that sound sources above ceilings may not act as point sources, and Equation (26) may not apply (AHRI Standard 885). Licensed for single user. © 2017 ASHRAE, Inc. 6. SOUND TRANSMISSION PATHS Sound from a source is transmitted along one or more paths to a receiver. Airborne and structureborne transmission paths are both of concern for the HVAC system designer. Sound transmission between rooms occurs along both airborne and structureborne transmission paths. Chapter 48 of the 2015 ASHRAE Handbook—HVAC Applications has additional information on transmission paths. Spreading Losses In a free field, the intensity I of sound radiated from a single source with dimensions that are not large compared to an acoustic wavelength is equal to the power W radiated by the source divided by the surface area A (expressed in m2) over which the power is spread: I = W/A (27) In the absence of reflection, the spherical area over which power spreads is A = 4r 2, so that the intensity is I = W/4r 2 (28) where r is the distance from the source in metres. Taking the level of the intensity (i.e., 10 log) and using Equation (20) to relate intensity to sound pressure levels leads to Lp = Lw – 10 log(4r 2) (29) Lp = Lw – 20 log r – 11 = Lw – 10 log(r 2) – 11 (30) which becomes Thus, the sound pressure level decreases as 10 log(r 2), or 6 dB per doubling of distance. This reduction in sound pressure level of sound radiated into the free field from a single source is called spherical spreading loss. Direct Versus Reverberant Fields Equation (24) relates the sound pressure level Lp in a room at distance r from a source to the sound power level Lw of the source. The first term in the brackets (Q/4r 2) represents sound radiated directly from the source to the receiver, and includes the source's directivity Q and the spreading loss 1/4r 2 from the source to the observation location. The second term in the brackets, 4/R, represents the reverberant field created by multiple reflections from room surfaces. The room constant is Si i i R = ----------------1– (31) where is the spatial average absorption coefficient, Si i i = ---------------- Si (32) i At distances close enough to the source that Q/4r 2 is larger than 4/R, the direct field is dominant and Equation (24) can be approximated by Q Lp = Lw + 10 log -----------2 4r (33) Equation (33) is independent of room absorption R, which indicates that adding absorption to the room will not change the sound pressure level. At distances far enough from the source that Q/4r2 is less than 4/R, Equation (24) can be approximated by 4 Lp = Lw + 10 log --- = Lw – 10 log R + 6 R (34) Adding absorption to the room increases the room constant and thereby reduces the sound pressure level. The reduction in reverberant sound pressure levels associated with adding absorption in the room is approximated by R2 Reduction 10 log ------ R1 (35) where R2 is the room constant for the room with added absorption and R1 is the room constant for the room before absorption is added. The distance from the source where the reverberant field first becomes dominant such that adding absorption to the room is effective is the critical distance rc , obtained by setting Q/4r 2 = 4/R. This leads to rc 0.14 QR (36) where R is in m2 and rc is in m. Airborne Transmission Sound transmits readily through air, both indoors and outdoors. Indoor sound transmission paths include the direct line of sight between the source and receiver, as well as reflected paths introduced by the room's walls, floor, ceiling, and furnishings, which cause multiple sound reflection paths. Outdoors, the effects of the reflections are small, unless the source is located near large reflecting surfaces. However, wind and temperature gradients can cause sound outdoors to refract (bend) and change propagation direction. Without strong wind and temperature gradients and at small distances, sound propagation outdoors follows the inverse square law. Therefore, Equations (20) and (21) can generally be used to calculate the relationship between sound power level and sound pressure level for fully free-field and hemispherical free-field conditions, respectively. Ductborne Transmission Ductwork can provide an effective sound transmission path because the sound is primarily contained within the boundaries of the ductwork and thus suffers only small spreading losses. Sound can transmit both upstream and downstream from the source. A This file is licensed to John Murray (). Publication Date: 6/1/2017 8.10 2017 ASHRAE Handbook—Fundamentals (SI) special case of ductborne transmission is crosstalk, where sound is transmitted from one room to another via the duct path. Where duct geometry changes abruptly (e.g., at elbows, branches, and terminations), the resulting change in the acoustic impedance reflects sound, which increases propagation losses. Chapter 48 of the 2015 ASHRAE Handbook—HVAC Applications has additional information on losses for airborne sound propagation in ducts. Room-to-Room Transmission Room-to-room sound transmission generally involves both airborne and structureborne sound paths. The sound power incident on a room surface element undergoes three processes: (1) some sound energy is reflected from the surface element back into the source room, (2) a portion of the energy is lost through energy transfer into the material comprising the surface element, and (3) the remainder is transmitted through the surface element to the other room. Airborne sound is radiated as the surface element vibrates in the receiving room, and structureborne sound can be transmitted via the studs of a partition or the floor and ceiling surfaces. Licensed for single user. © 2017 ASHRAE, Inc. Structureborne Transmission Solid structures are efficient transmission paths for sound, which frequently originates as a vibration imposed on the transmitting structure. Typically, only a small amount of the input energy is radiated by the structure as airborne sound. With the same force excitation, a low-mass structure with little inherent damping radiates more sound than a massive structure with greater damping. Flanking Transmission Sound from the source room can bypass the primary separating element and get into the receiving room along other paths, called flanking paths. Common sound flanking paths include return air plenums, doors, and windows. Less obvious paths are those along floor and adjoining wall structures. Such flanking paths can reduce sound isolation between rooms. Flanking can explain poor sound isolation between spaces when the partition between them is known to provide very good sound insulation, and how sound can be heard in a location far from the source in a building. Determining whether flanking sound transmission is important and what paths are involved can be difficult. Experience with actual situations and the theoretical aspects of flanking transmission is very helpful. Sound intensity methods may be useful in determining flanking paths. 7. TYPICAL SOURCES OF SOUND Whenever mechanical power is generated or transmitted, a fraction of the power is converted into sound power and radiated into the air. Therefore, virtually any major component of an HVAC system could be considered a sound source (e.g., fans, pumps, ductwork, piping, motors). The component's sound source characteristics depend on its construction, form of mechanical power, and integration with associated system components. The most important source characteristics include total sound power output Lw , frequency distribution, and radiation directivity Q. In addition, a vibrating HVAC system may be relatively quiet but transmit noise to connecting components, such as the unit casing, which may be serious sources of radiated noise. All of these characteristics vary with frequency. Source Strength For airborne noise, source strength should be expressed in terms of sound power levels. For structureborne noise (i.e., vibration), source strengths should be expressed in terms of free vibration levels (measured with the source free from any attachments). Because it is difficult to free a source from all attachments, measurements made with the source on soft mounts, with small mechanical impedances compared to the impedance of the source, can be used to obtain good approximations to free vibration levels. Directivity of Sources Noise radiation from sources can be directional. The larger the source, relative to an acoustic wavelength, the greater the potential of the source to be directional. Small sources tend to be nondirectional. The directivity of a source is expressed by the directivity factor Q as p2 Q = ------------p 2ave (37) where p2() is the squared pressure observed in direction and 2 p ave is the energy average of the squared pressures measured over all directions. Acoustic Nearfield Not all unsteady pressures produced by the vibrating surfaces of a source or directly by disturbances in flow result in radiated sound. Some unsteady pressures "cling" to the surface. Their magnitude decreases rapidly with distance from the source, whereas the magnitude of radiating pressures decreases far less rapidly. The region close to the source where nonradiating unsteady pressures are significant is called the acoustic nearfield. Sound pressure level measurements should not be made in the acoustic nearfield because it is difficult to relate sound pressure levels measured in the nearfield to radiated levels. In general, the nearfield for most sources extends no more than 1 m from the source. However, at lower frequencies and for large sources, sound pressure level measurements should be made more than 1 m from the source when possible. Sound and vibration sources in HVAC systems are so numerous that it is impractical to provide a complete listing here. Major sources include rotating and reciprocating equipment such as compressors, fans, motors, pumps, air-handling units, water-source heat pumps (WSHPs, often used in hotels), rooftop units, and chillers. Noise generation occurs from many mechanisms, including • Vortex shedding, which can be tonal, at the trailing edges of fan blades. Levels of vortex shedding noise increase with velocity of flow vb over the blade proportionate to log(vb). • Turbulence generated upstream of the fan and ingested into the fan. Levels of this broadband noise increase proportionate to log(v0), where v0 is the free-stream velocity of flow into the fan. Turbulence in the boundary layer on the surface of fan blades also causes broadband noise that increases proportionate to log(vb). Flow that separates from blade surfaces can cause low-frequency noise. Nonuniform inflow to fans, created by obstructions, can produce tonal noise at frequencies of blade passage (fb = Nfr, where N is the number of blades and fr is the rotation speed in rev/ s) and integer multiples. Fan imbalance produces vibration at frequencies of shaft rotation and multiples. These low-frequency vibrations can couple to the structures to which the fan is attached, which can transmit the vibration over long distances and radiate low-frequency noise into rooms. • Air and fluid sounds, such as those associated with flow through ductwork, piping systems, grilles, diffusers, terminal boxes, manifolds, and pressure-reducing stations. • Turbulent flow inside ducts, which is a source of broadband noise. Levels increase proportionate to log (v0). Sharp corners of elbows and branches can separate flow from duct walls, producing lowfrequency noise. • Excitation of surfaces (e.g., friction); movement of mechanical linkages; turbulent flow impacts on ducts, plenum panels, and pipes; and impacts within equipment, such as cams and valve slap. Broadband flow noise increases rapidly with flow velocity v This file is licensed to John Murray (). Publication Date: 6/1/2017 Sound and Vibration 8.11 [60 to 80 log(v)], so reducing flow velocities can be very effective in reducing broadband noise. • Magnetostriction (transformer hum), which becomes significant in motor laminations, transformers, switchgear, lighting ballasts, and dimmers. A characteristic of magnetostrictive oscillations is that their fundamental frequency is twice the electrical line frequency (120 Hz in a 60 Hz electrical distribution system.) 8. CONTROLLING SOUND Licensed for single user. © 2017 ASHRAE, Inc. Terminology The following nonninterchangeable terms are used to describe the acoustical performance of many system components. ASTM Standard C634 defines additional terms. Sound attenuation is a general term describing the reduction of the level of sound as it travels from a source to a receiver. Insertion loss (IL) of a silencer or other sound-attenuating element, expressed in dB, is the decrease in sound pressure level or sound intensity level, measured at a fixed receiver location, when the sound-attenuating element is inserted into the path between the source and receiver. For example, if a straight, unlined piece of ductwork were replaced with a duct silencer, the sound level difference at a fixed location would be considered the silencer's insertion loss. Measurements are typically in either octave or 1/3 octave bands. Sound transmission loss (TL) of a partition or other building element is equal to 10 times the logarithm of the ratio of the airborne sound power incident on the partition to the sound power transmitted by the partition and radiated on the other side, in decibels. Measurements are typically in octave or 1/3 octave bands. Chapter 48 of the 2015 ASHRAE Handbook—HVAC Applications defines the special case of breakout transmission loss through duct walls. Noise reduction (NR) is the difference between the space-average sound pressure levels produced in two enclosed spaces or rooms (a receiving room and a source room) by one or more sound sources in the source room. An alternative, non-ASTM definition of NR is the difference in sound pressure levels measured upstream and downstream of a duct silencer or sound-attenuating element. Measurements are typically in octave or 1/3 octave bands. For partitions, NR is related to the transmission loss TL as follows: S NR = TL – 10 log ---- R (38) where S is the partition's surface area and R is the room constant for the receiving room. Note that sound pressure levels measured close to the partition on the receiving side may be higher and should not be included in the space average used to compute the noise reduction. Random-incidence sound absorption coefficient is the fraction of incident sound energy absorbed by a surface exposed to randomly incident sound. It is measured in a reverberation room using 1/3 octave bands of broadband sound (ASTM Standard C423). The sound absorption coefficient of a material in a specific 1/3 octave band depends on the material's thickness, airflow resistivity, stiffness, and method of attachment to the supporting structure. Scattering is the change in direction of sound propagation caused by an obstacle or inhomogeneity in the transmission medium. It results in the incident sound energy being dispersed in many directions. Enclosures and Barriers Enclosing a sound source is a common means of controlling airborne radiation from a source. Enclosure performance is expressed in terms of insertion loss. The mass of the enclosure panels combines with the stiffness (provided by compression) of the air trapped between the source and enclosure panel to produce a resonance. At resonance, the insertion loss may be negative, indicating that radiated noise levels are higher with the enclosure than without it. Therefore, the enclosure design should avoid aligning the enclosure resonance with frequencies commonly radiated from the source at high levels. At low frequencies, insertion loss of enclosures is more sensitive to stiffness of the enclosure panels than to the surface mass density of the panels. At high frequencies, the opposite is true. The insertion loss of an enclosure may be severely compromised by openings or leaks. All penetrations must be sealed. Also, at higher frequencies, adding an enclosure creates a reverberant space between the outer surfaces of the source and the inner surfaces of the enclosure. To avoid build-up of reverberant noise, and thereby noise transmitted through the enclosure, add absorption inside the enclosure. A barrier is a solid element that blocks line-of-sight transmission but does not totally enclose the source or receiver. Properly designed barriers can effectively block sound that propagates directly from the source to the receiver. Barrier performance is expressed in terms of insertion loss: in general, the greater the increase in the path over or around the barrier relative to the direct path between the source and receiver without the barrier, the greater the barrier's insertion losses. Thus, placing the barrier close to the source or receiver is better than midway between the two. The barrier must break the line of sight between the source and receiver to be effective. Insertion losses increase as the barrier extends further above the line of sight. Barriers are only effective in reducing levels for sound propagated directly from the source to the receiver; they do not reduce levels of sound reflected from surfaces in rooms that bypass the barrier. Therefore, barriers are less effective in reverberant spaces than in nonreverberant spaces. Partitions Partitions are typically either single- or double-leaf. Single-leaf partitions are solid homogeneous panels with both faces rigidly connected. Examples are gypsum board, plywood, concrete block, brick, and poured concrete. The transmission loss of a single-leaf partition depends mainly on its surface mass (mass per unit area): the heavier the partition, the less it vibrates in response to sound waves and the less sound it radiates on the side opposite the sound source. Surface mass can be increased by increasing the partition's thickness or its density. The mass law is a semiempirical expression that can predict transmission loss for randomly incident sound for thin, homogeneous single-leaf panels below the critical frequency (discussed later in this section) for the panel. It is written as TL = 20 log(ws f) – 47 (39) where TL = transmission loss ws = surface mass of panel, kg/m2 f = frequency, Hz The mass law predicts that transmission loss increases by 6 dB for each doubling of surface mass or frequency. If sound is incident only perpendicularly on the panel (rarely found in real-world applications), TL is about 5 dB greater than that predicted by Equation (39). Transmission loss also depends on stiffness and internal damping. The transmission losses of three single-leaf walls are shown in Figure 2. For 16 mm gypsum board, TL depends mainly on the surface mass of the wall at frequencies below about 1 kHz; agreement with the mass law is good. At higher frequencies, there is a dip in the TL curve called the coincidence dip because it occurs at the frequency where the wavelength of flexural vibrations in the wall coincides with the wavelength of sound on the panel surface. The lowest frequency where coincidence between the flexural and surface pressure waves can occur is called the critical frequency fc: This file is licensed to John Murray (). Publication Date: 6/1/2017 8.12 2017 ASHRAE Handbook—Fundamentals (SI) c 12 fc = ------ --------2- 2 Eh 2 1/2 (40) where Licensed for single user. © 2017 ASHRAE, Inc. E h c = = = = density of panel material, kg/m3 Young's modulus of panel material, N/m2 thickness of outer panel of partition, m sound speed in air, m/s This equation indicates that increasing the material's stiffness and/ or thickness reduces the critical frequency, and that increasing the material's density increases the critical frequency. For example, the 150 mm concrete slab has a mass of about 370 kg/m2 and has a coincidence frequency at 125 Hz. Thus, over most of the frequency range shown in Figure 2, the transmission loss for the 150 mm concrete slab is well below that predicted by mass law. The coincidence dip for the 25 gage (0.531 mm thick) steel sheet occurs at high frequencies not shown in the figure. The sound transmission class (STC) rating of a partition or assembly is a single number rating often used to classify sound isolation for speech (ASTM Standards E90 and E413). To determine a partition's STC rating, compare transmission losses measured in 1/3 octave bands with center frequencies from 125 to 4000 Hz to the STC contour shown in Figure 3. This contour is moved up until either • The sum of differences between TL values below the contour and the corresponding value on the contour is no more than 32, or • One of the differences between the contour and a TL value is no greater than 8. The STC is then the value on the contour at 500 Hz. As shown in Figure 3, the STC contour deemphasizes transmission losses at low frequencies, so the STC rating should not be used as an indicator of an assembly's ability to control sound that is rich in low frequencies. Most fan sound spectra have dominant low-frequency sound; therefore, to control fan sound, walls and slabs should be selected only on the basis of 1/3 octave or octave band sound transmission loss values, particularly at low frequencies. Note also that sound transmission loss values for ceiling tile are inappropriate for estimating sound reduction between a sound source located in a ceiling plenum and the room below. See AHRI Standard 885 for guidance. Fig. 2 Sound Transmission Loss Spectra for Single Layers of Some Common Materials Walls with identical STC ratings may not provide identical sound insulation at all frequencies. Most single-number rating systems have limited frequency ranges, so designers should select partitions and floors based on their 1/3 octave or octave band sound transmission loss values instead, especially when frequencies below 125 Hz are important. For a given total mass in a wall or floor, much higher values of TL can be obtained by forming a double-leaf construction where each layer is independently or resiliently supported so vibration transmission between them is minimized. As well as mass, TL for such walls depends on cavity depth. Mechanical decoupling of leaves reduces sound transmission through the panel, relative to the transmission that would occur with the leaves structurally connected. However, transmission losses for a double-leaf panel are less than the sum of the transmission losses for each leaf. Air in the cavity couples the two mechanically decoupled leaves. Also, resonances occur inside the cavity between the leaves, thus increasing transmission (decreasing transmission loss) through the partition. Negative effects at resonances can be reduced by adding sound-absorbing material inside the cavity. For further information, see Chapter 48 of the 2015 ASHRAE Handbook—HVAC Applications. Transmission losses of an enclosure may be severely compromised by openings or leaks in the partition. Ducts that lead into or through a noisy space can carry sound to many areas of a building. Designers need to consider this factor when designing duct, piping, and electrical systems. When a partition contains two different constructions (e.g., a partition with a door), the transmission loss TLc of the composite partition may be estimated using the following equation: S1 + S2 TLc = 10 log ---------------------------S1 1 + S2 2 (41) where S1 and S2 are the surface areas of the two types of constructions, and 1 and 2 are the transmissibilities, where = 10–TL/10. For leaks, = 1. For a partition with a transmission of 40 dB, a hole that covers only 1% of the surface area results in a composite transmission loss of 20 dB, a 20 dB reduction in the transmission loss without the hole. This shows the importance of sealing penetrations through partitions to maintain design transmission losses. Fig. 3 Contour for Determining Partition's STC This file is licensed to John Murray (). Publication Date: 6/1/2017 Sound and Vibration Sound Attenuation in Ducts and Plenums Licensed for single user. © 2017 ASHRAE, Inc. Most ductwork, even a sheet metal duct without acoustical lining or silencers, attenuates sound to some degree. The natural attenuation of unlined ductwork is minimal, but can, especially for long runs of rectangular ductwork, significantly reduce ductborne sound. Acoustic lining of ductwork can greatly attenuate sound propagation through ducts, particularly at middle to high frequencies. Chapter 48 of the 2015 ASHRAE Handbook—HVAC Applications has a detailed discussion of lined and unlined ductwork attenuation. If analysis shows that lined ductwork will not reduce sound propagation adequately, commercially available sound attenuators (also known as sound traps or duct silencers) can be used. There are three types: dissipative, reactive, and active. The first two are commonly known as passive attenuators. • Dissipative silencers use absorptive media such as glass or rock fiber as the principal sound-absorption mechanism. Thick, perforated sheet metal baffles filled with low-density fiber insulation restrict the air passage width within the attenuator housing. The fiber is sometimes protected from the airstream by cloths or films. This type of attenuator is most effective in reducing mid- and high-frequency sound energy. • Reactive silencers (sometimes called mufflers) rely on changes in impedance to reflect energy back toward the source and away from the receiver. This attenuator type is typically used in HVAC systems serving hospitals, laboratories, or other areas with strict air quality standards. They are constructed only of metal, both solid and perforated. Chambers of specially designed shapes and sizes behind the perforated metal are tuned as resonators or expansion chambers to react with and reduce sound power at selected frequencies. When designed for a broad frequency range, they are usually not as effective as dissipative attenuators of the same length. However, they can be highly effective and compact if designed for a limited frequency range (e.g., for a pure tone). • Active silencer systems use microphones, loudspeakers, and appropriate electronics to reduce in-duct sound by generating sound 180 out of phase that destructively interferes with the incident sound energy. Microphones sample the sound field in the duct and loudspeakers generate signals with phase opposite to the original noise. Controlled laboratory experiments have shown that active attenuators reduce both broadband and tonal sound, but are typically only effective in the 31.5 through 250 Hz octave bands. Active silencers are more effective for tonal than for broadband noise. Insertion losses of as much as 30 dB have been achieved under controlled conditions. Because the system's microphones and loudspeakers are mounted flush with the duct wall, there is no obstruction to airflow and therefore negligible pressure drop. Because active silencers are not effective in excessively turbulent airflow, their use is limited to relatively long, straight duct sections with an air velocity less than about 7.5 m/s. Silencers are available for fans, cooling towers, air-cooled condensers, compressors, gas turbines, and many other pieces of commercial and industrial equipment. HVAC silencers are normally installed on the intake or discharge side (or both) of a fan or air-handling unit. They may also be used on the receiver side of other noise generators such as terminal boxes, valves, and dampers. Self-noise (i.e., noise generated by airflow through the silencer) can limit an attenuator's effective insertion loss for air velocities over about 10 m/s. Sound power at the silencer outlet is a combination of the power of the noise attenuated by the silencer and the noise generated inside the silencer by flow. Thus, output power WM is related to input power W0 as follows: 8.13 WM = W010–IL/10 + WSG (42) where IL is the insertion loss and WSG is the power of the self-noise. It is also important to determine the dynamic insertion loss at design airflow velocity through the silencer, because a silencer's insertion loss varies with flow velocity. End reflection losses caused by abrupt area changes in duct cross section are sometimes useful in controlling propagation at low frequencies. Low-frequency noise reduction is inversely proportional to the cross-sectional dimension of the duct, with the end reflection effect maximized in smaller cross sections and when the duct length of the smaller cross section is several duct diameters. Note, however, that abrupt area changes can increase flow velocities, which increase broadband high-frequency noise. Where space is available, a lined plenum can provide excellent attenuation across a broad frequency range, especially effective at low frequencies. The combination of end reflections at the plenum's entrance and exit, a large offset between the entrance and exit, and sound-absorbing lining on the plenum walls can result in an effective sound-attenuating device. Chapter 48 of the 2015 ASHRAE Handbook—HVAC Applications has additional information on sound control. Standards for Testing Duct Silencers Attenuators and duct liner materials are tested according to ASTM Standard E477 in North America and ISO Standard 7235 elsewhere. These define acoustic and aerodynamic performance in terms of dynamic insertion loss, self-noise, and airflow pressure drop. Many similarities exist, but the ASTM and ISO standards produce differing results because of variations in loudspeaker location, orientation, duct termination conditions, and computation methods. Currently, no standard test methods are available to measure attenuation by active silencers, although it is easy to measure the effectiveness simply by turning the active silencer control system on and off. Dynamic insertion loss is measured in the presence of both forward and reverse flows. Forward flow occurs when air and sound move in the same direction, as in a supply air or fan discharge system; reverse flow occurs when air and sound travel in opposite directions, as in a return air or fan intake system. 9. SYSTEM EFFECTS The way the HVAC components are assembled into a system affects the sound level generated by the system. Many engineers believe that satisfactory noise levels in occupied spaces can be achieved solely by using a manufacturer's sound ratings as a design tool, without considering the system influence. However, most manufacturers' sound data are obtained under standardized (ideal) laboratory test conditions. In the field, different configurations of connected ductwork, and interactions with other components of the installation, often significantly change the operating noise level. For example, uniform flow into or out of a fan is rare in typical field applications. Nonuniform flow conditions usually increase the noise generated by fans, and are difficult to predict. However, the increases can be large (e.g., approaching 10 dB), so it is desirable to design systems to provide uniform inlet conditions. One method is to avoid locating duct turns near the inlet or discharge of a fan. Furthermore, components such as dampers and silencers installed close to fan equipment can produce nonuniformities in the velocity profile at the entrance to the silencer, which results in a significantly higher-than-anticipated pressure drop across that component. The combination of these two system effects changes the operating point on the fan curve. As a result, airflow is reduced and must be compensated for by increasing fan speed, which may increase noise. Conversely, a well-designed damper or silencer can actually improve flow conditions, which may reduce noise levels. This file is licensed to John Murray (). Publication Date: 6/1/2017 8.14 2017 ASHRAE Handbook—Fundamentals (SI) 10. HUMAN RESPONSE TO SOUND Noise Noise may be defined as any unwanted sound. Sound becomes noise when it Licensed for single user. © 2017 ASHRAE, Inc. • Is too loud: the sound is uncomfortable or makes speech difficult to understand • Is unexpected (e.g., the sound of breaking glass) • Is uncontrolled (e.g., a neighbor's lawn mower) • Happens at the wrong time (e.g., a door slamming in the middle of the night) • Contains unwanted tones (e.g., a whine, whistle, or hum) • Contains unwanted information or is distracting (e.g., an adjacent telephone conversation or undesirable music) • Is unpleasant (e.g., a dripping faucet) • Connotes unpleasant experiences (e.g., a mosquito buzz or a siren wail) • Is any combination of the previous examples To be noise, sound does not have to be loud, just unwanted. In addition to being annoying, loud noise can cause hearing loss, and, depending on other factors, can affect stress level, sleep patterns, and heart rate. To increase privacy, broadband sound may be radiated into a room by an electronic sound-masking system that has a random noise generator, amplifier, and multiple loudspeakers. Noise from such a system can mask low-level intrusive sounds from adjacent spaces. This controlled sound may be referred to as noise, but not in the context of unwanted sound; rather, it is a broadband, neutral sound that is frequently unobtrusive. It is difficult to design airconditioning systems to produce noise that effectively masks lowlevel intrusive sound from adjacent spaces without also being a source of annoyance. Random noise is an oscillation, the instantaneous magnitude of which cannot be specified for any given instant. The instantaneous magnitudes of a random noise are specified only by probability distributions, giving the fraction of the total time that the magnitude, or some sequence of magnitudes, lies within a specified range (ANSI Standard S1.1). There are three types of random noise: white, pink, and red. • White noise has a continuous frequency spectrum with equal energy per hertz over a specified frequency range. Because octave bands double in width for each successive band, for white noise the energy also doubles in each successive octave band. Thus white noise displayed on a 1/3 octave or octave band chart increases in level by 3 dB per octave. • Pink noise has a continuous frequency spectrum with equal energy per constant-percentage bandwidth, such as per octave or 1/3 octave band. Thus pink noise appears on a 1/3 octave or octave band chart as a horizontal line. • Red noise has a continuous frequency spectrum with octave band levels that decrease at a rate of 4 to 5 dB per octave with increasing frequency. Red noise is typical of noise from well-designed HVAC systems. Sound Quality To determine the acoustic acceptability of a space to occupants, sound pressure levels in the space must be known. This, however, is often not sufficient; sound quality is important, too. Factors influencing sound quality include (1) loudness, (2) tone perception, (3) frequency balance, (4) harshness, (5) time and frequency fluctuation, and (6) vibration. People often perceive sounds with tones (such as a whine or hum) as particularly annoying. A tone can cause a relatively lowlevel sound to be perceived as noise. Loudness The primary method for determining subjective estimations of loudness is to present sounds to a sample of listeners under controlled conditions. Listeners compare an unknown sound with a standard sound. (The accepted standard sound is a pure tone of 1000 Hz or a narrow band of random noise centered on 1000 Hz.) Loudness level is expressed in phons, and the loudness level of any sound in phons is equal to the sound pressure level in decibels of a standard sound deemed to be equally loud. Thus, a sound that is judged as loud as a 40 dB, 1000 Hz tone has a loudness level of 40 phons. Average reactions of humans to tones are shown in Figure 4 (Robinson and Dadson 1956). The reaction changes when the sound is a band of random noise (Pollack 1952), rather than a pure tone (Figure 5). The figures indicate that people are most sensitive in the midfrequency range. The contours in Figure 4 are closer together at low frequencies, showing that at lower frequencies, people are less sensitive to sound level, but are more sensitive to changes in level. Under carefully controlled experimental conditions, humans can detect small changes in sound level. However, for humans to describe a sound as being half or twice as loud requires changes in the overall sound pressure level of about 10 dB. For many people, a 3 dB change is the minimum perceptible difference. This means that halving the power output of the source causes a barely noticeable change in sound pressure level, and power output must be reduced by a factor of 10 before humans determine that loudness has been halved. Table 8 summarizes the effect of changes in sound levels for simple sounds in the frequency range of 250 Hz and higher. The phon scale covers the large dynamic range of the ear, but does not fit a subjective linear loudness scale. Over most of the Predicting Human Response to Sound Predicting the response of people to any given sound is, at best, only a statistical concept, and, at worst, very inaccurate. This is because response to sound is not only physiological but psychological and depends on the varying attitude of the listener. Hence, the effect of sound is often unpredictable. However, people respond adversely if the sound is considered too loud for

the situation or if it sounds “wrong.” Therefore, criteria are based on descriptors that account for level and spectrum shape. Fig. 4 Free-Field Equal Loudness Contours for Pure Tones (Robinson and Dadson 1956) © IOP Publishing. Reproduced with permission. All rights reserved. This file is licensed to John Murray (). Publication Date: 6/1/2017 Sound and Vibration 8.15 Licensed for single user. © 2017 ASHRAE, Inc. Fig. 6 Frequencies at Which Various Types of Mechanical and Electrical Equipment Generally Control Sound Spectra Fig. 5 Equal Loudness Contours for Relatively Narrow Bands of Random Noise (Reprinted with permission from I. Pollack, Journal of the Acoustical Society of America, vol. 24, p. 533, 1952. Copyright 1952, Acoustical Society of America.) Table 8 Subjective Effect of Changes in Sound Pressure Level, Broadband Sounds (Frequency 250 Hz) Subjective Change Objective Change in Sound Level (Approximate) Much louder Twice as loud Louder Just perceptibly louder Just perceptibly quieter Quieter Half as loud Much quieter More than +10 dB +10 dB +5 dB +3 dB –3 dB –5 dB –10 dB Less than –10 dB audible range, a doubling of loudness corresponds to a change of approximately 10 phons. To obtain a quantity proportional to the loudness sensation, use a loudness scale based on the sone. One sone equals the loudness level of 40 phons. A rating of two sones corresponds to 50 phons, and so on. In HVAC, only the ventilation fan industry (e.g., bathroom exhaust and sidewall propeller fans) uses loudness ratings. Standard objective methods for calculating loudness have been developed. ANSI Standard S3.4 calculates loudness or loudness level using 1/3 octave band sound pressure level data as a starting point. The loudness index for each 1/3 octave band is obtained from a graph or by calculation. Total loudness is then calculated by combining the loudnesses for each band according to a formula given in the standard. A graphic method using 1/3 octave band sound pressure levels to predict loudness of sound spectra containing tones is presented in Zwicker (ISO Standard 532) and German Standard DIN 45631. Because of its complexity, loudness has not been widely used in engineering practice in the past. Acceptable Frequency Spectrum The most acceptable frequency spectrum for HVAC sound is a balanced or neutral spectrum in which octave band levels decrease at a rate of 4 to 5 dB per octave with increasing frequency. This means that it is not too hissy (excessive high-frequency content) or too rum- bly (excessive low-frequency content). Unfortunately, achieving a balanced sound spectrum is not always easy: there may be numerous sound sources to consider. As a design guide, Figure 6 shows the more common mechanical and electrical sound sources and frequency regions that control the indoor sound spectrum. Chapter 48 of the 2015 ASHRAE Handbook—HVAC Applications provides more detailed information on treating some of these sound sources. 11. SOUND RATING SYSTEMS AND ACOUSTICAL DESIGN GOALS The degree of occupant satisfaction with the background noise level in any architectural space depends on the sound quality of the noise itself, the occupant’s aural sensitivity, and specific task engagement. In most cases, background noise must be unobtrusive, meaning that the noise level must not be excessive enough to cause distraction or annoyance, or to interfere with, for example, music listening and speech intelligibility. In addition, the frequency content and temporal variations must not call attention to the noise intrusion, but rather present a bland and unobtrusive background. For critical listening conditions such as for music in a symphony hall or speech in grade schools, background noise must not exceed a relatively low exposure level. However, for speech and music in a high school gymnasium, a significantly higher background noise level will be tolerated. When low annoyance and distractions are a key factor, such as in open-plan offices for occupant productivity, a minimum acceptable background noise must be considered to effectively cover undesirable intruding sounds. Consequently, HVAC system sound control goals vary depending on the required use of the space. To be unobtrusive, HVAC-related background noise should have the following properties: • Frequency content that is broadband and smooth in nature, and at a level suitable for the use of the space • No audible tones or other characteristics such as roar, whistle, hum, or rumble • No significant time fluctuations in level or frequency such as throbbing or pulsing Unfortunately, there is no standard process to easily characterize the effects of audible tones and level fluctuations, so currently available rating methods do not adequately address these issues. Conventional approaches for rating sound in an occupied space include the following. This file is licensed to John Murray (). Publication Date: 6/1/2017 8.16 Licensed for single user. © 2017 ASHRAE, Inc. A-Weighted Sound Level (dBA) The A-weighted sound level LA is an easy-to-determine, singlenumber rating, expressed as a number followed by dBA (e.g., 40 dBA). A-weighted sound levels correlate well with human judgments of relative loudness, but do not indicate degree of spectral balance. Thus, they do not necessarily correlate well with the annoyance caused by the noise. Many different-sounding spectra can have the same numeric rating but quite different subjective qualities. A-weighted comparisons are best used with sounds that sound alike but differ in level. They should not be used to compare sounds with distinctly different spectral characteristics; two sounds at the same sound level but with different spectral content are likely to be judged differently by the listener in terms of acceptability as a background sound. One of the sounds might be completely acceptable; the other could be objectionable because its spectrum shape was rumbly, hissy, or tonal in character. A-weighted sound levels are used extensively in outdoor environmental noise standards and for estimating the risk of damage to hearing for long-term exposures to noise, such as in industrial environments and other workplaces. In outdoor environmental noise standards, the principal sources of noise are vehicular traffic and aircraft, for which A-weighted criteria of acceptability have been developed empirically. Outdoor HVAC equipment can create significant sound levels that affect nearby properties and buildings. Local noise ordinances often limit property line A-weighted sound levels and typically are more restrictive during nighttime hours. Noise Criteria (NC) Method The NC method remains the predominant design criterion used by HVAC engineers. This single-number rating is somewhat sensitive to the relative loudness and speech interference properties of a given sound spectrum. Its wide use derives in part from its ease of use and its publication in HVAC design textbooks. The method consists of a family of criterion curves now extending from 16 to 8000 Hz and a rating procedure based on speech interference levels (ANSI Standard S12.2-2008). The criterion curves define the limits of octave band spectra that must not be exceeded to meet acceptance in certain spaces. The NC curves shown in Figure 7 are in steps of 5 dB. NC-rating procedures for measured data use interpolation, rounded to the nearest dB. The rating is expressed as NC followed by a number. For example, the spectrum shown is rated NC 43 because this is the lowest rating curve that falls entirely above the measured data. An NC 35 design goal is common for private offices. The background sound level meets this goal if no portion of its spectrum lies above the designated NC 35 curve. The NC method is sensitive to level but has the disadvantage as a design criterion method that it does not require the sound spectrum to approximate the shape of the NC curves. Thus, many different sounds can have the same numeric rating, but rank differently on the basis of subjective sound quality. In many HVAC systems that do not produce excessive low-frequency sound, the NC rating correlates relatively well with occupant satisfaction if sound quality is not a significant concern or if the octave band levels have a shape similar to the nearest NC curves. Two problems occur in using the NC procedure as a diagnostic tool. First, when the NC level is determined by a prominent peak in the spectrum, the actual level of resulting background sound may be quieter than that desired for masking unwanted speech and activity sounds, because the spectrum on either side of the tangent peak drops off too rapidly. Second, when the measured spectrum does not match the shape of the NC curve, the resulting sound might be rumbly (levels at low frequencies determine the NC rating and levels at high frequencies roll off faster than the NC curve) or hissy (the NC 2017 ASHRAE Handbook—Fundamentals (SI) rating is determined by levels at high frequencies but levels at low frequencies are much less than the NC curve for the rating). Manufacturers of terminal units and diffusers commonly use NC ratings in their published product data. Because of the numerous assumptions made to arrive at these published values (e.g., size of room, type of ceiling, number of units), relying solely on NC ratings to select terminal units and diffusers is not recommended. Room Criterion (RC) Method The room criterion (RC) method (ANSI Standard S12.2; Blazier 1981a, 1981b) is based on measured levels of HVAC noise in spaces and is used primarily as a diagnostic tool. The RC method consists of a family of criteria curves and a rating procedure. The shape of these curves differs from the NC curves to approximate a wellbalanced, neutral-sounding spectrum; two additional octave bands (16 and 31.5 Hz) are added to deal with low-frequency sound, and the 8000 Hz octave band is dropped. This rating procedure assesses background sound in spaces based on its effect on speech communication, and on estimates of subjective sound quality. The rating is expressed as RC followed by a number to show the level of the sound and a letter to indicate the quality [e.g., RC 35(N), where N denotes neutral]. For a full explanation of RC curves and analysis procedures, see Chapter 48 of the 2015 ASHRAE Handbook—HVAC Applications. Criteria Selection Guidelines In general, these basic guidelines are important: • Sound levels below NC or RC 35 are generally not detrimental to good speech intelligibility. Sound levels at or above these levels may interfere with or mask speech. • Even if the occupancy sound is significantly higher than the anticipated background sound level generated by mechanical equipment, the sound design goal should not necessarily be raised to levels approaching the occupancy sound. This avoids occupants Fig. 7 NC (Noise Criteria) Curves and Sample Spectrum (Curve with Symbols) This file is licensed to John Murray (). Publication Date: 6/1/2017 Sound and Vibration 8.17 having to raise their voices uncomfortably to be heard over the noise. For full details and recommended background sound level criteria for different spaces, see Chapter 48 of the 2015 ASHRAE Handbook—HVAC Applications. 12. FUNDAMENTALS OF VIBRATION A rigidly mounted machine transmits its internal vibratory forces directly to the supporting structure. However, by inserting resilient mountings (vibration isolators) between the machine and supporting structure, the magnitude of transmitted force can be dramatically reduced. Vibration isolators can also be used to protect sensitive equipment from floor vibration. Licensed for single user. © 2017 ASHRAE, Inc. Single-Degree-of-Freedom Model The simplest representation of a vibration isolation system is the single-degree-of-freedom model, shown in Figure 8. Only motion along the vertical axis is considered. The isolated system is represented by a mass and the isolator is represented by a spring, which is considered fixed to ground. Excitation (i.e., the vibratory forces generated by the isolated equipment, such as shaft imbalance in rotating machinery) is applied to the mass. This simple model is the basis for catalog information provided by most manufacturers of vibration isolation hardware. Mechanical Impedance Mechanical impedance Zm is a structural property useful in understanding the performance of vibration isolators in a given installation. Zm is the ratio of the force F applied to the structure divided by the velocity v of the structure’s vibration response at the point of excitation: Zm = F/v (43) At low frequencies, the mechanical impedance of a vibration isolator is approximately equal to k/2f, where k is the stiffness of the isolator (force per unit deflection) and f is frequency in Hz (cycles per second). Note that the impedance of the isolator is inversely proportional to frequency. This characteristic is the basis for an isolator’s ability to block vibration from the supported structure. In the simple single-degree-of-freedom model, impedance of the isolated mass is proportional to frequency. Thus, as frequency increases, the isolator increasingly provides an impedance mismatch between the isolated structure and ground. This mismatch attenuates the forces imposed on the ground. However, at the system’s particular natural frequency (discussed in the following section), the effects of the isolator are decidedly detrimental. Fig. 8 Single-Degree-of-Freedom System Natural Frequency Using the single-degree-of-freedom model, the frequency at which the magnitude of the spring and mass impedances are equal is the natural frequency fn. At this frequency, the mass’s vibration response to the applied excitation is a maximum, and the isolator actually amplifies the force transmitted to ground. The natural frequency of the system (also called the isolation system resonance) is given approximately by 1 k fn = ------ ----2 M (44) where M is the mass of the equipment supported by the isolator. The stiffness k is expressed as N/m, and M as kg. This equation simplifies to 15.8 fn = ----------- st (45) where st is the isolator static deflection (the incremental distance the isolator spring compresses under the weight of the supported equipment) in millimetres. Thus, to achieve the appropriate system natural frequency for a given application, it is customary to specify the corresponding isolator static deflection and the load to be supported at each of the mounting points. The transmissibility T of this system is the ratio of the amplitudes of the force transmitted to the building structure to the exciting force produced inside the vibrating equipment. For disturbing frequency fd , T is given by 1 T = ---------------------------2 1 – fd fn (46) The transmissibility equation is plotted in Figure 9. It is important to note that T is inversely proportional to the square of the ratio of the disturbing frequency fd to the system natural frequency fn. At fd = fn, resonance occurs: the denominator of Equation (46) equals zero and transmission of vibration is theoretically infinite. In practice, transmissibility at resonance is limited by damping in the system, which is always present to some degree. Thus, the magnitude of vibration amplification at resonance always has a finite, though often dramatically high, value. Note that vibration isolation (attenuation of force applied to ground) does not occur until the ratio of the disturbing frequency fd Fig. 9 Vibration Transmissibility T as Function of fd / fn This file is licensed to John Murray (). Publication Date: 6/1/2017 8.18 2017 ASHRAE Handbook—Fundamentals (SI) to the system natural frequency fn is greater than 1.4. Above this ratio, vibration transmissibility decreases (attenuation increases) with the square of frequency. In designing isolators, it is customary to specify a frequency ratio of at least 3.5, which corresponds to an isolation efficiency of about 90%, or 10% transmissibility. Higher ratios may be specified, but in practice this often does not greatly increase isolation efficiency, especially at frequencies above about 10 times the natural frequency. The reason is that wave effects and other nonlinear characteristics in real isolators cause a deviation from the theoretical curve that limits performance at higher frequencies. To obtain the design objective of fd /fn 3.5, the lowest frequency of excitation fd is determined first. This is usually the shaft rotation rate in hertz (Hz; cycles/second). Because it is usually not possible to change the mass of the isolated equipment, the combined stiffness of the isolators is then selected such that Licensed for single user. © 2017 ASHRAE, Inc. k = (2fd /3.5) 2M (47) where M is the mass of the mounted equipment in kg, and k is in N/m. With four mounts, the stiffness of each mount is k/4, assuming equal mass distribution. For a given set of isolators, as shown by Equations (44) and (46), if equipment mass is increased, the resonance frequency decreases and isolation increases. In practice, the load-carrying capacity of isolators usually requires that their stiffness or their number be increased. Consequently, the static deflection and transmissibility may remain unchanged. For example, as shown in Figure 10, a 500 kg piece of equipment installed on isolators with stiffness k of 196 kN/m results in a 25 mm deflection and a system resonance frequency fn of 3.15 Hz. If the equipment operates at 564 rpm (9.4 Hz) and develops an internal force of 5000 N, 635 N is transmitted to the structure. If the total mass is increased to 5000 kg by placing the equipment on a concrete inertia base and the stiffness of the springs is increased to 1960 kN/m, the deflection is still 25 mm, the resonance frequency of the system is maintained at 3.15 Hz, and the force transmitted to the structure remains at 635 N. The increased mass, however, reduces equipment displacement. The forces F generated inside the mounted equipment, which do not change when mass is added to the equipment, now must excite more mass with the same internal force. Therefore, because F = Ma, where a is acceleration, the maximum dynamic displacement of the mounted equipment is reduced by a factor of M1/M2, where M1 and M2 are the masses before and after mass is added, respectively. structural floor, particularly on the roof of a building, significantly softer vibration isolators are usually required than in the on-grade or basement case. This is because the impedance of the supporting structure can no longer be ignored. For the two-degrees-of-freedom system in Figure 11, mass M1 and isolator K1 represent the supported equipment, and M2 and K2 represent the effective mass and stiffness of the floor structure. In this case, transmissibility refers to the vibratory force imposed on the floor, and is given by 1 T = ---2 fd 1 1 – ------ – ---------------------------------2 f n1 f n1 k 2 M 2 ------ ----- – ------ fd k1 M1 As in Equation (46), fd is the forcing frequency. Frequency fn1 is the natural frequency of the isolated equipment with a rigid foundation [Equation (44)]. The implication of Equation (48) relative to Equation (46) is that a nonrigid foundation can severely alter the effectiveness of the isolation system. Figure 12 shows transmissibility of a floor structure with twice the stiffness of the isolator, and a floor effective mass half that of the isolated equipment. Comparing Figure 12 to Figure 9 shows that the nonrigid floor introduces a second resonance well above that of the isolation system assuming a rigid floor. Unless care is taken in the isolation system design, this secondary amplification can cause a serious sound or vibration problem. As a general rule, it is advisable to design the system such that the static deflection of the isolator, under the applied equipment weight, is on the order of 10 times the incremental static deflection of the floor caused by the equipment mass (Figure 13). Above the rigid-foundation natural frequency fn1, transmissibility is comparable to that of the simple single-degree-of-freedom model. Other complicating factors exist in actual installations, which often depart from the two-degrees-of-freedom model. These include the effects of horizontal and rotational vibration. Given these complexities, it is often beneficial to collaborate with an experienced acoustical consultant or structural engineer when designing vibration isolation systems applied to flexible floor structures. 13. VIBRATION MEASUREMENT BASICS Control of HVAC system sound and vibration are of equal importance, but measurement of vibration is often not necessary to determine sources or transmission paths of disturbing sound. Practical Application for Nonrigid Foundations The single-degree-of-freedom model is valid only when the impedance of the supporting structure (ground) is high relative to the impedance of the vibration isolator. This condition is usually satisfied for mechanical equipment in on-grade or basement locations. However, when heavy mechanical equipment is installed on a Fig. 10 Effect of Mass on Transmissibility (48) Fig. 11 Two-Degrees-of-Freedom System This file is licensed to John Murray (). Publication Date: 6/1/2017 Sound and Vibration 8.19 The simplest measure is the overall signal level as a function of time. This is analogous to the unfiltered sound pressure level. If a detailed frequency analysis is needed, there is a choice of filters similar to those available for sound measurements: octave band, 1/3 octave band, or 1/12 octave band. In addition, there are narrowband analyzers that use the fast Fourier transform (FFT) to analyze and filter a signal. Though widely used, they should only be used by a specialist for accurate results. The most important issues in vibration measurement include (1) choosing a transducer with a frequency range appropriate to the measurement, (2) properly mounting the transducer to ensure that the frequency response claimed is achieved, and (3) properly calibrating the vibration measurement system for the frequency range of interest. For more thorough descriptions of specialized vibration measurement and analysis methods, designers should consult other sources [e.g., Harris (1991)]. Licensed for single user. © 2017 ASHRAE, Inc. Fig. 12 Transmissibility T as Function of fd /fn1 with k2 /k1 = 2 and M2 /M1 = 0.5 14. Fig. 13 Transmissibility T as Function of fd /fn1 with k2/k1 = 10 and M2/M1 = 40 The typical vibrations measured are periodic motions of a surface. This surface displacement oscillates at one or more frequencies produced by mechanical equipment (e.g., rotating shafts or gears), thermal processes (e.g., combustion), or fluid-dynamic means (e.g., airflow through a duct or fan interactions with air). A transducer detects displacement, velocity, or acceleration of a surface and converts the motion to electrical signals. Displacement transducers are often most appropriate for low-frequency measurements. For most HVAC applications, the transducer of choice is an accelerometer, which is rugged and compact. The accelerometer attaches to an amplifier, which connects to a meter, much like the microphone on a sound level meter. Readouts may be in acceleration level or decibels. The measurement also specifies whether the amplitude of the acceleration sinusoid is defined by its peak, peak-to-peak, or RMS level. For steady-state (continuous) vibration, simple relationships exist between displacement, velocity, and acceleration; output can be specified as any of these, regardless of which transducer type is used. For a given frequency f, a = (2f)2d = (2f)v where a is acceleration, v is velocity, and d is displacement. (49) A a c d = = = = df dI E F f fc fd fn fr h I Iave k L Leq LI Lp Lw M M1 M2 N p Q R r = Re = S = t or T = T = T60 = V = v = vb = v0 = W = w = W0 = WM = ws = WSG = x = za = Zf = ZI = SYMBOLS magnitude of physical property [Equation (1)], surface area, m2 acceleration, m/s2 speed of sound in air, 335 m/s distance of measurement from nearest reflecting surface, or displacement, m deflection of foundation deflection of mounts Young’s modulus, N/m2 force applied to structure, N frequency, Hz critical frequency, Hz disturbing frequency, Hz system natural frequency, Hz rotation speed of fan blades, rev/s thickness of outer panel of partition, mm sound intensity, dB time-averaged sound intensity stiffness of vibration isolator, N/m level of magnitude of sound or vibration equivalent continuous sound pressure level level of sound intensity sound pressure level, dB sound power level mass of equipment supported by isolator, kg mass of equipment before additional mass added, kg mass of equipment after additional mass added, kg number of fan blades acoustic pressure directivity factor, dimensionless room constant distance between site of measurement and nondirectional sound source, m real part of complex quantity surface area, m2 time, s system transmissibility reverberation time volume of room, m3 velocity velocity of flow over fan blade, m/s free stream velocity of flow into fan, m/s total sound power sound power of source, W input power output power surface mass of panel, kg/m2 power of self-noise location of measurement of pressure p, Equation (17) acoustic impedance, kg/(m2 ·s) or (N·s)/m3 impedance of foundation where isolator attached, (N·s)/m isolator impedance, (N·s)/m This file is licensed to John Murray (). Publication Date: 6/1/2017 8.20 Zm 2017 ASHRAE Handbook—Fundamentals (SI) = mechanical impedance of structure, (N·s)/m Greek st = = = = = = absorption coefficient average absorption coefficient of room surface at given frequency isolator static deflection, mm direction wavelength, m density, kg/m3 Subscripts 0 abs ave i inc ref w = = = = = = = = maximum amplitude absorbed average value for ith source incident reference magnitude of physical property [Equation (1)] sound power REFERENCES Licensed for single user. © 2017 ASHRAE, Inc. ASHRAE members can access ASHRAE Journal articles and ASHRAE research project final reports at technologyportal .ashrae.org. Articles and reports are also available for purchase by nonmembers in the online ASHRAE Bookstore at www.ashrae.org /bookstore. AHRI. 2014. Reverberation room qualification and testing procedures for determining sound power of HVAC equipment. ANSI/AHRI Standard 220-2014. 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Publication Date: 6/1/2017 Related Commercial Resources CHAPTER 9 THERMAL COMFORT Human Thermoregulation... 9.1 Energy Balance.. 9.2 Thermal Exchanges with Environment .... 9.2 Engineering Data and Measurements .... 9.6 Conditions for Thermal Comfort .... 9.12 Thermal Comfort and Task Performance .... 9.14 Thermal Nonuniform Conditions and Local Discomfort .... Secondary Factors Affecting Comfort..................................... Prediction of Thermal Comfort.. Environmental Indices .... Special Environments... Symbols .... principal purpose of HVAC is to provide conditions for human thermal comfort, “that condition of mind that expresses satisfaction with the thermal environment and is assessed by subjective evaluation” (ASHRAE Standard 55). This definition leaves open what is meant by “condition of mind” or “satisfaction,” but it correctly emphasizes that judgment of comfort is a cognitive process involving many inputs influenced by physical, physiological, psychological, and other processes. This chapter summarizes the fundamentals of human thermoregulation and comfort in terms useful to the engineer for operating systems and designing for the comfort and health of building occupants. The conscious mind appears to reach conclusions about thermal comfort and discomfort from direct temperature and moisture sensations from the skin, deep body temperatures, and the efforts necessary to regulate body temperatures (Berglund 1995; Gagge 1937; Hardy et al. 1971; Hensel 1973, 1981). In general, comfort occurs when body temperatures are held within narrow ranges, skin moisture is low, and the physiological effort of regulation is minimized. Comfort also depends on behaviors that are initiated consciously or unconsciously and guided by thermal and moisture sensations to reduce discomfort. Some examples are altering clothing, altering activity, changing posture or location, changing the thermostat setting, opening a window, complaining, or leaving the space. Surprisingly, although climates, living conditions, and cultures differ widely throughout the world, the temperature that people choose for comfort under similar conditions of clothing, activity, humidity, and air movement has been found to be very similar (Busch 1992; de Dear et al. 1991; Fanger 1972). per unit area of skin. For a resting person, this is about 58 W/m2 and is called 1 met. This is based on the average male European, with a skin surface area of about 1.8 m2. For comparison, female Europeans have an average surface area of 1.6 m2. Systematic differences in this parameter may occur between ethnic and geographical groups. Higher metabolic rates are often described in terms of the resting rate. Thus, a person working at metabolic rate five times the resting rate would have a metabolic rate of 5 met. The hypothalamus, located in the brain, is the central control organ for body temperature. It has hot and cold temperature sensors and is bathed by arterial blood. Because the recirculation rate of blood is rapid and returning blood is mixed together in the heart before returning to the body, arterial blood is indicative of the average internal body temperature. The hypothalamus also receives thermal information from temperature sensors in the skin and perhaps other locations as well (e.g., spinal cord, gut), as summarized by Hensel (1981). The hypothalamus controls various physiological processes to regulate body temperature. Its control behavior is primarily proportional to deviations from set-point temperatures with some integral and derivative response aspects. The most important and often-used physiological process is regulating blood flow to the skin: when internal temperatures rise above a set point, more blood is directed to the skin. This vasodilation of skin blood vessels can increase skin blood flow by 15 times [from 1.7 mL/(s·m2) at resting comfort to 25 mL/(s·m2)] in extreme heat to carry internal heat to the skin for transfer to the environment. When body temperatures fall below the set point, skin blood flow is reduced (vasoconstricted) to conserve heat. The effect of maximum vasoconstriction is equivalent to the insulating effect of a heavy sweater. At temperatures less than the set point, muscle tension increases to generate additional heat; where muscle groups are opposed, this may increase to visible shivering, which can increase resting heat production to 4.5 met. At elevated internal temperatures, sweating occurs. This defense mechanism is a powerful way to cool the skin and increase heat loss from the core. The sweating function of the skin and its control is more advanced in humans than in other animals and is increasingly necessary for comfort at metabolic rates above resting level (Fanger 1967). Sweat glands pump perspiration onto the skin surface for evaporation. If conditions are good for evaporation, the skin can remain relatively dry even at high sweat rates with little perception of sweating. At skin conditions less favorable for evaporation, the sweat must spread on the skin around the sweat gland until the sweatcovered area is sufficient to evaporate the sweat coming to the surface. The fraction of the skin that is covered with water to account for the observed total evaporation rate is termed skin wettedness (Gagge 1937). Humans are quite good at sensing skin moisture from perspiration (Berglund 1994; Berglund and Cunningham 1986), and skin moisture correlates well with warm discomfort and unpleasantness (Winslow et al. 1937). It is rare for a sedentary or slightly active person to be comfortable with a skin wettedness greater than 25%. In addition to Licensed for single user. © 2017 ASHRAE, Inc. A 1. HUMAN THERMOREGULATION Metabolic activities of the body result almost completely in heat that must be continuously dissipated and regulated to maintain normal body temperatures. Insufficient heat loss leads to overheating (hyperthermia), and excessive heat loss results in body cooling (hypothermia). Skin temperature greater than 45°C or less than 18°C causes pain (Hardy et al. 1952). Skin temperatures associated with comfort at sedentary activities are 33 to 34°C and decrease with increasing activity (Fanger 1967). In contrast, internal temperatures rise with activity. The temperature regulatory center in the brain is about 36.8°C at rest in comfort and increases to about 37.4°C when walking and 37.9°C when jogging. An internal temperature less than about 28°C can lead to serious cardiac arrhythmia and death, and a temperature greater than 43°C can cause irreversible brain damage. Therefore, careful regulation of body temperature is critical to comfort and health. A resting adult produces about 100 W of heat. Because most of this is transferred to the environment through the skin, it is often convenient to characterize metabolic activity in terms of heat production The preparation of this chapter is assigned to TC 2.1, Physiology and Human Environment. 9.1 Copyright © 2017, ASHRAE 9.14 9.17 9.17 9.21 9.23 9.28 This file is licensed to John Murray (). Publication Date: 6/1/2017 Licensed for single user. © 2017 ASHRAE, Inc. 9.2 2017 ASHRAE Handbook—Fundamentals (SI) the perception of skin moisture, skin wettedness increases the friction between skin and fabrics, making clothing feel less pleasant and fabrics feel more coarse (Gwosdow et al. 1986). This also occurs with architectural materials and surfaces, particularly smooth, nonhygroscopic surfaces. With repeated intermittent heat exposure, the set point for the onset of sweating decreases and the proportional gain or temperature sensitivity of the sweating system increases (Gonzalez et al. 1978; Hensel 1981). However, under long-term exposure to hot conditions, the set point increases, perhaps to reduce the physiological effort of sweating. Perspiration, as secreted, has a lower salt concentration than interstitial body fluid or blood plasma. After prolonged heat exposure, sweat glands further reduce the salt concentration of sweat to conserve salt. At the skin’s surface, the water in sweat evaporates while the dissolved salt and other constituents remain and accumulate. Because salt lowers the vapor pressure of water and thereby impedes its evaporation, the accumulating salt results in increased skin wettedness. Some of the relief and pleasure of washing after a warm day is related to the restoration of a hypotonic sweat film and decreased skin wettedness. Other adaptations to heat are increased blood flow and sweating in peripheral regions where heat transfer is better. Such adaptations are examples of integral control. Role of Thermoregulatory Effort in Comfort. Chatonnet and Cabanac (1965) compared the sensation of placing a subject’s hand in relatively hot or cold water (30 to 38°C) for 30 s with the subject at different thermal states. When the person was overheated (hyperthermic), the cold water was pleasant and the hot water was very unpleasant, but when the subject was cold (hypothermic), the hand felt pleasant in hot water and unpleasant in cold water. Kuno (1995) describes similar observations during transient whole-body exposures to hot and cold environment. When a subject is in a state of thermal discomfort, any move away from the thermal stress of the uncomfortable environment is perceived as pleasant during the transition. 2. ENERGY BALANCE Figure 1 shows the thermal interaction of the human body with its environment. The total metabolic rate M within the body is the metabolic rate required for the person’s activity Mact plus the metabolic level required for shivering Mshiv (should shivering occur). Some of the body’s energy production may be expended as external work W; the net heat production M – W is transferred to the environment through the skin surface (qsk) and respiratory tract (qres) with any surplus or deficit stored (S), causing the body’s temperature to rise or fall. M – W = qsk + qres + S = (C + R + Esk) + (Cres + Eres) + (Ssk + Scr) = = = = = = = = = Thermal Interaction of Human Body and Environment moisture during respiration Eres . Sensible heat flow from the skin may be a complex mixture of conduction, convection, and radiation for a clothed person; however, it is equal to the sum of the convection C and radiation R heat transfer at the outer clothing surface (or exposed skin). Sensible and latent heat losses from the skin are typically expressed in terms of environmental factors, skin temperature tsk , and skin wettedness w. Factors also account for thermal insulation and moisture permeability of clothing. The independent environmental variables can be summarized as air temperature ta, mean radiant temperature t r , relative air velocity V, and ambient water vapor pressure pa. The independent personal variables that influence thermal comfort are activity and clothing. The rate of heat storage in the body equals the rate of increase in internal energy. The body can be considered as two thermal compartments: the skin and the core (see the Two-Node Model section under Prediction of Thermal Comfort). The storage rate can be written separately for each compartment in terms of thermal capacity and time rate of change of temperature in each compartment: 1 – sk mc p, b dt cr Scr = ------------------------------------ --------d AD (2) sk mc p, b dt sk Ssk = ---------------------- -------d AD (3) where (1) where M W qsk qres C+R Esk Cres Eres Ssk Scr Fig. 1 rate of metabolic heat production, W/m2 rate of mechanical work accomplished, W/m2 total rate of heat loss from skin, W/m2 total rate of heat loss through respiration, W/m2 sensible heat loss from skin, W/m2 total rate of evaporative heat loss from skin, W/m2 rate of convective heat loss from respiration, W/m2 rate of evaporative heat loss from respiration, W/m2 rate of heat storage in skin compartment, W/m2 rate of heat storage in core compartment, W/m2 Heat dissipates from the body to the immediate surroundings by several modes of heat exchange: sensible heat flow C + R from the skin; latent heat flow from sweat evaporation Ersw and from evaporation of moisture diffused through the skin Edif ; sensible heat flow during respiration Cres ; and latent heat flow from evaporation of sk m cp,b AD tcr tsk = = = = = = = fraction of body mass concentrated in skin compartment body mass, kg specific heat capacity of body = 3490 J/(kg·K) DuBois surface area, m2 temperature of core compartment, °C temperature of skin compartment, °C time, s The fractional skin mass sk depends on the rate m· bl of blood flowing to the skin surface. 3. THERMAL EXCHANGES WITH ENVIRONMENT Fanger (1967, 1970), Gagge and Hardy (1967), Hardy (1949), and Rapp and Gagge (1967) give quantitative information on calculating heat exchange between people and the environment. This section summarizes the mathematical statements for various terms of heat exchange used in the heat balance equations (C, R, Esk , Cres , Eres). Terms describing the heat exchanges associated with the This file is licensed to John Murray (). Publication Date: 6/1/2017 Thermal Comfort 9.3 thermoregulatory control mechanisms (qcr,sk , Mshiv , Ersw), values for the coefficients, and appropriate equations for Mact and AD are presented in later sections. Mathematical description of the energy balance of the human body combines rational and empirical approaches to describing thermal exchanges with the environment. Fundamental heat transfer theory is used to describe the various mechanisms of sensible and latent heat exchange, and empirical expressions are used to determine the values of coefficients describing these rates of heat exchange. Empirical equations are also used to describe the thermophysiological control mechanisms as a function of skin and core temperatures in the body. Body Surface Area The terms in Equation (1) have units of power per unit area and refer to the surface area of the nude body. The most useful measure of nude body surface area, originally proposed by DuBois and DuBois (1916), is described by AD = 0.202m0.425l 0.725 (4) Licensed for single user. © 2017 ASHRAE, Inc. C + R = (tsk – tcl)Rcl (10) where Rcl is the thermal resistance of clothing in (m2 ·K)/W. Because it is often inconvenient to include the clothing surface temperature in calculations, Equations (7) and (10) can be combined to eliminate tcl : t sk – t o C + R = -----------------------------------R cl + 1 f cl h (11) where to is defined in Equation (8). Evaporative Heat Loss from Skin Evaporative heat loss Esk from skin depends on the amount of moisture on the skin and the difference between the water vapor pressure at the skin and in the ambient environment: where AD = DuBois surface area, m2 m = mass, kg l = height, m A correction factor fcl = Acl /AD must be applied to the heat transfer terms from the skin (C, R, and Esk) to account for the actual surface area Acl of the clothed body. Table 7 presents fcl values for various clothing ensembles. For a 1.73 m tall, 70 kg man, AD = 1.8 m2. All terms in the basic heat balance equations are expressed per unit DuBois surface area. Sensible Heat Loss from Skin Sensible heat exchange from the skin must pass through clothing to the surrounding environment. These paths are treated in series and can be described in terms of heat transfer (1) from the skin surface, through the clothing insulation, to the outer clothing surface, and (2) from the outer clothing surface to the environment. Both convective C and radiative R heat losses from the outer surface of a clothed body can be expressed in terms of a heat transfer coefficient and the difference between the mean temperature tcl of the outer surface of the clothed body and the appropriate environmental temperature: C = fcl hc(tcl – ta) (5) R = fcl hr t cl – tr (6) where w p sk , s – p a Esk = ---R e, cl + 1 f cl h e (12) where w = skin wettedness, dimensionless psk,s = water vapor pressure at skin, normally assumed to be that of saturated water vapor at tsk , kPa pa = water vapor pressure in ambient air, kPa Re,cl = evaporative heat transfer resistance of clothing layer (analogous to Rcl), (m2 ·kPa)/W he = evaporative heat transfer coefficient (analogous to hc), W/(m2 ·kPa) Procedures for calculating Re,cl and he are given in the section on Engineering Data and Measurements. Skin wettedness is the ratio of the actual evaporative heat loss Esk to the maximum possible evaporative heat loss Emax with the same conditions and a completely wet skin (w = 1). Skin wettedness is important in determining evaporative heat loss. Maximum evaporative potential Emax occurs when w = 1. Evaporative heat loss from the skin is a combination of the evaporation of sweat secreted because of thermoregulatory control mechanisms Ersw and the natural diffusion of water through the skin Edif : Esk = Ersw + Edif (13) Evaporative heat loss by regulatory sweating is directly proportional to the rate of regulatory sweat generation: hc = convective heat transfer coefficient, W/(m2 ·K) hr = linear radiative heat transfer coefficient, W/(m2 ·K) fcl = clothing area factor Acl /AD, dimensionless Ersw = m· rsw hfg The coefficients hc and hr are both evaluated at the clothing surface. Equations (5) and (6) are commonly combined to describe the total sensible heat exchange by these two mechanisms in terms of an operative temperature to and a combined heat transfer coefficient h: C + R = fcl h(tcl – to) Based on Equation (8), operative temperature to can be defined as the average of the mean radiant and ambient air temperatures, weighted by their respective heat transfer coefficients. The actual transport of sensible heat through clothing involves conduction, convection, and radiation. It is usually most convenient to combine these into a single thermal resistance value Rcl , defined by (7) where where hfg = heat of vaporization of water = 2.43 106 J/kg at 30°C m· rsw = rate at which regulatory sweat is generated, kg/(s·m2) The portion wrsw of a body that must be wetted to evaporate the regulatory sweat is wrsw = Ersw /Emax h r tr + h c t a to = ------------------------hr + hc (8) h = hr + hc (9) (14) (15) With no regulatory sweating, skin wettedness caused by diffusion is approximately 0.06 for normal conditions. For large values of Emax or long exposures to low humidities, the value may drop to as low as 0.02, because dehydration of the outer skin layers alters its diffusive This file is licensed to John Murray (). Publication Date: 6/1/2017 9.4 2017 ASHRAE Handbook—Fundamentals (SI) characteristics. With regulatory sweating, the 0.06 value applies only to the portion of skin not covered with sweat (1 wrsw); the diffusion evaporative heat loss is Edif = (1 – wrsw)0.06Emax (16) These equations can be solved for w, given the maximum evaporative potential Emax and the regulatory sweat generation Ersw: w = wrsw + 0.06(1 – wrsw) = 0.06 + 0.94Ersw Emax (17) Once skin wettedness is determined, evaporative heat loss from the skin is calculated from Equation (12), or by Esk = wEmax (18) To summarize, the following calculations determine w and Esk: Licensed for single user. © 2017 ASHRAE, Inc. Emax Ersw w Esk Equation (12), with w = 1.0 Equation (14) Equation (17) Equation (18) or (12) where M = metabolic rate, W/m2 Kres = proportionality constant 1.43 10–6 kg/J For typical indoor environments (McCutchan and Taylor 1951), the exhaled temperature and humidity ratio are given in terms of ambient conditions: tex = 32.6 + 0.066ta + 32Wa (22) Wex = 0.0277 + 0.000065ta + 0.2Wa (23) where ambient ta and exhaled tex air temperatures are in °C. For extreme conditions, such as outdoor winter environments, different relationships may be required (Holmér 1984). The humidity ratio of ambient air can be expressed in terms of total or barometric pressure pt and ambient water vapor pressure pa: 0.622 p a Wa = -------------------pt – pa Although evaporation from the skin Esk as described in Equation (12) depends on w, the body does not directly regulate skin wettedness but, rather, regulates sweat rate m· rsw [Equation (14)]. Skin wettedness is then an indirect result of the relative activity of the sweat glands and the evaporative potential of the environment. Skin wettedness of 1.0 is the upper theoretical limit. If the aforementioned calculations yield a wettedness of more than 1.0, then Equation (14) is no longer valid because not all the sweat is evaporated. In this case, Esk = Emax. Skin wettedness is strongly correlated with warm discomfort and is also a good measure of thermal stress. Theoretically, skin wettedness can approach 1.0 while the body still maintains thermoregulatory control. In most situations, it is difficult to exceed 0.8 (Berglund and Gonzalez 1977). Azer (1982) recommends 0.5 as a practical upper limit for sustained activity for a healthy, acclimatized person. (24) Respiratory heat loss is often expressed in terms of sensible Cres and latent Eres heat losses. Two approximations are commonly used to simplify Equations (22) and (23) for that purpose. First, because dry respiratory heat loss is relatively small compared to the other terms in the heat balance, an average value for tex is determined by evaluating Equation (22) at standard conditions of 20°C, 50% rh, sea level. Second, noting in Equation (23) that there is only a weak dependence on ta, the second term in Equation (23) and the denominator in Equation (24) are evaluated at standard conditions. Using these approximations and substituting latent heat hfg and specific heat of air cp,a at standard conditions, Cres and Eres can be determined by Cres = 0.0014M(34 – ta) (25) Eres = 0.0173M(5.87 – pa) (26) where pa is expressed in kPa and ta is in °C. Respiratory Losses During respiration, the body loses both sensible and latent heat by convection and evaporation of heat and water vapor from the respiratory tract to the inhaled air. A significant amount of heat can be associated with respiration because air is inspired at ambient conditions and expired nearly saturated at a temperature only slightly cooler than tcr . The total heat and moisture losses through respiration are m· res h ex – h a qres = Cres + Eres = ---------------------------------AD (19) m· res W ex – W a m· w, res = -------------------------------------AD (20) where m· res = pulmonary ventilation rate, kg/s hex = enthalpy of exhaled air, J/kg (dry air) ha = enthalpy of inspired (ambient) air, J/kg (dry air) m· w, res = pulmonary water loss rate, kg/s Equations (11) and (12) describe heat loss from skin for clothed people in terms of clothing parameters Rcl , Re,cl , and fcl ; parameters h and he describe outer surface resistances. Other parameters and definitions are also used. Although these alternative parameters and definitions may be confusing, note that information presented in one form can be converted to another form. Table 1 presents common parameters and their qualitative descriptions. Table 2 presents equations showing their relationship to each other. Generally, parameters related to dry or evaporative heat flows are not independent because they both rely, in part, on the same physical processes. The Lewis relation describes the relationship between convective heat transfer and mass transfer coefficients for a surface [see Equation (41) in Chapter 6]. The Lewis relation can be used to relate convective and evaporative heat transfer coefficients defined in Equations (5) and (12) according to LR = he hc Wex = humidity ratio of exhaled air, kg (water vapor)/kg (dry air) Wa = humidity ratio of inspired (ambient) air, kg (water vapor)/kg (dry air) Under normal circumstances, pulmonary ventilation rate is primarily a function of metabolic rate (Fanger 1970): m· res = Kres MAD Alternative Formulations (21) (27) where LR is the Lewis ratio and, at typical indoor conditions, equals approximately 16.5 K/kPa. The Lewis relation applies to surface convection coefficients. Heat transfer coefficients that include the effects of insulation layers and/or radiation are still coupled, but the relationship may deviate significantly from that for a surface. The i terms in Tables 1 and 2 describe how the actual ratios of these parameters deviate from the ideal Lewis ratio (Oohori et al. 1984; Woodcock 1962). This file is licensed to John Murray (). Publication Date: 6/1/2017 Thermal Comfort 9.5 Table 1 Parameters Used to Describe Clothing Licensed for single user. © 2017 ASHRAE, Inc. Sensible Heat Flow Rcl = intrinsic clothing insulation: thermal resistance of a uniform layer of insulation covering entire body that has same effect on sensible heat flow as actual clothing. Rt = total insulation: total equivalent uniform thermal resistance between body and environment: clothing and boundary resistance. Rcle = effective clothing insulation: increased body insulation due to clothing as compared to nude state. Ra = boundary insulation: thermal resistance at skin boundary for nude body. Ra,cl = outer boundary insulation: thermal resistance at outer boundary (skin or clothing). Rte = total effective insulation. h = overall sensible heat transfer coefficient: overall equivalent uniform conductance between body (including clothing) and environment. hcl = clothing conductance: thermal conductance of uniform layer of insulation covering entire body that has same effect on sensible heat flow as actual clothing. Fcle = effective clothing thermal efficiency: ratio of actual sensible heat loss to that of nude body at same conditions. Fcl = intrinsic clothing thermal efficiency: ratio of actual sensible heat loss to that of nude body at same conditions including adjustment for increase in surface area due to clothing. Table 2 Relationships Between Clothing Parameters Sensible Heat Flow Sensible Heat Loss C + R = (tsk to)/[Rcl + 1/(fc l h)] Rte = Rcle + 1/h = Rcle + Ra C + R = (tsk to)/Rt h cl = 1/Rcl C + R = Fc le h(tsk to) h = 1/Rt C + R = Fc l fcl h(tsk to) h = 1/Ra Fcl = h /(hfcl) = 1/(1 + fcl hR cl) Fcle = h /h = fc l /(1 + fcl hRcl) = fcl Fc l C + R = h (tsk to) Evaporative Heat Loss Esk = w(psk,s pa)/[Re,cl + 1/(fc l he)] Esk = w(psk,s pa)/Re,t Evaporative Heat Flow Esk = wFpcl fc l he (psk,s pa) Re,t = Re,cl + 1/(he fc l) = Re,cl + Re,a /fc l Esk = h e w(psk,s pa) Esk = h wimLR(psk,s pa) he = 1/Re,a h e,cl = 1/Re,cl h e = 1/Re,t = fc l Fpc l he Fpcl = 1/(1 + fcl he Re,cl) t sk – t o w p sk , s – p a qsk = ------------------------- + ---R cl + R a, cl R e, cl + 1 LRhc fcl Parameters Relating Sensible and Evaporative Heat Flows icl LR = he ,cl /hc l = R cl /Re,cl im LR = h e /h = Rt /Re,t im = (Rcl + Ra,cl)/[(Rcl /icl) + (Ra,cl /ia)] iaLR = he /h qsk = Fcl fcl h(tsk – to) + wLRFpcl hc(psk,s – pa) (29) This equation allows evaluation of the trade-off between any two or more parameters under given conditions. If the trade-off between two specific variables (e.g., operative temperature and humidity) is to be examined, then a simplified form of the equation suffices (Fobelets and Gagge 1988): qsk = h[(tsk + wimLRpsk,s) – (to + wimLRpa)] Total Skin Heat Loss Total skin heat loss (sensible heat plus evaporative heat) can be calculated from any combination of the equations presented in Table 3. Total skin heat loss is used as a measure of the thermal environment; two combinations of parameters that yield the same total heat loss for a given set of body conditions (tsk and w) are considered to be approximately equivalent. The fully expanded skin heat loss equation, showing each parameter that must be known or specified, is as follows: (28) where to is the operative temperature and represents the temperature of a uniform environment (ta – t r) that transfers dry heat at the same rate as in the actual environment [to = (t r hr + ta hc)/(hc + hr)]. After rearranging, Equation (28) becomes ia = hc /(hc + hr) Depending on the combination of parameters used, heat transfer from the skin can be calculated using several different formulations (see Tables 2 and 3). If the parameters are used correctly, the end result will be the same regardless of the formulation used. Parameters Relating Sensible and Evaporative Heat Flows icl = clothing vapor permeation efficiency: ratio of actual evaporative heat flow capability through clothing to sensible heat flow capability as compared to Lewis ratio. im = total vapor permeation efficiency: ratio of actual evaporative heat flow capability between skin and environment to sensible heat flow capability as compared to Lewis ratio. ia = air layer vapor permeation efficiency: ratio of actual evaporative heat flow capability through outer air layer to sensible heat flow capability as compared to Lewis ratio. Table 3 Skin Heat Loss Equations Rt = Rcl + 1/(hfcl) = Rcl + Ra /fcl Ra,cl = Ra /fcl Evaporative Heat Flow Re,cl = evaporative heat transfer resistance of clothing: impedance to transport of water vapor of uniform layer of insulation covering entire body that has same effect on evaporative heat flow as actual clothing. Re,t = total evaporative resistance: total equivalent uniform impedance to transport of water vapor from skin to environment. Fpcl = permeation efficiency: ratio of actual evaporative heat loss to that of nude body at same conditions, including adjustment for increase in surface area due to clothing. (30) Equation (30) can be used to define a combined temperature tcom, which reflects the combined effect of operative temperature and humidity for an actual environment: tcom + wimLRptcom = to + wimLRpa or tcom = to + wimLRpa – wimLRptcom (31) This file is licensed to John Murray (). Publication Date: 6/1/2017 9.6 2017 ASHRAE Handbook—Fundamentals (SI) Licensed for single user. © 2017 ASHRAE, Inc. Table 4 Typical Metabolic Heat Generation for Various Activities Fig. 2 Constant Skin Heat Loss Line and Its Relationship to toh and ET* where ptcom is a vapor pressure related in some fixed way to tcom and is analogous to pwb,s for twb. The term wimLRptcom is constant to the extent that im is constant, and any combination of to and pa that gives the same tcom results in the same total heat loss. Two important environmental indices, humid operative temperature toh and effective temperature ET*, can be represented in terms of Equation (31). The humid operative temperature is that temperature which at 100% rh yields the same total heat loss as for the actual environment: toh = to + wimLR(pa – poh,s) (32) where poh,s is saturated vapor pressure, in kPa, at toh. The effective temperature is the temperature at 50% rh that yields the same total heat loss from the skin as for the actual environment: ET* = to + wimLR(pa – 0.5pET*,s) (33) where pET*,s is saturated vapor pressure, in kPa, at ET*. The psychrometric chart in Figure 2 shows a constant total heat loss line and the relationship between these indices. This line represents only one specific skin wettedness and permeation efficiency index. The relationship between indices depends on these two parameters (see the section on Environmental Indices). 4. ENGINEERING DATA AND MEASUREMENTS Applying basic equations to practical problems of the thermal environment requires quantitative estimates of the body’s surface area, metabolic requirements for a given activity and the mechanical efficiency for the work accomplished, evaluation of heat transfer coefficients hr and hc, and the general nature of clothing insulation used. This section provides the necessary data and describes how to measure the parameters of the heat balance equation. Metabolic Rate and Mechanical Efficiency Maximum Capacity. In choosing optimal conditions for comfort and health, the rate of work done during routine physical activities W/m2 met* Resting Sleeping Reclining Seated, quiet Standing, relaxed 40 45 60 70 0.7 0.8 1.0 1.2 Walking (on level surface) 3.2 km/h (0.9 m/s) 4.3 km/h (1.2 m/s) 6.4 km/h (1.8 m/s) 115 150 220 2.0 2.6 3.8 Office Activities Reading, seated Writing Typing Filing, seated Filing, standing Walking about Lifting/packing 55 60 65 70 80 100 120 1.0 1.0 1.1 1.2 1.4 1.7 2.1 60 to 115 70 105 140 185 1.0 to 2.0 1.2 1.8 2.4 3.2 95 to 115 115 to 200 130 1.6 to 2.0 2.0 to 3.4 2.2 105 115 to 140 235 235 235 to 280 1.8 2.0 to 2.4 4.0 4.0 4.0 to 4.8 140 to 255 175 to 235 210 to 270 290 to 440 410 to 505 2.4 to 4.4 3.0 to 4.0 3.6 to 4.0 5.0 to 7.6 7.0 to 8.7 Driving/Flying Car Aircraft, routine Aircraft, instrument landing Aircraft, combat Heavy vehicle Miscellaneous Occupational Activities Cooking Housecleaning Seated, heavy limb movement Machine work sawing (table saw) light (electrical industry) heavy Handling 50 kg bags Pick and shovel work Miscellaneous Leisure Activities Dancing, social Calisthenics/exercise Tennis, singles Basketball Wrestling, competitive Sources: Compiled from various sources. For additional information, see Buskirk (1960), Passmore and Durnin (1967), and Webb (1964). *1 met = 58.1 W/m2 must be known, because metabolic power increases in proportion to exercise intensity. Metabolic rate varies over a wide range, depending on the activity, person, and conditions under which the activity is performed. Table 4 lists typical metabolic rates for an average adult (AD = 1.8 m2) for activities performed continuously. The highest power a person can maintain for any continuous period is approximately 50% of the maximal capacity to use oxygen (maximum energy capacity). A unit used to express the metabolic rate per unit DuBois area is the met, defined as the metabolic rate of a sedentary person (seated, quiet): 1 met = 58.1 W/m2 = 50 kcal/(h · m2). A normal, healthy man at age 20 has a maximum capacity of approximately Mact = 12 met, which drops to 7 met at age 70. Maximum rates for women are on average about 30% lower. Long-distance runners and trained athletes have maximum rates as high as 20 met. An average 35-year-old who does not exercise has a maximum rate of about 10 met, and activities with Mact > 5 met are likely to prove exhausting. Intermittent Activity. Often, people’s activity consists of a mixture of activities or a combination of work/rest periods. A This file is licensed to John Murray (). Publication Date: 6/1/2017 Licensed for single user. © 2017 ASHRAE, Inc. Thermal Comfort 9.7 weighted average metabolic rate is generally satisfactory, provided that activities alternate frequently (several times per hour). For example, a person whose activities consist of typing 50% of the time, filing while seated 25% of the time, and walking about 25% of the time would have an average metabolic rate of 0.5065 + 0.2570 + 0.25100 = 75 W/m2 (see Table 4). Accuracy. Estimating metabolic rates is difficult. Values given in Table 4 indicate metabolic rates only for the specific activities listed. Some entries give a range and some a single value, depending on the data source. The level of accuracy depends on the value of Mact and how well the activity can be defined. For well-defined activities with Mact < 1.5 met (e.g., reading), Table 4 is sufficiently accurate for most engineering purposes. For values of Mact > 3, where a task is poorly defined or where there are various ways of performing a task (e.g., heavy machine work), the values may be in error by as much as ±50% for a given application. Engineering calculations should thus allow for potential variations. Measurement. When metabolic rates must be determined more accurately than is possible with tabulated data, physiological measurements with human subjects may be necessary. The rate of metabolic heat produced by the body is most accurately measured by the rate of respiratory oxygen consumption and carbon dioxide production. An empirical equation for metabolic rate is given by Nishi (1981): 21 0.23RQ + 0.77 Q O M = --2 AD (34) where M = metabolic rate, W/m2 RQ = respiratory quotient; molar ratio of QCO2 exhaled to QO2 inhaled, dimensionless QO2 = volumetric rate of oxygen consumption at conditions (STPD) of 0°C, 101.325 kPa, mL/s The exact value of the respiratory quotient RQ

depends on a person’s activity, diet, and physical condition. It can be determined by measuring both carbon dioxide and oxygen in the respiratory airflows, or it can be estimated with reasonable accuracy. A good estimate for the average adult is RQ = 0.83 for light or sedentary activities (M < 1.5 met), increasing proportionately to RQ = 1.0 for extremely heavy exertion (M = 5.0 met). Estimating RQ is generally sufficient for all except precision laboratory measurements because it does not strongly affect the value of the metabolic rate: a 10% error in estimating RQ results in an error of less than 3% in the metabolic rate. A second, much less accurate, method of estimating metabolic rate physiologically is to measure the heart rate. Table 5 shows the relationship between heart rate and oxygen consumption at different levels of physical exertion for a typical person. Once oxygen consumption is estimated from heart rate information, Equation (34) can be used to estimate the metabolic rate. Other factors that affect heart rate include physical condition, heat, emotional factors, and muscles used. Astrand and Rodahl (1977) show that heart rate is only a very approximate measure of metabolic rate and should not be the only source of information where accuracy is required. Table 5 Heart Rate and Oxygen Consumption at Different Activity Levels Level of Exertion Light work Moderate work Heavy work Very heavy work Extremely heavy work Source: Astrand and Rodahl (1977). Heart Rate, bpm Oxygen Consumed, mL/s 100%, insert PD = 100%. Vsd is the standard deviation of the velocity measured with an omnidirectional anemometer having a 0.2 s time constant. The model extends the Fanger and Christensen (1986) draft chart model to include turbulence intensity. In this study, Tu decreases when V increases. Thus, the effects of V for the experimental data to which the model is fitted are 20 < ta < 26°C, 0.05 < V < 0.5 m/s, and 0 < Tu < 70%. Figure 11 gives more precisely the curves that result from intersections between planes of constant Tu and the surfaces of PD = 15%. At thermal conditions above neutrality, air movement can be beneficial for thermal comfort. Arens et al. (2009) found people prefer more air movement under some conditions in office spaces. This file is licensed to John Murray (). Publication Date: 6/1/2017 9.16 2017 ASHRAE Handbook—Fundamentals (SI) Applications of this include ceiling fans and personal environmental control systems in offices and transportation systems. Licensed for single user. © 2017 ASHRAE, Inc. Vertical Air Temperature Difference In most buildings, air temperature normally increases with height above the floor. If the gradient is sufficiently large, local warm discomfort can occur at the head and/or cold discomfort can occur at the feet, although the body as a whole is thermally neutral. Among the few studies of vertical air temperature differences and the influence of thermal comfort reported are Eriksson (1975), McNair (1973), McNair and Fishman (1974), and Olesen et al. (1979). Subjects were seated in a climatic chamber and individually exposed to different air temperature differences between head and ankles (Olesen et al. 1979). During the tests, the subjects were in thermal neutrality because they were allowed to change the temperature level in the test room whenever they desired; the vertical temperature difference, however, was kept unchanged. Subjects gave subjective reactions to their thermal sensation; Figure 12 shows the percentage of dissatisfied as a function of the vertical air temperature difference between head (1.1 m above the floor) and ankles (0.1 m above the floor). A head-level air temperature lower than that at ankle level is not as critical for occupants. Eriksson (1975) indicated that subjects could tolerate much greater differences if the head were cooler. This observation is verified in experiments with asymmetric thermal radiation from a cooled ceiling (Fanger et al. 1985). Warm or Cold Floors Because of direct contact between the feet and the floor, local discomfort of the feet can often be caused by a too-high or too-low floor temperature. Also, floor temperature significantly influences a room’s mean radiant temperature. Floor temperature is greatly affected by building construction (e.g., insulation of the floor, above a basement, directly on the ground, above another room, use of floor heating, floors in radiant-heated areas). If a floor is too cold and the occupants feel cold discomfort in their feet, a common reaction is to increase the temperature level in the room; in the heating season, this also increases energy consumption. A radiant system, which radiates heat from the floor, can also prevent discomfort from cold floors. The most extensive studies of the influence of floor temperature on foot comfort were performed by Olesen (1977a, 1977b), who, based on his own experiments and reanalysis of data from Nevins and Feyerherm (1967), Nevins and Flinner (1958), and Nevins et al. (1964), found that flooring material is important for people with bare feet (e.g., in swimming halls, gymnasiums, dressing rooms, bathrooms, bedrooms). Ranges for some typical floor materials are as follows: Textiles (rugs) Pine floor Oak floor Hard linoleum Concrete 21 to 28°C 22.5 to 28°C 24.5 to 28°C 24 to 28°C 26 to 28.5°C To save energy, insulating flooring materials (cork, wood, carpets), radiant heated floors, or floor heating systems can be used to eliminate the desire for higher ambient temperatures caused by cold feet. These recommendations should also be followed in schools, where children often play directly on the floor. For people wearing normal indoor footwear, flooring material is insignificant. Olesen (1977b) found an optimal temperature of 25°C for sedentary and 23°C for standing or walking persons. At the optimal temperature, 6% of occupants felt warm or cold discomfort in the feet. Figure 13 shows the relationship between floor temperature and percent dissatisfied, combining data from experiments with seated and standing subjects. In all experiments, subjects were in thermal neutrality; thus, the percentage of dissatisfied is only related to discomfort caused by cold or warm feet. No significant difference in preferred floor temperature was found between females and males. 8. SECONDARY FACTORS AFFECTING COMFORT Temperature, air speed, humidity, their variation, and personal parameters of metabolism and clothing insulation are primary factors that directly affect energy flow and thermal comfort. However, many secondary factors, some of which are discussed in this section, may more subtly influence comfort. Fig. 11 Draft Conditions Dissatisfying 15% of Population (PD = 15%) Fig. 12 Percentage of Seated People Dissatisfied as Function of Air Temperature Difference Between Head and Ankles This file is licensed to John Murray (). Publication Date: 6/1/2017 Thermal Comfort Day-to-Day Variations Fanger (1973) determined the preferred ambient temperature for each of a group of subjects under identical conditions on four different days. Because the standard deviation was only 0.6 K, Fanger concluded that comfort conditions for an individual can be reproduced and vary only slightly from day to day. Licensed for single user. © 2017 ASHRAE, Inc. Age Because metabolism decreases slightly with age, many have stated that comfort conditions based on experiments with young and healthy subjects cannot be used for other age groups. Fanger (1982), Fanger and Langkilde (1975), Langkilde (1979), Nevins et al. (1966), and Rohles and Johnson (1972) conducted comfort studies in Denmark and the United States on different age groups (mean ages 21 to 84). The studies revealed that the thermal environments preferred by older people do not differ from those preferred by younger people. The lower metabolism in older people is compensated for by a lower evaporative loss. Collins and Hoinville (1980) confirmed these results. The fact that young and old people prefer the same thermal environment does not necessarily mean that they are equally sensitive to cold or heat. In practice, the ambient temperature level in the homes of older people is often higher than that for younger people. This may be explained by the lower activity level of elderly people, who are normally sedentary for a greater part of the day. Adaptation Many believe that people can acclimatize themselves by exposure to hot or cold surroundings, so that they prefer other thermal environments. Fanger (1982) conducted experiments involving subjects from the United States, Denmark, and tropical countries. The latter group was tested in Copenhagen immediately after their arrival by plane from the tropics, where they had lived all their lives. Other experiments were conducted for two groups exposed to cold daily. One group comprised subjects who had been doing sedentary work in cold surroundings (in the meat-packing industry) for 8 h daily for at least 1 year. The other group consisted of winter swimmers who bathed in the sea daily. Only slight differences in preferred ambient temperature and physiological parameters in the comfort conditions were reported for the various groups. These results indicate that people cannot adapt to preferring warmer or colder environments, and therefore the same comfort conditions can likely be applied throughout the world. However, in determining the preferred ambient temperature from the comfort equations, a clo-value corresponding to local clothing 9.17 habits should be used, such as those given in Table 8 and in Havenith et al. (2015). A comparison of field comfort studies from different parts of the world shows significant differences in clothing habits depending on, among other things, outdoor climate (Nicol and Humphreys 1972). According to these results, adaptation has little influence on preferred ambient temperature. In uncomfortable warm or cold environments, however, adaptation often has an influence. People used to working and living in warm climates can more easily accept and maintain a higher work performance in hot environments than people from colder climates. Sex Fanger (1982), Fanger and Langkilde (1975), and Nevins et al. (1966) used equal numbers of male and female subjects, so comfort conditions for the two sexes can be compared. The experiments show that men and women prefer almost the same thermal environments. Women’s skin temperature and evaporative loss are slightly lower than those for men, and this balances the somewhat lower metabolism of women. The reason that women often prefer higher ambient temperatures than men may be partly explained by the lighter clothing often worn by women. Seasonal and Circadian Rhythms Because people cannot adapt to prefer warmer or colder environments, it follows that there is no difference between comfort conditions in winter and in summer. McNall et al. (1968) confirmed this in an investigation where results of winter and summer experiments showed no difference. On the other hand, it is reasonable to expect comfort conditions to alter during the day because internal body temperature has a daily rhythm, with a maximum late in the afternoon, and a minimum early in the morning. In determining the preferred ambient temperature for each of 16 subjects both in the morning and in the evening, Fanger et al. (1974) and Ostberg and McNicholl (1973) observed no difference. Furthermore, Fanger et al. (1973) found only small fluctuations in preferred ambient temperature during a simulated 8 h workday (sedentary work). There is a slight tendency to prefer somewhat warmer surroundings before lunch, but none of the fluctuations are significant. 9. PREDICTION OF THERMAL COMFORT Thermal comfort and thermal sensation can be predicted several ways. One way is to use Figure 5 and Table 10 and adjust for clothing and activity levels that differ from those of the figure. More numerical and rigorous predictions are possible by using the PMVPPD and two-node models described in this section. Steady-State Energy Balance Fanger (1982) related comfort data to physiological variables. At a given level of metabolic activity M, and when the body is not far from thermal neutrality, mean skin temperature tsk and sweat rate Ersw are the only physiological parameters influencing heat balance. However, heat balance alone is not sufficient to establish thermal comfort. In the wide range of environmental conditions where heat balance can be obtained, only a narrow range provides thermal comfort. The following linear regression equations, based on data from Rohles and Nevins (1971), indicate values of tsk and Ersw that provide thermal comfort: Fig. 13 Percentage of People Dissatisfied as Function of Floor Temperature tsk,req = 35.7 – 0.0275(M – W) (61) Ersw,req = 0.42 (M – W – 58.15) (62) At higher activity levels, sweat loss increases and mean skin temperature decreases, both of which increase heat loss from the body This file is licensed to John Murray (). Publication Date: 6/1/2017 9.18 2017 ASHRAE Handbook—Fundamentals (SI) Licensed for single user. © 2017 ASHRAE, Inc. core to the environment. These two empirical relationships link the physiological and heat flow equations and thermal comfort perceptions. By substituting these values into Equation (11) for C + R, and into Equations (17) and (18) for Esk , Equation (1) (the energy balance equation) can be used to determine combinations of the six environmental and personal parameters that optimize comfort for steady-state conditions. Fanger (1982) reduced these relationships to a single equation, which assumed all sweat generated is evaporated, eliminating clothing permeation efficiency icl as a factor in the equation. This assumption is valid for normal indoor clothing worn in typical indoor environments with low or moderate activity levels. At higher activity levels (Mact > 3 met), where a significant amount of sweating occurs even at optimum comfort conditions, this assumption may limit accuracy. The reduced equation is slightly different from the heat transfer equations developed here. The radiant heat exchange is expressed in terms of the Stefan-Boltzmann law (instead of using hr), and diffusion of water vapor through the skin is expressed as a diffusivity coefficient and a linear approximation for saturated vapor pressure evaluated at tsk. The combination of environmental and personal variables that produces a neutral sensation may be expressed as follows: Fig. 14 Air Velocities and Operative Temperatures at 50% rh Necessary for Comfort (PMV = 0) of Persons in Summer Clothing at Various Levels of Activity M – W = 3.96 10–8fcl [(tcl + 273)4 – (tr + 273)4] + fcl hc(tcl – ta) + 3.05[5.73 – 0.007(M – W) – pa] + 0.42[(M – W) – 58.15] + 0.0173M(5.87 – pa) + 0.0014M(34 – ta) (63) where tcl = 35.7 – 0.0275(M – W) – Rcl {(M – W) – 3.05[5.73 – 0.007(M – W) – pa] – 0.42[(M – W) – 58.15] – 0.0173M(5.87 – pa) – 0.0014M(34 – ta)} (64) The values of hc and fcl can be estimated from tables and equations given in the section on Engineering Data and Measurements. Fanger used the following relationships: 0.25 2.38 t cl – t a hc = 12.1 V 1.0 + 0.2I cl fcl = 1.05 + 0.1I cl 2.38 t cl – t a 0.25 > 12.1 V 0.25 < 12.1 V 2.38 t cl – t a I cl < 0.5 clo I cl > 0.5 clo (65) (66) Figures 14 and 15 show examples of how Equation (63) can be used. Equation (63) is expanded to include a range of thermal sensations by using a predicted mean vote (PMV) index. The PMV index predicts the mean response of a large group of people according to the ASHRAE thermal sensation scale. Fanger (1970) related PMV to the imbalance between actual heat flow from the body in a given environment and the heat flow required for optimum comfort at the specified activity by the following equation: PMV = [0.303 exp (–0.036M) + 0.028]L (67) where L is the thermal load on the body, defined as the difference between internal heat production and heat loss to the actual environment for a person hypothetically kept at comfort values of tsk and Ersw at the actual activity level. Thermal load L is then the difference between the left and right sides of Equation (63) calculated for the actual values of the environmental conditions. As part of this calculation, clothing temperature tcl is found by iteration as Fig. 15 Air Temperatures and Mean Radiant Temperatures Necessary for Comfort (PMV = 0) of Sedentary Persons in Summer Clothing at 50% rh tcl = 35.7 – 0.028(M – W) – Rcl {39.6 10–9fcl [(tcl + 273)4 – (tr + 273)4] + fcl hc(tcl – ta)} (68) After estimating the PMV with Equation (67) or another method, the predicted percent dissatisfied (PPD) with a condition can also be estimated. Fanger (1982) related the PPD to the PMV as follows: PPD = 100 – 95 exp[–(0.03353PMV 4 + 0.2179PMV 2)] (69) where dissatisfied is defined as anybody not voting 1, +1, or 0. This relationship is shown in Figure 16. A PPD of 10% corresponds to the PMV range of ±0.5, and even with PMV = 0, about 5% of the people are dissatisfied. The PMV-PPD model is widely used and accepted for design and field assessment of comfort conditions. ISO Standard 7730 includes a short computer listing that facilitates computing PMV and PPD for a wide range of parameters. This file is licensed to John Murray (). Publication Date: 6/1/2017 Thermal Comfort 9.19 The temperature terms can only be > 0. If the deviation is negative, the term is set to zero. For average persons, the coefficients BFN, cdil , and Str are 6.3, 50, and 0.5. Further, skin blood flow Qbl is limited to a maximum of 90 L/(h·m2). A very fit and well-trained athlete could expect to have cdil = 175. Dry (sensible) heat loss qdry from the skin flows through the clothing by conduction and then by parallel paths to the air and surrounding surfaces. Evaporative heat follows a similar path, flowing through the clothing and through the air boundary layer. Maximum evaporation Emax occurs if the skin is completely covered with sweat. The actual evaporation rate Esw depends on the size w of the sweat film: Esw = wEmax (72) where w is Ersw /Emax. Fig. 16 Predicted Percentage of Dissatisfied (PPD) as Function of Predicted Mean Vote (PMV) The rate of regulatory sweating Ersw (rate at which water is brought to the surface of the skin in W/m) can be predicted by skin and core temperature deviations from their set points: Licensed for single user. © 2017 ASHRAE, Inc. Two-Node Model The PMV model is useful only for predicting steady-state comfort responses. The two-node model can be used to predict physiological responses or responses to transient situations, at least for low and moderate activity levels in cool to very hot environments (Gagge et al. 1971a, 1986). This model is a simplification of thermoregulatory models developed by Stolwijk and Hardy (1966). The simple, lumped parameter model considers a human as two concentric thermal compartments that represent the skin and the core of the body. The skin compartment simulates the epidermis and dermis and is about 1.6 mm thick. Its mass, which is about 10% of the total body, depends on the amount of blood flowing through it for thermoregulation. Compartment temperature is assumed to be uniform so that the only temperature gradients are between compartments. In a cold environment, blood flow to the extremities may be reduced to conserve the heat of vital organs, resulting in axial temperature gradients in the arms, legs, hands, and feet. Heavy exercise with certain muscle groups or asymmetric environmental conditions may also cause nonuniform compartment temperatures and limit the model’s accuracy. All the heat is assumed to be generated in the core compartment. In the cold, shivering and muscle tension may generate additional metabolic heat. This increase is related to skin and core temperature depressions from their set point values, or Mshiv = [156(37 – tc) + 47(33 – tsk) – 1.57(33 – tsk)2]/BF0.5 (70) where BF is percentage body fat and the temperature difference terms are set to zero if they become negative (Tikusis and Giesbrecht 1999). The core loses energy when the muscles do work on the surroundings. Heat is also lost from the core through respiration. The rate of respiratory heat loss depends on sensible and latent changes in respired air and the ventilation rate as in Equations (19) and (20). In addition, heat is conducted passively from the core to the skin. This is modeled as a massless thermal conductor [K = 5.28 W/(m2 · K)]. A controllable heat loss path from the core consists of pumping variable amounts of warm blood to the skin for cooling. This peripheral blood flow Qbl in L/(h · m2) depends on skin and core temperature deviations from their respective set points: BFN + c dil t cr – 37 Qbl = ---1 + S tr 34 – t sk (71) Ersw = csw(tb – tbset)exp[(tsk – 34)10.7] (73) where tb = (1 – sk)tcr + sk tsk and is the mean body temperature, tbset = 36.49°C, and csw = 116 W/(m2 ·K). The temperature deviation terms are set to zero when negative. he fraction of the total body mass considered to be thermally in the skin compartment is sk: 0.745 sk = 0.0418 + -----------------------------Q bl + 0.585 (74) Regulatory sweating Qrsw in the model is limited to 1 L/(h·m2) or 670 W/m2. Ersw evaporates from the skin, but if Ersw is greater than Emax , the excess drips off. An energy balance on the core yields M + Mshiv = W + qres + (K + SkBFcp,bl)(tcr – tsk) + mcr ccr dt cr (75) d and for the skin, (K + SkBFcp,bl)(tcr – tsk) = qdry + qevap + mskcsk dt sk d (76) where ccr , csk , and cp,bl are specific heats of core, skin, and blood [3500, 3500, and 4190 J/(kg· K), respectively], and SkBF is bl Qbl, where bl is density of blood (1.06 kg/L). Equations (75) and (76) can be rearranged in terms of dtsk /d and dtcr/d and numerically integrated with small time steps (10 to 60 s) from initial conditions or previous values to find tcr and tsk at any time. After calculating values of tsk , tcr , and w, the model uses empirical expressions to predict thermal sensation (TSENS) and thermal discomfort (DISC). These indices are based on 11-point numerical scales, where positive values represent the warm side of neutral sensation or comfort, and negative values represent the cool side. TSENS is based on the same scale as PMV, but with extra terms for ±4 (very hot/cold) and ±5 (intolerably hot/cold). Recognizing the same positive/negative convention for warm/cold discomfort, DISC is defined as 5 4 3 2 1 0 intolerable limited tolerance very uncomfortable uncomfortable and unpleasant slightly uncomfortable but acceptable comfortable This file is licensed to John Murray (). Publication Date: 6/1/2017 9.20 2017 ASHRAE Handbook —Fundamentals (SI) TSENS is defined in terms of deviations of mean body temperature tb from cold and hot set points representing the lower and upper limits for the zone of evaporative regulation: tb,c and tb,h, respectively. The values of these set points depend on the net rate of internal heat production and are calculated by 0.194 tb,c = ------------ (M – W) + 36.301 58.15 (77) 0.347 tb,h = ------------ (M – W) + 36.669 58.15 (78) TSENS is then determined by t b < t b, c 0.4685 t b – t b , c t – t t – t 4.7 TSENS = ev b b, c b, h b, c b, c t b t b , h (79) tb , h < tb 4.7 ev + 0.4685 t b – t b , h Licensed for single user. © 2017 ASHRAE, Inc. where ev is the evaporative efficiency (assumed to be 0.85). DISC is numerically equal to TSENS when tb is below its cold set point tb,c and it is related to skin wettedness when body temperature is regulated by sweating: 0.4685 t b – t b , c DISC = 4.7 E rsw – E rsw , req --- E max – E rsw , req – E dif tb < t b , c t b , c tb (80) where Ersw,req is calculated as in Fanger’s model, using Equation (62). Multisegment Thermal Physiology and Comfort Models Unlike the two-node model, which represents the body as one cylinder with two nodes of core and skin, multisegment models divide the body into more segments (e.g., head, chest, hands, feet) and more tissue layers (e.g., core, muscle, fat, skin). They are intended to predict thermal physiology and thermal comfort in nonuniform [e.g., offices with displacement ventilation or underfloor air, radiant-cooled ceiling/floors, or natural and mixed-mode ventilation; personal environmental control (PEC) systems] and transient (e.g., occupants moving between different environments in offices, quick-responding PECs, automobiles) environments. Major multisegment physiological models include Fiala (1998), Fiala et al. (2003), Gordon (1974), Huizenga et al. (2001), Kraning and Gonzalez (1997), Smith (1991), Stolwijk (1971), Tanabe et al. (2002), Werner and Webb (1993), and Wissler (1964, 1985, 1988). These models mostly use finite-difference or finite-element methods, and include active thermoregulatory control in addition to passive heat transfer. They predict skin temperature for several local body segments, and central core temperature. They also predict other physiological parameters, such as segment sweat rate and skin wettedness, shivering, and cardiac blood flow. Comfort is independently predicted from the output of physiological models. One comfort model, based on a review of literature addressing human sensation testing, uses an average of local skin temperatures and its time derivative to predict whole-body thermal sensation under stable and transient environments (Fiala 1998; Fiala et al. 2003). Another model uses the heat storage rate of the skin or core to predict whole-body thermal sensation under stable and transient environments (Wang 1994; Wang and Peterson 1992). The Berkeley comfort model predicts thermal sensation and comfort for each segment as well as for the whole body, using local skin temperatures, core temperature, and their time derivatives (Zhang 2003; Zhang et al. 2010a, 2010b, 2010c). The equivalent homogeneous temperature (EHT) approach uses segmented electrical manikin measurements to determine the equivalent uniform environment for each body part (Nilsson 2007; Wyon et al. 1989). From these, comfortable environmental temperature ranges have been defined for each of the body segments. The EHT can determine comfort under nonuniform environments that are at steady state. Adaptive Models Adaptive models do not actually predict comfort responses but rather the almost constant conditions under which people are likely to be comfortable in buildings. In general, people naturally adapt and may also make various adjustments to themselves and their surroundings to reduce discomfort and physiological strain. It has been observed that, through adaptive actions, an acceptable degree of comfort in residences and offices is possible over a range of air temperatures from about 17 to 31°C (Humphreys and Nicol 1998). Adaptive adjustments are typically conscious actions such as altering clothing, posture, activity schedules or levels, rate of working, diet, ventilation, air movement, and local temperature. They may also include unconscious longer-term changes to physiological set points and gains for control of shivering, skin blood flow, and sweating, as well as adjustments to body fluid levels and salt loss. However, only limited documentation and information on such changes is available. An important driving force behind the adaptive process is the pattern of outdoor weather conditions and exposure to them. This is the principal input to adaptive models, which predict likely comfort temperatures tc or ranges of tc from monthly mean outdoor temperatures tout . Humphreys and Nicol’s (1998) model is based on data from a wide range of buildings, climates, and cultures: t out – 22 tc = 24.2 + 0.43(tout – 22)exp – -------------------- 24 2 2 (81) Adaptive models are useful to guide design and energy decisions, and to specify building temperature set points throughout the year. An ASHRAE-sponsored study (de Dear and Brager 1998) on adaptive models compiled an extensive database from field studies to study, develop, and test adaptive models. For climates and buildings where cooling and central heating are not required, the study suggests the following model: toc = 17.8 + 0.31tout (82) where toc is the operative comfort temperature. The adaptive model boundary temperatures for 90% thermal acceptability are approximately toc+ 2.5°C and toc – 2.2°C according to ASHRAE Standard 55-2013. In general, the value of using an adaptive model to specify set points or guide temperature control strategies is likely to increase with the freedom that occupants are given to adapt (e.g., by having flexible working hours, locations, or dress codes). Zones of Comfort and Discomfort The Two-Node Model section shows that comfort and thermal sensation are not necessarily the same variable, especially for a person in the zone of evaporative thermal regulation. Figures 17 and 18 show this difference for the standard combination of met-clo-air movement used in the standard effective temperature ET*. Figure 17 demonstrates that practically all basic physiological variables predicted by the two-node model are functions of ambient temperature and are relatively independent of vapor pressure. All exceptions occur at relative humidities above 80% and as the isotherms reach the ET* = 41.5°C line, where regulation by evaporation fails. Figure 18 shows that lines of constant ET* and wettedness are functions of both ambient temperature and vapor pressure. Thus, human thermal responses are divided into two classes: those in Figure 17, which respond only to heat stress from the environment, and those in This file is licensed to John Murray (). Publication Date: 6/1/2017 Thermal Comfort 9.21 10. ENVIRONMENTAL INDICES An environmental index combines two or more parameters (e.g., air temperature, mean radiant temperature, humidity, air velocity) into a single variable. Indices simplify description of the thermal environment and the stress it imposes. Environmental indices may be classified according to how they are developed. Rational indices are based on the theoretical concepts presented earlier. Empirical indices are based on measurements with subjects or on simplified relationships that do not necessarily follow theory. Indices may also be classified according to their application, generally either heat stress or cold stress. Effective Temperature Licensed for single user. © 2017 ASHRAE, Inc. Fig. 17 Effect of Environmental Conditions on Physiological Variables Fig. 18 Effect of Thermal Environment on Discomfort Figure 18, which respond to both heat stress from the environment and the resultant heat strain (Stolwijk et al. 1968). For warm environments, any index with isotherms parallel to skin temperature is a reliable index of thermal sensation alone, and not of discomfort caused by increased humidity. Indices with isotherms parallel to ET* are reliable indicators of discomfort or dissatisfaction with thermal environments. For a fixed exposure time to cold, lines of constant tsk , ET*, and to are essentially identical, and cold sensation is no different from cold discomfort. For a state of comfort with sedentary or light activity, lines of constant tsk and ET* coincide. Thus, comfort and thermal sensations coincide in this region as well. The upper and lower temperature limits for comfort at these levels can be specified either by thermal sensation (Fanger 1982) or by ET*, as is done in ASHRAE Standard 55, because lines of constant comfort and lines of co0nstant thermal sensation should be identical. Effective temperature ET* is probably the most common environmental index, and has the widest range of application. It combines temperature and humidity into a single index, so two environments with the same ET* should evoke the same thermal response even though they have different temperatures and humidities, as long as they have the same air velocities. The original empirical effective temperature was developed by Houghten and Yaglou (1923). Gagge et al. (1971a, 1971b) defined a new effective temperature using a rational approach. Defined mathematically in Equation (33), this is the temperature of an environment at 50% rh that results in the same total heat loss Esk from the skin as in the actual environment. Because the index is defined in terms of operative temperature to , it combines the effects of three parameters (tr , ta, and pa) into a single index. Skin wettedness w and the permeability index im must be specified and are constant for a given ET* line for a particular situation. The two-node model is used to determine skin wettedness in the zone of evaporative regulation. At the upper limit of regulation, w approaches 1.0; at the lower limit, w approaches 0.06. Skin wettedness equals one of these values when the body is outside the zone of evaporative regulation. Because the slope of a constant ET* line depends on skin wettedness and clothing moisture permeability, effective temperature for a given temperature and humidity may depend on the person’s clothing and activity. This difference is shown in Figure 19. At low skin wettedness, air humidity has little influence, and lines of constant ET* are nearly vertical. As skin wettedness increases because of activity and/or heat stress, the lines become more horizontal and the influence of humidity is much more pronounced. The ASHRAE comfort envelope shown in Figure 5 is described in terms of ET*. Because ET* depends on clothing and activity, it is not possible to generate a universal ET* chart. A standard set of conditions representative of typical indoor applications is used to define a standard effective temperature SET*, defined as the equivalent air temperature of an isothermal environment at 50% rh in which a subject, wearing clothing standardized for the activity concerned, has the same heat stress (skin temperature tsk) and thermoregulatory strain (skin wettedness w) as in the actual environment. Humid Operative Temperature The humid operative temperature toh is the temperature of a uniform environment at 100% rh in which a person loses the same total amount of heat from the skin as in the actual environment. This index is defined mathematically in Equation (32). It is analogous to ET*, except that it is defined at 100% rh and 0% rh rather than at 50% rh. Figures 2 and 19 indicate that lines of constant ET* are also lines of constant toh . However, the values of these two indices differ for a given environment. Heat Stress Index Originally proposed by Belding and Hatch (1955), this rational index is the ratio of total evaporative heat loss Esk required for This file is licensed to John Murray (). Publication Date: 6/1/2017 Licensed for single user. © 2017 ASHRAE, Inc. 9.22 2017 ASHRAE Handbook—Fundamentals (SI) Fig. 19 Effective Temperature ET* and Skin Wettedness w [Adapted from Gonzalez et al. (1978) and Nishi et al. (1975)] Table 12 Evaluation of Heat Stress Index Heat Stress Index 0 10 20 30 40 50 60 70 80 90 100 Physiological and Hygienic Implications of 8 h Exposures to Various Heat Stresses No thermal strain. Mild to moderate heat strain. If job involves higher intellectual functions, dexterity, or alertness, subtle to substantial decrements in performance may be expected. In performing heavy physical work, little decrement is expected, unless ability of individuals to perform such work under no thermal stress is marginal. Severe heat strain involving a threat to health unless workers are physically fit. Break-in period required for men not previously acclimatized. Some decrement in performance of physical work is to be expected. Medical selection of personnel desirable, because these conditions are unsuitable for those with cardiovascular or respiratory impairment or with chronic dermatitis. These working conditions are also unsuitable for activities requiring sustained mental effort. Very severe heat strain. Only a small percentage of the population may be expected to qualify for this work. Personnel should be selected (a) by medical examination, and (b) by trial on the job (after acclimatization). Special measures are needed to ensure adequate water and salt intake. Amelioration of working conditions by any feasible means is highly desirable, and may be expected to decrease the health hazard while increasing job efficiency. Slight “indisposition,” which in most jobs would be insufficient to affect performance, may render workers unfit for this exposure. The maximum strain tolerated daily by fit, acclimatized young men. thermal equilibrium (the sum of metabolism plus dry heat load) to maximum evaporative heat loss Emax possible for the environment, multiplied by 100, for steady-state conditions (Ssk and Scr are zero) and with tsk held constant at 35°C. The ratio Esk /Emax equals skin wettedness w [Equation (18)]. When heat stress index (HSI) > 100, body heating occurs; when HSI < 0, body cooling occurs. Belding and Hatch (1955) limited Emax to 700 W/m2, which corresponds to a sweat rate of approximately 280 mg/(s·m2). When tsk is constant, loci of constant HSI coincide with lines of constant ET* on a psychrometric chart. Other indices based on wettedness have the same applications (Belding 1970; Gonzalez et al. 1978; ISO Standard 7933) but differ in their treatment of Emax and the effect of clothing. Table 12 describes physiological factors associated with HSI values. Index of Skin Wettedness Skin wettedness w is the ratio of observed skin sweating Esk to the Emax of the environment as defined by tsk , ta , humidity, air movement, and clothing in Equation (12). Except for the factor of 100, it is essentially the same as HSI. Skin wettedness is more closely related to the sense of discomfort or unpleasantness than to temperature sensation (Gagge et al. 1969a, 1969b; Gonzalez et al. 1978). Wet-Bulb Globe Temperature The WBGT is an environmental heat stress index that combines dry-bulb temperature tdb , a naturally ventilated (not aspirated) wet-bulb temperature tnwb , and black globe temperature tg , according to the relation (Dukes-Dobos and Henschel 1971, 1973) WBGT = 0.7tnwb + 0.2tg + 0.1ta (83) This form of the equation is usually used where solar radiation is present. The naturally ventilated wet-bulb thermometer is left exposed to sunlight, but the air temperature ta sensor is shaded. In enclosed environments, Equation (83) is simplified by dropping the ta term and using a 0.3 weighting factor for tg . The black globe thermometer responds to air temperature, mean radiant temperature, and air movement, whereas the naturally ventilated wet-bulb thermometer responds to air humidity, air movement, radiant temperature, and air temperature. Thus, WBGT is a function of all four environmental factors affecting human environmental heat stress. The WBGT index is widely used for estimating the heat stress potential of industrial environments (Davis 1976). In the United States, the National Institute of Occupational Safety and Health (NIOSH) developed criteria for a heat-stress-limiting standard This file is licensed to John Murray (). Publication Date: 6/1/2017 Thermal Comfort 9.23 WBGT = 1.044(WGT) – 0.187 (84) This equation should not be used outside the experimental range given because data from hot/dry desert environments show differences between WBGT and WGT that are too large (6 K and above) to be adjusted by Equation (84) (Matthew et al. 1986). At very low humidity and high wind, WGT approaches the psychrometric wetbulb temperature, which is greatly depressed below ta . However, in the WBGT, tnwb accounts for only 70% of the index value, with the remaining 30% at or above ta . Wind Chill Index The wind chill index (WCI) is an empirical index developed from cooling measurements obtained in Antarctica on a cylindrical flask partly filled with water (Siple and Passel 1945). The index describes the rate of heat loss from the cylinder by radiation and convection for a surface temperature of 33°C, as a function of ambient temperature and wind velocity. As originally proposed, Licensed for single user. © 2017 ASHRAE, Inc. WCI = (10.45 + 10 V – V)(33 – ta) in kcal/(h·m2) Fig. 20 Recommended Heat Stress Exposure Limits for Heat Acclimatized Workers [Adapted from NIOSH (1986)] (NIOSH 1986). ISO Standard 7243 also uses the WBGT. Figure 20 summarizes permissible heat exposure limits, expressed as working time per hour, for a fit individual, as specified for various WBGT levels. Values apply for normal permeable clothing (0.6 clo) and must be adjusted for heavy or partly vapor-permeable clothing. For example, the U.S. Air Force (USAF) recommended adjusting the measured WBGT upwards by 6 K for personnel wearing chemical protective clothing or body armor. This type of clothing increases resistance to sweat evaporation about threefold (higher if it is totally impermeable), requiring an adjustment in WBGT level to compensate for reduced evaporative cooling at the skin. Several mathematical models are available for predicting WBGT from environmental factors: air temperature, psychrometric wetbulb temperature, mean radiant temperature, and air motion (Azer and Hsu 1977; Sullivan and Gorton 1976). Wet-Globe Temperature The WGT, introduced by Botsford (1971), is a simpler approach to measuring environmental heat stress than the WBGT. The measurement is made with a wetted globe thermometer called a Botsball, which consists of a 65 mm black copper sphere covered with a fitted wet black mesh fabric, into which the sensor of a dial thermometer is inserted. A polished stem attached to the sphere supports the thermometer and contains a water reservoir for keeping the sphere covering wet. This instrument is suspended by the stem at the site to be measured. Onkaram et al. (1980) showed that WBGT can be predicted with reasonable accuracy from WGT for temperate to warm environments with medium to high humidities. With air temperatures between 20 and 35°C, dew points from 7 to 25°C (relative humidities above 30%), and wind speeds of 7 m/s or less, the experimental regression equation (r = 0.98) in °C for an outdoor environment is (85) where V and ta are in m/s and °C, respectively. Multiply WCI by 1.163 to convert to W/m2. The 33°C surface temperature was chosen to be representative of the mean skin temperature of a resting human in comfortable surroundings. Some valid objections have been raised about this formulation. Cooling rate data from which it was derived were measured on a 57 mm diameter plastic cylinder, making it unlikely that WCI would be an accurate measure of heat loss from exposed flesh, which has different characteristics from plastic (curvature, roughness, and radiation exchange properties) and is invariably below 33°C in a cold environment. Moreover, values given by the equation peak at 90 km/h, then decrease with increasing velocity. Nevertheless, for velocities below 80 km/h, this index reliably expresses combined effects of temperature and wind on subjective discomfort. For example, if the calculated WCI is less than 1400 kcal/(h·m2) and actual air temperature is above 10°C, there is little risk of frostbite during brief exposures (1 h or less), even for bare skin. However, at a WCI of 2000 or more, the probability is high that exposed flesh will begin to freeze in 1 min or less unless measures are taken to shield exposed skin (such as a fur ruff to break up wind around the face). Rather than using the WCI to express the severity of a cold environment, meteorologists use an index derived from the WCI called the equivalent wind chill temperature teq,wc . This is the ambient temperature that would produce, in a calm wind (defined for this application as 6.4 km/h), the same WCI as the actual combination of air temperature and wind velocity: teq,wc = –0.0528(WCI) + 33 (86) where teq,wc is in °C (and frequently referred to as a wind chill factor), thus distinguishing it from WCI, which is given either as a cooling rate or as a plain number with no units. For velocities less than 6.4 km/h (1.8 m/s), Equation (86) does not apply, and the wind chill temperature is equal to the air temperature. Equation (86) does not imply cooling to below ambient temperature, but recognizes that, because of wind, the cooling rate is increased as though it were occurring at the lower equivalent wind chill temperature. Wind accelerates the rate of heat loss, so that the skin surface cools more quickly toward the ambient temperature. Table 13 shows a typical wind chill chart, expressed in equivalent wind chill temperature. This file is licensed to John Murray (). Publication Date: 6/1/2017 9.24 2017 ASHRAE Handbook—Fundamentals (SI) Table 13 Equivalent Wind Chill Temperatures of Cold Environments Actual Thermometer Reading, °C Wind Speed, km/h 10 5 0 –5 –10 –15 –20 –25 Calm 10 20 30 40 50 60 70 10 8 3 1 –1 –2 –3 –4 5 2 –3 –6 –8 –10 –11 –12 0 –3 –10 –13 –16 –18 –19 –20 –5 –9 –16 –20 –23 –25 –27 –28 –10 –14 –23 –27 –31 –33 –35 –35 –15 –20 –29 –34 –38 –41 –42 –43 –20 –25 –35 –42 –46 –48 –50 –51 –25 –31 –42 –49 –53 –56 –58 –59 –35 –40 –45 –50 –30 –37 –48 –56 –60 –64 –66 –67 –35 –42 –55 –63 –68 –71 –74 –75 –40 –48 –61 –70 –75 –79 –82 –83 –45 –53 –68 –77 –83 –87 –90 –91 –50 –59 –74 –84 –90 –94 –97 –99 Little danger: In less than 5 h, with dry skin. Maximum danger from false sense of security. Great danger: Flesh may freeze within 30 s. Increasing danger: Danger of freezing exposed flesh within 1 min. (WCI < 1400) (1400 WCI 2000) (WCI > 2000) Source: U.S. Army Research Institute of Environmental Medicine. Notes: Cooling power of environment expressed as an equivalent temperature under calm conditions [Equation (86)]. 11. Licensed for single user. © 2017 ASHRAE, Inc. –30 Equivalent Wind Chill Temperature, °C Winds greater than 70 km/h have little added chilling effect. SPECIAL ENVIRONMENTS Infrared Heating Optical and thermal properties of skin must be considered in studies of the effects of infrared radiation in (1) producing changes in skin temperature and skin blood flow, and (2) evoking sensations of temperature and comfort (Hardy 1961). Although the body can be considered to have the properties of water, thermal sensation and heat transfer with the environment require a study of the skin and its interaction with visible and infrared radiation. Figure 21 shows how skin reflectance and absorptance vary for a blackbody heat source at the temperature (in K) indicated. These curves show that darkly pigmented skin is heated more by direct radiation from a high-intensity heater at 2500 K than is lightly pigmented skin. With low-temperature, low-intensity heating equipment used for total area heating, there is minimal, if any, difference. Also, in practice, clothing minimizes differences. Changes in skin temperature caused by high-intensity infrared radiation depend on the thermal conductivity, density, and specific heat of living skin (Lipkin and Hardy 1954). Modeling skin heating with the heat transfer theory yields a parabolic relation between exposure time and skin temperature rise for nonpenetrating radiation: tsf – tsi = t = 2J kc p Fig. 21 Variation in Skin Reflection and Absorptivity for Blackbody Heat Sources (87) where tsf tsi J k cp = = = = = = = = final skin temperature, °C initial skin temperature, °C irradiance from source radiation temperatures, W/m2 skin absorptance at radiation temperatures, dimensionless time, h specific thermal conductivity of tissue, W/(m·K) density, kg/m3 specific heat, J/(kg·K) Product kcp is the physiologically important quantity that determines temperature elevation of skin or other tissue on exposure to nonpenetrating radiation. Fatty tissue, because of its relatively low specific heat, is heated more rapidly than moist skin or bone. Experimentally, kcp values can be determined by plotting t 2 against 1.13J 2 (Figure 22). The relationship is linear, and the slopes are inversely proportional to the kcp of the specimen. Comparing leather and water with body tissues suggests that thermal inertia values depend largely on tissue water content. Fig. 22 Comparing Thermal Inertia of Fat, Bone, Moist Muscle, and Excised Skin to That of Leather and Water Living tissues do not conform strictly to this simple mathematical formula. Figure 23 compares excised skin with living skin with normal blood flow, and skin with blood flow occluded. For short exposure times, the kcp of normal skin is the same as that in which blood flow has been stopped; excised skin heats more rapidly because of unavoidable dehydration that occurs postmortem. This file is licensed to John Murray (). Publication Date: 6/1/2017 Thermal Comfort 9.25 preliminary analysis, a useful value for hr is 4.7 W/(m2 ·K), which corresponds to a normally clothed (at 24°C) sedentary subject. Ambient air movement affects hc , which appears only in the righthand term of Equation (89). Although the linear radiation coefficient hr is used in Equations (88) and (89), the same definition of ERF follows if the fourth power radiation law is used. By this law, assuming emissivity of the body surface is unity, the ERF term in Equation (89) is ERF = (Ar /AD)[(tr – 273)4 – (ta + 273)4]Fcle (90) where is the Stefan-Boltzmann constant, 5.67 10–8 W/(m2 ·K4). Because tr equals the radiation of several surfaces at different temperatures (T1, T2, . . . , Tj), ERF = (ERF)1 + (ERF)2 + ... + (ERF)j Licensed for single user. © 2017 ASHRAE, Inc. Fig. 23 Thermal Inertias of Excised, Bloodless, and Normal Living Skin where However, with longer exposure to thermal radiation, vasodilation increases blood flow, cooling the skin. For the first 20 s of irradiation, skin with normally constricted blood vessels has a kcp value one-fourth that for skin with fully dilated vessels. Skin temperature is the best single index of thermal comfort. The most rapid changes in skin temperature occur during the first 60 s of exposure to infrared radiation. During this initial period, thermal sensation and the heating rate of the skin vary with the quality of infrared radiation (color temperature in K). Because radiant heat from a gas-fired heater is absorbed at the skin surface, the same unit level of absorbed radiation during the first 60 s of exposure can cause an even warmer initial sensation than penetrating solar radiation. Skin heating curves tend to level off after a 60 s exposure (Figure 23), which means that a relative balance is quickly created between heat absorbed, heat flow to the skin surface, and heat loss to the ambient environment. Therefore, the effects of radiant heating on thermal comfort should be examined for conditions approaching thermal equilibrium. Stolwijk and Hardy (1966) described an unclothed subject’s response for a 2 h exposure to temperatures of 5 to 35°C. Nevins et al. (1966) showed a relation between ambient temperatures and thermal comfort of clothed, resting subjects. For any given uniform environmental temperature, both initial physiological response and degree of comfort can be determined for a subject at rest. Physiological implications for radiant heating can be defined by two environmental temperatures: (1) mean radiant temperature tr and (2) ambient air temperature ta . For this discussion on radiant heat, assume that (1) relative humidity is less than 50%, and (2) air movement is low and constant, with an equivalent convection coefficient of 2.9 W/(m2 ·K). The equilibrium equation, describing heat exchange between skin surface at mean temperature tsk and the radiant environment, is given in Equation (28), and can be transformed to give (see Table 2) M – Esk – Fcle[hr(tsk – tr) + hc(tsk – to] = 0 (88) where M is the net heat production (M W) less respiratory losses. By algebraic transformation, Equation (88) can be rewritten as M + ERF Fcle = Esk + (hr + hc)(tsk – ta)Fcle (91) (89) where ERF = hr(tr – ta) is the effective radiant field and represents the additional radiant exchange with the body when tr ta. The last term in Equation (89) describes heat exchange with an environment uniformly heated to temperature ta . The term hr , evaluated in Equation (35), is also a function of posture, for which factor Ar /AD can vary from 0.67 for crouching to 0.73 for standing. For ERFj = (Ar /AD)jFm–j T j4 – Ta4 Fcle j = absorptance of skin or clothing surface for source radiating at temperature Tj Fm j = angle factor to subject m from source j Ta = ambient air temperature, K ERF is the sum of the fields caused by each surface Tj [e.g., T1 may be an infrared beam heater; T2, a heated floor; T3, a warm ceiling; T4, a cold plate glass window (T4 < Ta); etc.]. Only surfaces with Tj differing from Ta contribute to the ERF. Comfort Equations for Radiant Heating The comfort equation for radiant heat (Gagge et al. 1967a, 1967b) follows from definition of ERF and Equation (8): to (for comfort) = ta + ERF (for comfort)/h (92) Thus, operative temperature for comfort is the temperature of the ambient air plus a temperature increment ERF/h, a ratio that measures the effectiveness of the incident radiant heating on occupants. Higher air movement (which increases the value of h or hc) reduces the effectiveness of radiant heating systems. Clothing lowers to for comfort and for thermal neutrality. Values for ERF and h must be determined to apply the comfort equation for radiant heating. Table 3 may be used to estimate h. One method of determining ERF is to calculate it directly from radiometric data that give (1) radiation emission spectrum of the source, (2) concentration of the beam, (3) radiation from the floor, ceiling, and windows, and (4) corresponding angle factors involved. This analytical approach is described in Chapter 54 of the 2015 ASHRAE Handbook—HVAC Applications. For direct measurement, a black globe, 150 mm in diameter, can measure the radiant field ERF for comfort, by the following relation: ERF = (Ar /AD)(6.1 + 13.6 V)(tg – ta) (93) where tg is uncorrected globe temperature in °C and V is air movement in m/s. The average value of Ar /AD is 0.7. For a black globe, ERF must be multiplied by for the exposed clothing/skin surface. For a subject with 0.6 to 1.0 clo, to for comfort should agree numerically with ta for comfort in Figure 5. When to replaces ta in Figure 5, humidity is measured in vapor pressure rather than relative humidity, which refers only to air temperature. Other methods may be used to measure ERF. The most accurate is by physiological means. In Equation (89), when M, tsk ta , and the associated transfer coefficients are experimentally held constant, This file is licensed to John Murray (). Publication Date: 6/1/2017 9.26 2017 ASHRAE Handbook—Fundamentals (SI) E = ERF (94) The variation in evaporative heat loss E (rate of mass loss) caused by changing the wattage of two T-3 infrared lamps is a measure in absolute terms of the radiant heat received by the body. A third method uses a directional radiometer to measure ERF directly. For example, radiation absorbed at the body surface (in W/m2) is ERF = (Ai /AD)J (95) where irradiance J can be measured by a directional (Hardy-type) radiometer, is the surface absorptance effective for the source used, and Ai is the projection area of the body normal to the directional irradiance. Equation (95) can be used to calculate ERF only for the simplest geometrical arrangements. For a human subject lying supine and irradiated uniformly from above, Ai /AD is 0.3. Figure 21 shows variance of for human skin with blackbody temperature (in K) of the radiating source. When irradiance J is uneven and coming from many directions, as is usually the case, the previous physiological method can be used to obtain an effective Ai /AD from the observed E and (J). Licensed for single user. © 2017 ASHRAE, Inc. Personal Environmental Control (PEC) Systems Because of the large interpersonal variability in thermal requirements, some occupants in any uniformly conditioned environment will be too warm at the same time as others are too cool. The ASHRAE 80% acceptability criterion reflects this physiological constraint. Only environments that respond to individual preferences are capable of thermally satisfying all occupants (Bauman et al. 1998). Such occupant-specific microenvironments may be conditioned with low energy input because their aggregate volume is smaller than the total space volume, and because heating or cooling the occupants themselves may be more energy efficient than space conditioning. Such designs require attention to the thermal sensitivities of different parts of the human body and to the physical properties of its microenvironment. In warm conditions, the comfort of the head and hands dictates a person’s overall discomfort; in cool conditions, the feet and hands dictate overall discomfort (Arens et al. 2006; Zhang 2003). Keeping the feet and hands warm is necessary to prevent discomfort from vasoconstriction in the limbs. However in warm conditions, the hands and wrists are important heat dissipaters, and cooling them is important. Arens et al. (2006) and Zhang (2003) suggest that a personal environmental control (PEC) system, also called task-ambient conditioning (TAC) or personal ventilation (PV) systems, that focuses directly on these body parts may offer an energy-efficient means for improving comfort in office environments. PEC fan systems using either recirculated room air or outdoor air can provide comfort and improve perceived air quality (Amai et al. 2007; Arens et al. 2008, 2011; Dygert and Dang 2011; Melikov 2003; Russo and Khalifa 2011; Sekhar et al. 2005; Tham and Willem 2004; Yang et al. 2009, 2010; Zhang et al. 2010d). Air quality can also be improved, because fan flows above 0.3 m/s disrupt the body’s thermal plume that carries pollutants upward to the breathing zone (Arens et al. 2008, 2011). Using air movement for cooling has constraints. Strong airflow directed at the eyes might cause dry-eye discomfort and should be avoided (Melikov et al. 2011). However, a large percentage of office occupants in neutral and warm conditions prefer an increase in available air movement (Arens et al. 2009). A recent study of hemoglobin levels showed that air movement also reduces fatigue (Nishihara and Tanabe 2011; Tanabe and Nishihara 2004). Foot heating is usually done by radiant heating or through contact with a heated surface. Efficiency of these systems depends greatly on confining the heating to the body surfaces without too much loss to the surrounding air. Fig. 24 Recommended Temperature Set Points for HVAC with PEC Systems and Energy Savings from Extending HVAC Temperature Set Points [Based on Hoyt et al. (2009) and Zhang et al. (2011)] Hands and wrists may both be heated and cooled by contact with conductive surfaces. Wrist cooling may not require actively cooled surfaces, because the skin is almost always at a higher temperature than surfaces in a normal environment. Some researchers suggest that a PEC system can be part of an energy-saving strategy (Hoyt et al. 2009; Zhang et al. 2011) by keeping occupants comfortable while allowing the surrounding spaces to be less conditioned (Figure 24). The success of this strategy also depends on the length of time occupants are away from the PEC zone. Once steady state is reached, the change of sensations when moving from a comfortable environment to one less comfortable is much slower than the change on returning to comfortable conditions (Zhang et al. 2010a). For example, 10-min excursions climbing stairs were judged comfortable throughout, despite a 28°C stairwell temperature, whereas 15-min excursions climbing stairs became uncomfortable; occupants judged their status comfortable/ acceptable within 30 s of returning to the PEC zone. Hot and Humid Environments Tolerance limits to high temperature vary with the ability to (1) sense temperature, (2) lose heat by regulatory sweating, and (3) move heat from the body core by blood flow to the skin surface, where cooling is the most effective. many interrelating processes are involved in heat stress (Figure 25). Skin surface temperatures of 46°C trigger pain receptors in the skin; direct contact with metal at this temperature is painful. However, because thermal insulation of the air layer around the skin is high, much higher dry-air temperatures can be tolerated (e.g., 85°C for brief periods in a sauna). For lightly clothed subjects at rest, tolerance times of nearly 50 min have been reported at 82°C db; 33 min at 93°C; 26 min at 104°C; and 24 min at 115°C. In each case, dew points were lower than 30°C. Short exposures to these extremely hot environments are tolerable because of cooling by sweat evaporation. However, when ambient vapor pressure approaches 6.0 kPa (36°C dp, typically found on sweating skin), tolerance is drastically reduced. Temperatures of 50°C can be intolerable if the dew-point temperature is greater than 25°C, and both deep This file is licensed to John Murray (). Publication Date: 6/1/2017 Thermal Comfort 9.27 Fig. 25 Schematic Design of Heat Stress and Heat Disorders Licensed for single user. © 2017 ASHRAE, Inc. [Modified by Buskirk (1960) from scale diagram by Belding (1967) and Leithead and Lind (1964)] body temperature and heart rate rise within minutes (Gonzalez et al. 1978). The rate at which and length of time a body can sweat are limited. The maximum rate of sweating for an average man is about 0.5 g/s. If all this sweat evaporates from the skin surface under conditions of low humidity and air movement, maximum cooling is about 675 W/m2. However, because sweat rolls off the skin surface without evaporative cooling or is absorbed by or evaporated within clothing, a more typical cooling limit is 6 met (350 W/m2), representing approximately 0.3 g/s (1 L/h) of sweating for the average man. Thermal equilibrium is maintained by dissipation of resting heat production (1 met) plus any radiant and convective load. If the environment does not limit heat loss from the body during heavy activity, decreasing skin temperature compensates for the core temperature rise. Therefore, mean body temperature is maintained, although the gradient from core to skin is increased. Blood flow through the skin is reduced, but muscle blood flow necessary for exercise is preserved. The upper limit of skin blood flow is about 25 g/s (Burton and Bazett 1936). Body heat storage of 335 kJ (or a rise in tb of 1.4 K) for an average-sized man represents an average voluntary tolerance limit. Continuing work beyond this limit increases the risk of heat exhaustion. Collapse can occur at about 670 kJ of storage (2.8 K rise); few individuals can tolerate heat storage of 920 kJ (3.8 K above normal). The cardiovascular system affects tolerance limits. In normal, healthy subjects exposed to extreme heat, heart rate and cardiac output increase in an attempt to maintain blood pressure and supply of blood to the brain. At a heart rate of about 180 bpm, the short time between contractions prevents adequate blood supply to the heart chambers. As heart rate continues to increase, cardiac output drops, causing inadequate convective blood exchange with the skin and, perhaps more important, inadequate blood supply to the brain. Victims of this heat exhaustion faint or black out. Accelerated heart rate can also result from inadequate venous return to the heart caused by blood pooling in the skin and lower extremities. In this case, cardiac output is limited because not enough blood is available to refill the heart between beats. This occurs most frequently when an overheated individual, having worked hard in the heat, suddenly stops working. The muscles no longer massage the blood back past the valves in the veins toward the heart. Dehydration compounds the problem by reducing fluid volume in the vascular system. If core temperature tcr increases above 41°C, critical hypothalamic proteins can be damaged, resulting in inappropriate vasoconstriction, cessation of sweating, increased heat production by shivering, or some combination of these. Heat stroke damage is frequently irreversible and carries a high risk of death. Another problem, hyperventilation, occurs mainly in hot/wet conditions, when too much CO2 is washed from the blood. This can lead to tingling sensations, skin numbness, and vasoconstriction in the brain with occasional loss of consciousness. Because a rise in heart rate or rectal temperature is essentially linear with ambient vapor pressure above a dew point of 25°C, these two changes can measure severe heat stress. Although individual heart rate and rectal temperature responses to mild heat stress vary, severe heat stress saturates physiological regulating systems, producing uniform increases in heart rate and rectal temperature. In contrast, sweat production measures stress under milder conditions but becomes less useful under more severe stress. The maximal sweat rate compatible with body cooling varies with (1) degree of heat acclimatization, (2) duration of sweating, and (3) whether the sweat evaporates or merely saturates the skin and drips off. Total sweat rates over 2 L/h can occur in short exposures, but about 1 L/h is an average maximum sustainable level for an acclimatized man. Figure 26 shows the decline in heart rate, rectal temperature, and skin temperature when exercising subjects are exposed to 40°C over a period of days. Acclimatization can be achieved by working in the heat for 100 min each day: 30% improvement occurs after the first day, 50% after 3 days, and 95% after 6 or 7 days. Increased sweat secretion while working in the heat can be induced by rest. Although reducing salt intake during the first few days in the heat can conserve sodium, heat cramps may result. Working regularly in the heat improves cardiovascular efficiency, sweat secretion, and sodium conservation. Once induced, heat acclimatization can be maintained by as little as one workout a week in the heat; otherwise, it diminishes slowly over a 2- to 3-week period and disappears. Extremely Cold Environments Human performance in extreme cold ultimately depends on maintaining thermal balance. Subjective discomfort is reported by a 70 kg man with 1.8 m2 of body surface area when a heat debt of about 104 kJ is incurred. A heat debt of about 630 kJ is acutely This file is licensed to John Murray (). Publication Date: 6/1/2017 Licensed for single user. © 2017 ASHRAE, Inc. 9.28 2017 ASHRAE Handbook—Fundamentals (SI) Fig. 26 Acclimatization to Heat Resulting from Daily Exposure of Five Subjects to Extremely Hot Room (Robinson et al. 1943) uncomfortable; this represents a drop of approximately 2.6 K (or about 7% of total heat content) in mean body temperature. This loss can occur during 1 to 2 h of sedentary activity outdoors. A sleeping individual will wake after losing about 314 kJ, decreasing mean skin temperature by about 3 K and deep body temperature by about 0.5 K. A drop in deep body temperature (e.g., rectal temperature) below 35°C threatens a loss of body temperature regulation, and 28°C is considered critical for survival, despite recorded survival from a deep body temperature of 18°C. Activity level also affects human performance. Subjective sensations reported by sedentary subjects at a mean skin temperature of 33.3°C are comfortable; at 31°C, uncomfortably cold; at 30°C, shivering cold; and at 29°C, extremely cold. The critical subjective tolerance limit (without numbing) for mean skin temperature appears to be about 25°C. However, during moderate to heavy activity, subjects reported the same skin temperatures as comfortable. Although mean skin temperature is significant, the temperature of the extremities is more frequently the critical factor for comfort in the cold. Consistent with this, one of the first responses to cold exposure is vasoconstriction, which reduces circulatory heat input to the hands and feet. A hand-skin temperature of 20°C causes a report of uncomfortably cold; 15°C, extremely cold; and 5°C, painful. Identical verbal responses for the foot surface occur at approximately 1.5 to 2 K warmer temperatures. An ambient temperature of 35°C is the lower limit for useful outdoor activity, even with adequate insulative clothing. At 50°C, almost all outdoor effort becomes exceedingly difficult; even with appropriate protective equipment, only limited exposure is possible. Reported exposures of 30 min at 75°C have occurred in the Antarctic without injury. In response to extreme heat loss, maximal heat production becomes very important. When the less-efficient vasoconstriction cannot prevent body heat loss, shivering is an automatic, more efficient defense against cold. This can be triggered by low deep body temperature, low skin temperature, rapid change of skin temperature, or some combination of all three. Shivering is usually preceded by an imperceptible increase in muscle tension and by noticeable gooseflesh produced by muscle contraction in the skin. It begins slowly in small muscle groups, initially increasing total heat production by 1.5 to 2 times resting levels. As body cooling increases, the reaction spreads to additional body segments. Ultimately violent, wholebody shivering causes maximum heat production of about 6 times resting levels, rendering the individual totally ineffective. Given sufficient cold exposure, the body undergoes changes that indicate cold acclimatization. These physiological changes include (1) endocrine changes (e.g., sensitivity to norepinephrine), causing nonshivering heat production by metabolism of free fatty acids released from adipose tissue; (2) improved circulatory heat flow to skin, causing an overall sensation of greater comfort; and (3) improved circulatory heat flow to the extremities, reducing the risk of injury and allowing activities at what ordinarily would be severely uncomfortable temperatures in the extremities. Generally, these physiological changes are minor and are induced only by repeated extreme exposures. Nonphysiological factors, including training, experience, and selection of adequate protective clothing, are more useful and may be safer than dependence on physiological changes. Food energy intake requirements for adequately clothed subjects in extreme cold are only slightly greater than those for subjects living and working in temperate climates. This greater requirement results from added work caused by (1) carrying the weight of heavy clothing (energy cost for heavy protective footwear may be six times that of an equivalent weight on the torso); and (2) the inefficiency of walking in snow, snowshoeing, or skiing, which can increase energy cost up to 300%. To achieve proper protection in low temperatures, a person must either maintain high metabolic heat production by activity or reduce heat loss by controlling the body’s microclimate with clothing. Other protective measures include spot radiant heating, showers of hot air for work at a fixed site, and warm-air-ventilated or electrically heated clothing. Extremities (e.g., fingers and toes) are at greater risk than the torso because, as thin cylinders, they are particularly susceptible to heat loss and difficult to insulate without increasing the surface for heat loss. Vasoconstriction can reduce circulatory heat input to extremities by over 90%. Although there is no ideal insulating material for protective clothing, radiation-reflective materials are promising. Insulation is primarily a function of clothing thickness; the thickness of trapped air, rather than fibers used, determines insulation effectiveness. Protection for the respiratory tract seems unnecessary in healthy individuals, even at 45°C. However, asthmatics or individuals with mild cardiovascular problems may benefit from a face mask that warms inspired air. Masks are unnecessary for protecting the face because heat to facial skin is not reduced by local vasoconstriction, as it is for hands. If wind chill is great, there is always a risk of cold injury caused by freezing of exposed skin. Using properly designed torso clothing, such as a parka with a fur-lined hood to minimize wind penetration to the face, and 10 W of auxiliary heat to each hand and foot, inactive people can tolerate 55°C with a 16 km/h wind for more than 6 h. As long as the skin temperature of fingers remains This file is licensed to John Murray (). Publication Date: 6/1/2017 Thermal Comfort 9.29 above 15°C, manual dexterity can be maintained and useful work performed without difficulty. Licensed for single user. © 2017 ASHRAE, Inc. 12. A BFN c cdil csw C C+R DISC E ERF ET* fcl F h HSI i I J k K Kres l L LR m m· M p PD PMV PPD q Q = R= RQ S SET* SkBF t tr T TSENS Tu V Vsd = = = = = = = = = = = w= W= WBGT WCI WGT xf = = = = SYMBOLS area, m2 neutral skin blood flow, g/(m2 ·s) specific heat, J/(kg· K) specific heat (constant) for skin blood flow proportionality constant for sweat control, 170 W/(m2 ·K) convective heat loss, W/m2 total sensible heat loss from skin, W/m2 thermal discomfort evaporative heat loss, W/m2 effective radiant field, W/m2 effective temperature based on 50% rh, °C clothing area factor, Acl /AD , dimensionless thermal efficiency, or angle factor enthalpy, kJ/kg (dry air), or heat transfer coefficient, W/(m2 ·K) heat stress index vapor permeation efficiency, dimensionless thermal resistance in clo units, clo irradiance, W/m 2 thermal conductivity of body tissue, W/(m·K effective conductance between core and skin, W/(m2 ·K) proportionality constant, 1.43 kg/kJ height, m thermal load on body, W/m 2 Lewis ratio, K/kPa mass, kg mass flow, kg/(s·m2) metabolic heat production, W/m2 water vapor pressure, kPa percent dissatisfied predicted mean vote predicted percent dissatisfied heat flow, W/m2 volume rate, m3/h, or volume flow rate of blood per unit surface area, L/(h·m2) thermal resistance, (m2 ·K)/W, or radiative heat loss from skin, W/m2 respiratory quotient, dimensionless heat storage, W/m2 standard effective temperature, °C skin blood flow, g/(m2·s) temperature, °C mean temperature, °C absolute temperature, K thermal sensation turbulence

intensity, % air velocity, m/s standard deviation of velocity measured with omnidirectional anemometer with 0.2 s time constant skin wettedness, dimensionless external work accomplished, W/m2, or humidity ratio of air, kg (water vapor)/kg (dry air) wet-bulb globe temperature, °C wind chill index, W/m2 wet-globe temperature, °C fabric thickness, mm Greek = skin absorptance, dimensionless = emissivity, dimensionless ev = evaporative efficiency, dimensionless = time, s = mechanical efficiency of body = W/M, dimensionless T = mean space temperature, °C = unsolicited thermal complaint rate, complaints/(h·zone area) = density, kg/m3 bl = density of blood, 1.06 kg/L = Stefan-Boltzmann constant = 5.67 10–8 W/(m2 ·K4) T = standard deviation of space temperature, K T·H,T·L = standard deviation of rate of change of high and low space temperature, K/h Superscripts and Subscripts a act b B b,c b,h bl c cc ch cl cle clu,i com cr cr,sk D db dif dil dp dry e ec eq,wc evap ex fg g G ge h l m max m–j N nwb o oc oh out p pcl p–N pr r req res rsw s sf shiv si sk sw t tr wb w,res = overall, net ambient air activity of body tissue building lower limit for evaporative regulation zone upper limit for evaporative regulation zone of blood convection, or comfort corrected convection value between chair and body of clothed body or clothing of clothing, effective effective insulation of garment i combined body core from core to skin DuBois value dry bulb due to moisture diffusion through skin skin blood flow dew point sensible evaporative, at surface at surface, corrected equivalent wind chill latent exhaled air vaporization of water globe covered by garment gas exchange too hot too cold total maximum from person to source j of surface N naturally ventilated wet bulb operative operative comfort humid operation monthly mean outside at constant pressure permeation between person and source N plane radiant radiation, radiant required respiration regulatory sweat saturated final skin shivering initial skin skin sweat atmospheric, or total constriction constant for skin blood flow wet bulb respiratory water loss

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Related Commercial Resources This file is licensed to John Murray (). Publication Date: 6/1/2017 Related Commercial Resources CHAPTER 10 INDOOR ENVIRONMENTAL HEALTH BACKGROUND.. Health Sciences Relevant to Indoor Environment .... Hazard Recognition, Analysis, and Control .... AIRBORNE CONTAMINANTS.. Particles .... Gaseous Contaminants .... 10.1 10.3 10.4 10.4 10.5 10.9 NDOOR environmental health comprises those aspects of human health and disease that are determined by factors in the indoor environment. It also refers to the theory and practice of assessing and controlling factors in the indoor environment that can potentially affect health. The practice of indoor environmental health requires consideration of chemical, biological, physical and ergonomic hazards, and has the goal of increasing healthy indoor environments. Diseases are caused by genetics and exposures [biological (biotic) and/or chemical or physical (abiotic)]. Despite a huge investment in DNA research in recent decades, few diseases can be solely explained by our genes. An interaction between genes and environmental exposures is needed, and understanding indoor environmental exposures is essential in this respect. Over a 70-year lifespan in a developed region, indoor air (in homes, schools, day cares, offices, shops, etc.) constitutes around 65% of the total lifetime exposure (in mass), whereas outdoor air, air during transportation, food, and liquid makes up the rest. For more vulnerable populations, such as newborns, the elderly, and the homebound ill, indoor air in homes makes up around 80% of the exposure. It is essential for engineers and others involved in building design and operation to understand the fundamentals of indoor environmental health because the design, operation, and maintenance of buildings and their HVAC systems significantly affect the health of building occupants. In many cases, buildings and systems can be designed and operated to reduce the exposure of occupants to potential hazards. Unfortunately, neglecting to consider indoor environmental health can lead to conditions that create or worsen those hazards and increased associated exposure. This chapter provides general background information and introduces important concepts of hazard recognition, analysis, and control. It also presents information on specific hazards, and describes sources of exposure to each hazard, potential health effects, relevant exposure standards and guidelines, and methods to control exposure. This chapter also includes a brief introduction to the very broad and dynamic field of indoor environmental health. Thus, descriptions of potential hazards (and especially their controls) presented do not constitute a comprehensive, state-of-the-art review. Additional detail is available on many important topics in other ASHRAE Handbook chapters, including Licensed for single user. © 2017 ASHRAE, Inc. I • • • • • Chapter 9, Thermal Comfort, of this volume Chapter 11, Air Contaminants, of this volume Chapter 12, Odors, of this volume Chapter 16, Ventilation and Infiltration, of this volume Chapter 29, Air Cleaners for Particulate Contaminants, of the 2016 ASHRAE Handbook—HVAC Systems and Equipment The preparation of this chapter is assigned to the Environmental Health Committee. 10.1 Copyright © 2017, ASHRAE PHYSICAL AGENTS.. Thermal Environment.. Electrical Hazards .... Mechanical Energies.. Electromagnetic Radiation.. Ergonomics .... Outdoor Air Ventilation and Health .... 10.16 10.16 10.19 10.19 10.21 10.23 10.23 • Chapter 31, Ventilation of the Industrial Environment, of the 2015 ASHRAE Handbook—HVAC Applications • Chapter 46, Air Cleaners for Gaseous Contaminants, of the 2015 ASHRAE Handbook—HVAC Applications Other important sources of information from ASHRAE include the building ventilation and related requirements in Standards 62.1 and 62.2, as well as Standard 170 for health care occupancies and the Indoor Air Quality Guide (ASHRAE 2009). Additional details are available from governmental and private sources, including the U.S. Department of Health and Human Services' Centers for Disease Control and Prevention, U.S. Environmental Protection Agency, Occupational Safety and Health Administration, American Conference of Governmental Industrial Hygienists, National Institute for Occupational Safety and Health, parallel institutions in other countries, and the World Health Organization. 1. BACKGROUND Evaluation of exposure incidents and laboratory studies with humans and animals have generated reasonable consensus on safe and unsafe workplace exposures for about 1000 chemicals and particles. Consequently, many countries regulate exposures of workers to these agents. However, chemical and particle concentrations that meet occupational health criteria usually exceed levels acceptable to occupants in nonindustrial spaces such as offices, schools, and residences, where exposure times often last longer and exposures may involve mixtures of many contaminants and where those exposed comprise a less robust population (e.g., infants, the elderly, the infirm) (NAS 1981). The generally accepted broad definition of health is that in the constitution of the World Health Organization (WHO): "Health is a state of complete physical, mental, and social well-being and not merely the absence of disease or infirmity." Another definition of health, more narrowly focused on air pollution, presented by the American Thoracic Society (ATS 1999) takes into account broader, societal decision-making processes in defining what constitutes an adverse health effect of air pollution. Key points of the ATS definition of adverse effects include • Biomarkers, or biological indicators (e.g., in blood, exhaled air, sputum) of environmental effects. Because few markers have yet been sufficiently validated for use in defining thresholds, not all changes in biomarkers related to air pollution should be considered adverse effects. • Quality of life. Adverse effects of air pollution can range from watering, stinging eyes to cardiopulmonary symptoms, and even psychiatric conditions. • Physiological impact. Physical effects of pollution can be transitory or permanent, and appear alone or accompanied by other symptoms. The ATS minimum requirement for considering pollution to have an adverse effect is reversible damage accompanied This file is licensed to John Murray (). Publication Date: 6/1/2017 10.2 • • Licensed for single user. © 2017 ASHRAE, Inc. • • 2017 ASHRAE Handbook—Fundamentals (SI) by other symptoms (reversible damage alone is not sufficient). Also, effects such as developmental damage to lungs, or exacerbation of age-related decay in function, must be considered. Symptoms. Not all increased occurrences of symptoms are considered adverse effects of air pollution: only those diminishing an individual's quality of life or changing a patient's clinical status should be considered adverse. Clinical outcomes. Detectable effects of air pollution on clinical tests should be considered adverse. Mortality. Any increase in mortality should be judged adverse. Population health versus individual risk. Any increase in the risk of an exposed population should be considered adverse, even if there is no immediate, outright illness. Definitions of comfort vary. Comfort encompasses perception of the environment (e.g., hot/cold, humid/dry, noisy/quiet, bright/dark) and a value rating of affective implications (e.g., too hot, too cold). Rohles et al. (1989) noted that acceptability may represent a more useful concept of evaluating occupant response, because it allows progression toward a concrete goal. Acceptability is the foundation of a number of standards covering thermal comfort and acoustics, as well as odor comfort. Nevertheless, acceptability varies between climatic regions and cultures, and may change over time as expectations change. Concern about the health effects associated with indoor air dates back several hundred years, and has increased significantly in recent decades. During the 1970s and 1980s, this attention was mainly a result of concerns about radon and lung cancer, and about increased Table 1 Illness Physical Examination reporting by building occupants of complaints about poor health associated with exposure to indoor air or sick building syndrome (SBS). More recently, interest has largely focused on asthma, allergies, and airway infections. SBS encompasses a number of adverse health symptoms related to occupancy in a "sick" building or room, including mucosal irritation, fatigue, headache, and, occasionally, lower respiratory symptoms, and nausea. Large field studies (EPA 2012; Skov and Valbjorn 1987; Sundell et al. 1994) have shed light on the causes. Widespread occurrence of these symptoms prompted the World Health Organization to classify SBS symptoms (WHO 1983): • General symptoms, such as headache, tiredness, nausea • Mucous membrane symptoms in the nose, eyes, or throat, including coughing, sensations of dryness • Skin symptoms: redness, itching, on upper body parts Sick building syndrome is characterized by an absence of routine physical signs and clinical laboratory abnormalities with regard to sensory irritation and neurotoxic symptoms, while skin symptoms often can be objectively verified. Some investigations have sought to correlate SBS symptoms with reduced neurological and physiological performance. In controlled studies, SBS symptoms can reduce performance in susceptible individuals (Mølhave et al. 1986). Building-related illnesses (BRIs) have similar symptoms, but include physical signs and abnormalities that can be more easily clinically identified (e.g., hypersensitivity illnesses, including hypersensitivity pneumonitis, humidifier fever, asthma, and allergic rhinitis). Some illnesses associated with exposure in indoor environments are listed in Table 1. Selected Illnesses Related to Exposure in Buildings Laboratory Testing Linkage Causes/Exposures Anterior and posterior rhinomanome- Immunologic skin prick or try, acoustic rhinometry, nasal lavage, immunoassay (IgE) or in vitro testing biopsy, rhinoscopy, immunoassay (IgE) or skin prick testing Asthma Coughing, wheezing, episodic Spirometry peak expiratory flow diary, Immunology testing: skin prick methacholine challenge, exhaled NO or immunoassay (IgE); physdyspnea, wheezing on iology testing* examination, chest tightness, temporal pattern at work Organic dust toxic Cough, dyspnea, chest tight- DLCO, TLC Temporal pattern related to syndrome ness, feverishness work DLCO, FVC, TLC, CXR, lung biopsy Immunology testing: IgG antiHypersensitivity Cough, dyspnea, myalgia, body to agents present, chalpneumonitis weakness, rales, clubbing, lenge testing, physiology feverishness testing (in acute forms): spirometry, DLCO Contact dermatitis Dry skin, itching, scaling skin Scaling, rash, eczema, biopsy Patch testing; allergy testing Urticaria (hives) Multiple swollen raised itchy Inspection, biopsy Provocation testing areas of skin Eye irritation Eye itching, irritation, Tear-film break-up time, conjunctival Temporal pattern dryness staining (fluorescein) Temporal pattern Nasal irritation Stuffy, congested nose, Acoustic rhinometry, posterior and rhinitis anterior rhinomanometry, nasal lavage, nasal biopsy Central nervous Headache, fatigue, irritability, Neuropsychological testing Temporal pattern (epidemiolsystem symptoms difficulty concentrating ogy) Allergic rhinitis Stuffy/runny nose, postnasal drip, pale or erythematous mucosa Pollen and dust mites are common examples Pet dander, mold, environmental tobacco smoke, and dust mites are common examples Gram-negative bacteria or endotoxin Causative agents include thermophilic actinomycetes; molds; mixed amoebae, fungi, and bacteria; avian proteins; certain metals and chemicals Skin irritation, foods, heat/cold, direct pressure, sunlight, drugs VOCs and particulate matter are common examples VOCs and particulate matter are common examples Chemical compounds, noise, lighting, work stress, and carbon monoxide are common examples Legionnaires' dis- Pneumonia, high fever, organ Environmental surveillance (water sys- Organism isolated from patient Legionella (and other ease, Aspergillosis, dysfunction tem monitoring), Legionella pneuand source; immunology microorganism)-contaminated Pseudomonas mophila identification from patient testing aerosols from water sources infection Pontiac fever Non-pneumonic flulike Environmental surveillance (water sysRange of microorganisms, illness tem monitoring) chemicals *(1) 10% decrement in FEV1 across workday, (2) peak flow changes suggestive of work relatedness (3) methacholine reactivity resolving after six weeks away from exposure DLCO = single breath carbon monoxide diffusing capacity FVC = forced vital capacity TLC = total lung capacity CXR = chest X-ray IgE = immunoassay IgG = class G immunoglobulins FEV1 = forced expiratory volume in the first second This file is licensed to John Murray (). Publication Date: 6/1/2017 Indoor Environmental Health 1.1 HEALTH SCIENCES RELEVANT TO INDOOR ENVIRONMENT The study of health effects in indoor environments involves a number of scientific disciplines. A few are briefly described here to further the engineer's understanding of which health sciences may be applicable to a given environmental health problem. Licensed for single user. © 2017 ASHRAE, Inc. Epidemiology and Biostatistics Epidemiology studies the causes, distribution, and control of disease in a population. It represents the application of quantitative methods to evaluate health-related events and effects. Epidemiology is traditionally subdivided into observational and analytical components; the focus may be descriptive, or may attempt to identify causal relationships. Some classical criteria for determining causal relationships in epidemiology are consistency, temporality, plausibility, specificity, strength of association, and dose/response (Hill 1965). Observational epidemiology studies are generally performed with a defined group of interest because of a specific exposure or risk factor. A control group is selected on the basis of similar criteria, but without the exposure or risk factor present. A prospective study (cohort study) consists of observations of a specific group over a long time. Examples of epidemiological investigations are cross-sectional, experimental, and case-control studies. Observations conducted at one point in time are considered cross-sectional studies. In experimental studies, individuals are selectively exposed to a specific agent or condition. These studies are performed with the consent of the participants unless the condition is part of the usual working condition and it is known to be harmless. Control groups must be observed in parallel. Case-control studies are conducted by identifying individuals with the condition of interest and comparing factors of interest in individuals without that condition. Industrial, Occupational, and Environmental Medicine or Hygiene Industrial or occupational hygiene is about anticipating, recognizing, evaluating, controlling, and preventing conditions that may lead to illness or injury, or affect the well-being of workers, consumers, and/or communities. Important aspects include identifying hazardous exposures and physical stressors, determining methods for collecting and analyzing contaminant samples, evaluating measurement results, and developing suitable control measures. Occupational hygienists work closely with toxicologists for understanding chemical hazards, physicists for physical hazards (e.g., ionizing radiation), and physicians and microbiologists for biological hazards. Microbiology Buildings are more than inanimate physical entities, masses of inert material that remain relatively stable over time. The building and its occupants, contents, and surroundings constitute a dynamic tetrad in which all elements affect each other. In fact, a building is a dynamic combination of physical, chemical, and biological dimensions. Buildings can be described and understood as complex systems. Some new approaches, based on the frameworks, tools, and methods used by ecologists to understand ecosystems, can help engineers understand the processes and microbes continually occurring indoors and how they affect the building's inhabitants, durability, and function (Bassler 2009; Humphries 2012). Building scientists need to understand the complex and bidirectional relationship between the physical/chemical parameters of a building and the microbiology of that environment. Attempting to control a single parameter (e.g., temperature) to regulate the growth of a single microbe (e.g., mold), for example, does not address the complexity of the system. 10.3 Microbiologists must recognize the importance of understanding all of the environmental variables that are present in a given habitat. Simply collecting microbes from surfaces or materials in a structure is not enough to understand the organisms' behavior and relationships in the context of the building. Collecting appropriate information about the building (metadata) such as air turnover rates and material composition is essential to understanding the microbial communities that live inside it. Microbiologists must also be ever mindful of the need to distinguish between the occurrence of a microbe and the activity (metabolome) of a microbe or microbial community. Culture-independent (genetic) methods of identifying microorganisms in microbiology are rapidly changing our understanding of the occurrence and nature of microbes in indoor environments [see Microbiology of the Built Environment network (www.microbe .net)]. These methods have increased tenfold the number of known bacteria over the past decade. Efforts to better understand the relationship between the indoor environment and its microbial ecology are yielding new knowledge about the complexity of the indoor environment as an ecosystem (Corsi et al. 2012). Viruses require living cells for replication, so abiotic building components, strictly speaking, do not serve as a source of viruses. Viruses in buildings come from the building's occupants. Building components or systems can provide surfaces that facilitate transmission (e.g., doorknobs), and some viruses can become airborne. A substantial body of literature compares airborne and other routes of transmission; from a building design and operations standpoint, though, avoid generalizations because transmission routes are not the same for all viruses. Empirical data (Lowen et al. 2007) demonstrate that some airborne viruses (e.g., influenza) are inactivated more quickly at high humidity, and that low humidity rapidly reduces the size of respiratory droplets, thereby prolonging time aloft. Maintaining humidity in the range deemed comfortable to a majority of occupants reduces these effects that favor influenza transmission. Much of the literature on fungi focuses on temperature and moisture, with some emphasis on moisture and nutrient availability, although there is sufficient nutrient content on most indoor surfaces: fungi can grow even on what appears to be clean glass. There is less literature on the factors determining bacterial species and survival indoors, in spite of the growing interest in the hygiene hypothesis and humans' intimate relationships with both beneficial and harmful bacteria (e.g., probiotics). See Flanigan et al. (2001) for details. Moisture on many surfaces supports the life, reproduction, and evolution of microorganisms. The microorganisms themselves produce chemicals, some of which can alter the pH of the surface and subsequent surface chemistry. Many additional microbes arrive on human skin, which sheds on a regular basis. Skin cells and the oils and other chemicals in and on them, as well the bacteria living on them, end up on the floor, furniture, and even the walls and windows. When bacteria colonize stable surfaces, they often form complex communities. The structure and composition of these communities depend not only on the organisms present, but also on the conditions surrounding them: the moisture, chemicals, and other particles present on the surface or in the air nearby. Some of these communities may even develop into biofilms, which are very stable communities that are resistant to many antimicrobial compounds and can shelter pathogenic microbes. Bacterial communities sense the presence of other bacteria, and when they are enough of them to collectively affect their host, they all excrete chemicals that collectively affect the host and, in the case of human hosts, can make the bacteria more infectious (Bassler 2009). Toxicology Toxicology studies the influence of chemicals, particles, ultrafine particles, and bioaerosols on health. All chemical substances This file is licensed to John Murray (). Publication Date: 6/1/2017 10.4 2017 ASHRAE Handbook—Fundamentals (SI) may function as toxins, but low concentrations prevent many, but not all, of them from being harmful. Defining which component of the structure of a chemical predicts the harmful effect is of fundamental importance in toxicology. A second issue is defining the dose/response relationships of a chemical and the exposed population. Dose may refer to delivered dose (exposure presented to the target tissue) or absorbed dose (the dose actually absorbed by the body and available for metabolism). For many substances, the time of exposure may be most important: low-level exposure during a specific week during pregnancy, for instance, may be critical, whereas higher doses later may have less of an effect. Because permission to conduct exposure of human subjects in experimental conditions is difficult to obtain, most toxicological literature is based on animal studies. Isolated animal systems (e.g., homogenized rat livers, purified enzyme systems, other isolated living tissues) are used to study the effects of chemicals, but extrapolation between dose level effects from animals to humans is problematic. Licensed for single user. © 2017 ASHRAE, Inc. 1.2 HAZARD RECOGNITION, ANALYSIS, AND CONTROL Hazard recognition and analysis are conducted to determine the presence of hazardous materials or conditions as sources of potential problems. Research, inspection, and analysis determine how a particular hazard affects health. Exposure assessment, an element of hazard recognition, relies on qualitative, semiquantitative, or quantitative approaches. In many situations, air sampling can determine whether a hazardous material is present. An appropriate sampling strategy must be used to ensure validity of collected samples, determining worst-case (for compliance) or usual (average) exposures. Air sampling can be conducted to determine time-weighted average (TWA) exposures, which cover a defined period of time, or short-term exposures, which determine the magnitude of exposures to materials that are acutely hazardous. Samples may be collected for a single substance or a multicomponent mixture. Hazard analysis also characterizes the potential skin absorption or ingestion hazards of an indoor environment. Analyses of bulk material samples and surface wipe samples are also used to determine whether hazardous conditions exist. Physical agent characterization may require direct-reading sampling methods. After collection and analysis, the results must be interpreted and an appropriate control strategy developed to control, reduce, or eliminate the hazard. A main problem today is identifying which hazards, and particularly which chemical compounds, to study, although chemicals mimicking hormones (often female hormones) are increasingly of interest. Hazards are generally grouped into one of the following four classes of environmental stressors: • Chemical hazards. Routes of exposure to airborne chemicals are inhalation (aspiration), dermal (skin) contact, dermal absorption, and ingestion. The degree of risk from exposure depends on the nature and potency of the toxic effects; the endocrine effects; susceptibility of the person exposed; and timing, magnitude, and/or duration of exposure. Airborne contaminants are very important because of their ease of dispersal from sources and the risk of exposure through the lungs or skin. Airborne chemical hazards can be gaseous (vapors or gases) or particulate (e.g., dusts, fumes, mists, aerosols, fibers). Some chemicals, such as semivolatile organic compounds (SVOCs), are both gaseous and particulate. For more information, see Chapter 11. • Biological hazards. Bacteria, viruses, fungi, and other living or nonliving organisms that can cause acute and chronic illness in building occupants are classified as biological hazards in indoor environments. Routes of exposure are inhalation, dermal (skin) contact, and ingestion. The degree of risk from exposure depends on the nature and potency of the biological hazard, susceptibility of the person exposed, and magnitude and duration of exposure. • Physical hazards. These include excessive levels of ionizing and nonionizing electromagnetic radiation, noise, vibration, illumination, temperature, and force. • Ergonomic hazards. Tasks that involve repetitive motions, require excessive force, or must be carried out in awkward postures can damage muscles, nerves, and joints. Hazard Control Strategies for controlling exposures in the indoor environment include substitution (removal of the hazardous substance), isolation, disinfection, dilution ventilation, and air cleaning. Not all measures may be applicable to all types of hazards, but all hazards can be controlled by using one of them. Personal protective equipment and engineering, work practice, and administrative controls are used to apply these methods. Source removal or substitution, customarily the most effective measure, is not always feasible. Engineering controls (e.g., ventilation, air cleaning) may be effective for a range of hazards. Local exhaust ventilation is more effective for controlling point-source contaminants than is general dilution ventilation. Hazard Analysis and Control Processes. The goal of hazard analysis and control processes is to prevent harm to people from hazards associated with buildings. Quantitative hazard analysis and control processes are practical and cost-effective. Preventing disease from hazards requires facility managers and owners to answer three simple, site-specific questions: • What is the hazard? • How can it be prevented from harming people? • How can it be verified that the hazard has been prevented from harming people? Seven principles comprise effective hazard analysis and control: • Use process flow diagrams to perform systematic hazard analysis. • Identify critical control points (process steps at which the hazard can be eliminated or prevented from harming people). • Establish hazard control critical limits at each critical control point. • Establish a hazard control monitoring plan for critical limits at critical control points. • Establish hazard control corrective actions for each critical limit. • Establish procedures to document all activities and results. • Establish procedures to confirm that the plan (1) actually works under operating conditions (validation), (2) is implemented properly (verification), and (3) is periodically reassessed. 2. AIRBORNE CONTAMINANTS Many airborne contaminants cause problems in both industrial and nonindustrial indoor environments. These include biological and nonbiological particles [e.g., synthetic vitreous fibers, asbestos, environmental tobacco smoke (ETS), combustion nuclei, dust (including human skin scales)], bioaerosols, and chemical gases and vapors. Airborne contaminants may enter the building from the outdoors or be released indoors by processes, building materials, furnishings, equipment, or occupant activities. In industrial environments, airborne contaminants are usually associated with the type of process that occurs in a specific setting, and exposures may be determined relatively easily by air sampling. Airborne contaminants in nonindustrial environments may result from emissions and/or shedding of building materials and systems; originate in outdoor air; or result from building operating and maintenance programs, procedures, or conditions. In general, compared to industrial settings, nonindustrial environments include many more contaminants with the potential to contribute to health-related problems. These contaminants are usually present in lower concentrations and often are more difficult to This file is licensed to John Murray (). Publication Date: 6/1/2017 Indoor Environmental Health identify. More information on contaminant types, characteristics, typical levels, and measurement methods is presented in Chapter 11. Licensed for single user. © 2017 ASHRAE, Inc. 2.1 PARTICLES Particulate matter can be solid or liquid; typical examples include dust, smoke, fumes, and mists. Dusts are particles of a size and density that will settle; their behavior is affected mostly by gravity. Smoke, fumes, and mists contain mixtures of overlapping particle sizes, many in the smaller ranges (~0.1 μm) where gravity is less important than temperature, particle charge, and other factors in determining how long particles remain aloft. Fibers are solid particles with length several times greater than their diameter, such as asbestos, manufactured mineral fibers, synthetic vitreous fibers, and refractory ceramic fibers. Bioaerosols of concern to human health range from 0.5 to 30 m in diameter, but generally bacterial and fungal aerosols range from 2 to 8 m in diameter because of agglomeration or rafting of cells or spores (Lighthart 1994). Units of Measurement. The quantity of particles in the air is frequently reported as the mass concentration or count concentration in a given volume of air. Mass concentration units are milligrams per cubic metre of air sampled (mg/m3) or micrograms per cubic metre of air sampled (g/m3). For conversion, 1 mg/m3 = 1000 g/m3. Mass units are widely used in industrial environments because these units are used to express occupational exposure limits. Count concentration units are usually expressed as counts per cubic foot, cubic centimetre (cc), or cubic metre, and are specified for a given range of particle diameter. Count concentration measurements are generally used in environments such as office buildings and industrial cleanrooms. General Health Effects of Exposure. Health effects of airborne particulate matter depend on several factors, including particle dimension, durability, dose, and toxicity of materials in the particle. Respirable particles vary in size from Aspergillus Tritirachium > Aspergillus > Cladosporium Tritirachium > Aspergillus > Cladosporium Notes: CFU/m3 = colony-forming units per cubic metre of air. Culture media, for this example, was malt extract agar (ACGIH 1989). U.S. Environmental Protection Agency (EPA) and Department of Housing and Urban Development (HUD) researchers have developed a metric called the environmental relative moldiness index (ERMI) to objectively describe the home mold burden (Vesper et al. 2007). A DNA-based analysis called mold-specific quantitative PCR (MSQPCR) of 36 molds, including 26 species associated with homes with water damage and 10 found in homes independent of water damage, forms the basis of the ERMI. Controlling Exposures to Particulate Matter Control of airborne particulate levels may be achieved by one of four methods: • • • • Reduction of source emissions Capture of emissions at the source using local exhaust Dilution using mechanical ventilation Removal from ventilation air by filtration Of these, filtration is of particular importance to HVAC&R. This topic is covered in detail in Chapter 29 in the 2016 ASHRAE Handbook—HVAC Systems and Equipment. Control of bioaerosols is a complex issue because of their capacity for growth and dispersion. However, in general, particulate removal devices and controls are effective in collecting and removing bioaerosols, including allergens (Foarde et al. 1994). Control may also be achieved by ultraviolet irradiation, as described in Chapter 60 of the 2015 ASHRAE Handbook—HVAC Applications. 3. GASEOUS CONTAMINANTS The terms gas and vapor are both used to describe the gaseous state of a substance. Gas is the correct term for describing any pure substance or mixture that naturally exists in the gaseous state at normal atmospheric conditions. That is, its vapor pressure is greater than ambient pressure at ambient temperature. Examples are oxygen, helium, ammonia, and nitrogen. Vapor is used to describe a substance in the gaseous state whose natural state is a liquid or solid at normal atmospheric conditions. The vapor pressure is below ambient pressure at ambient temperature. Examples include benzene, carbon tetrachloride, and water. Differences between the two classes reflect their preferred states: • For a strong source, the concentration of a gas in air in a confined space can rise above one atmosphere. Thus, even nontoxic gases can be lethal if they completely fill a space, displacing the oxygen necessary for survival. • Vapors can never exceed their saturated vapor pressure in air. The most familiar example of a vapor is water, with relative humidity expressing the air concentration as a percentage of the saturated vapor pressure. • Vapors, because their natural state is liquid or solid (low vapor pressure), tend to condense on surfaces and be adsorbed. Gaseous contaminants can also usefully be divided into organic and inorganic types. Organic compounds include all chemicals based on a skeleton of carbon atoms. Because carbon atoms easily combine to form chain, branched, and ring structures, there is a wide variety of organic compounds. Despite the variety, they have similarities that can be used in sampling, analysis, and removal. Chemists subclassify organic compounds based on families having similar structure and predictable properties. Organic gaseous contaminants include gases such as methane, but the majority are vapors. All other gaseous contaminants are classified as inorganic. Most inorganic air contaminants of interest to ventilation engineers are gases (mercury is an important exception). Major chemical families of inorganic and organic gaseous contaminants, with examples of specific compounds, are shown in Table 5, along with information This file is licensed to John Murray (). Publication Date: 6/1/2017 Air Contaminants 11.9 Table 5 Major Chemical Families of Gaseous Air Contaminants No. Family Examples Inorganic Contaminants 1. Single-element atoms Chlorine, radon, mercury and molecules 2. Oxidants Chlorine is a strong respiratory irritant used as a disinfectant; outdoor sources include seawater, chlorinated pools, and road salt. Radon is an important soil gas. Mercury is the vapor in fluorescent light bulbs and tubes. Corrosive; respiratory irritants. 6. Miscellaneous Toxic; fuel combustion product. Carbon dioxide and hydrogen sulfide are only weakly acidic. Hydrogen sulfide is the main agent in sewer gas. Other members are corrosive and respiratory irritants. Some are important outdoor contaminants. Ammonia is used in cleaning products; it is a strong irritant. Hydrazine is used as an anticorrosion agent. Nitrous oxide (laughing gas) is used as an anesthetic. Used in the semiconductor industry. Ozone, nitrogen dioxide, hydrogen peroxide 3. Reducing agents Carbon monoxide 4. Acid gases Carbon dioxide, hydrogen chloride, hydrogen fluoride, hydrogen sulfide, nitric acid, sulfur dioxide, sulfuric acid 5. Nitrogen compounds Ammonia, hydrazine, nitrous oxide Organic Contaminants 7. n-Alkanes Licensed for single user. © 2017 ASHRAE, Inc. Other Information Arsine, phosphine 8. Branched alkanes Methane, propane, n-butane, n-hexane, n-heptane, n-octane, n-nonane, n-decane, n-undecane, n-dodecane 2-methyl pentane, 2-methyl hexane 9. Alkenes and cyclic hydrocarbons Ethylene, butadiene, 1-octene, cyclohexane, 4-phenyl cyclohexene (4-PC) 10. Chlorofluorocarbons R-11 (trichlorofluoromethane), R-12 (dichlorodifluoromethane), R-114 (dichlorotetrafluoroethane) 11. Chlorinated Carbon tetrachloride, chloroform, hydrocarbons dichloromethane, 1,1,1-trichloroethane, trichloroethylene, tetrachloroethylene, p-dichlorobenzene 12. Halide compounds Methyl bromide, methyl iodide 13. Alcohols Methanol, ethanol, 2-propanol (isopropanol), 3-methyl 1-butanol, ethylene glycol, 2-butoxyethanol, phenol, texanol 14. Ethers Ethyl ether, methyl tertiary butyl ether (MTBE), 2-butoxyethanol Formaldehyde, acetaldehyde, acrolein, 15. Aldehydes benzaldehyde 16. Ketones 2-propanone (acetone), 2-butanone (MEK), methyl isobutyl ketone (MIBK), 2-hexanone 17. Esters Ethyl acetate, vinyl acetate, butyl acetate, texanol 18. Nitrogen compounds Nitromethane, acetonitrile, acrylonitrile, other than amines urea, hydrogen cyanide, peroxyacetal nitrite (PAN) 19. Aromatic Benzene, toluene, p-xylene, styrene, hydrocarbons 1,2,4 trimethyl benzene, naphthalene, benz--pyrene 20. Terpenes -pinene, limonene n-Alkanes are linear molecules and relatively easily identified analytically. Along with the far more numerous branched alkanes, they are components of solvents such as mineral spirits. Numerous; members are difficult to separate and identify. Many occur as components of products such as gasoline, kerosene, mineral spirits, etc. Ethylene gas is produced by ripening fruit (and used in the fruit industry). Some liquid members are components of gasoline, etc. 4-PC is responsible for "new carpet" odor. Widely used as refrigerants; are being phased out because of their ozone-depleting potential. Dichlorobenzene, an aromatic chemical, is a solid used as an air freshener. Others shown here are liquids and effective nonpolar solvents. Some are used as degreasers or in the dry-cleaning industry. Low combustibility; some are used as flame retardants. Strongly polar. Some (including 2-butoxyethanol and texanol) are used as solvents in water-based products. Phenol is used as a disinfectant. 3-methyl 1-butanol is emitted by some molds. Ethyl ether and 2-butoxyethanol are used as solvents. MTBE is added to gasoline to improve combustion in vehicle motors. Formaldehyde, acetaldehyde, and acrolein have unpleasant odors and are strong irritants formed during combustion of fuels and tobacco. Medium-polarity chemicals; some are useful solvents. Acetone and 2-hexanone are emitted by some molds. Medium-polarity chemicals; some have pleasant odors and are added as fragrances to consumer products. Includes several different types of chemicals with few common properties. Acetonitrile is used as a solvent; urea is a

metabolic product; PAN is found in vehicle exhaust. Benzene, toluene, and xylene are widely used as solvents and in manufacturing, and are ubiquitous in indoor air. Naphthalene is used as moth repellent. A variety of terpenes are emitted by wood. The two listed here have pleasant odors and are used as fragrances in cleaners, perfumes, etc. 21. Heterocylics Ethylene oxide, tetrahydrofuran, 3-methyl Most are of medium polarity. Ethylene oxide is used as a disinfectant. furan, 1,4-dioxane, pyridine, nicotine Tetrahydrofuran and pyridine are used as solvents. Nicotine is a component of tobacco smoke. 22. Organophosphates Malathion, tabun, sarin, soman Listed are components of agricultural pesticides and occur as outdoor air contaminants. 23. Amines Trimethylamine, ethanolamine, Typically have unpleasant odors detectable at very low concentrations. Some cyclohexylamine, morpholine (cyclohexylamine and morpholine) are used as antioxidants in boilers. 24. Monomers Vinyl chloride, ethylene, methyl Potential to be released from their respective polymers (PVC, polythene, perspex, methacrylate, styrene polystyrene) if materials are heated. 25. Mercaptans and other Bis-2-chloroethyl sulfide (mustard gas), Sulfur-containing chemicals typically have unpleasant odors detectable at very low sulfur compounds ethyl mercaptan, dimethyl disulfide concentrations. Ethyl mercaptan is added to natural gas so that gas leaks can be detected by odor. Mustard gas has been used in chemical warfare. 26. Organic acids Formic acid, acetic acid, butyric acid Formic and acetic acids (vinegar) are emitted by some types of wood. Butyric acid is a component of "new car" odor. 27. Miscellaneous Phosgene, siloxanes Phosgene is a toxic gas released during combustion of some chlorinated organic chemicals. Siloxanes occur widely in consumer products, including adhesives, sealants, cleaners, and hair and skin care products. This file is licensed to John Murray (). Publication Date: 6/1/2017 Licensed for single user. © 2017 ASHRAE, Inc. 11.10 about occurrence and use. Some organics belong to more than one class and carry the attributes of both. Another useful gaseous contaminant classification is polar versus nonpolar. There is a continuous distribution between these extremes. For polar compounds, charge separation occurs between atoms, which affects physical characteristics as well as chemical reactivity. Water is one of the best examples of a polar compound, and consequently polar gaseous contaminants tend to be soluble in water. Nonpolar compounds are much less soluble in water, but dissolve in nonpolar liquids. This classification provides the basis for dividing consumer products that contain organic compounds into water-based and solvent-based. Contaminant classes in Table 5 that are strongly polar include acid gases, chemicals containing oxygen (e.g., alcohols, aldehydes, ketones, esters, organic acids), and some nitrogen-containing chemicals. Nonpolar classes include all hydrocarbons (alkyl, alkene, cyclic, aromatic), chlorinated hydrocarbons, terpenes, and some sulfur-containing chemicals. Because no single sampling and analysis method applies to every (or even most) potential contaminant, having some idea what the contaminants and their properties might be is very helpful. Contaminants have sources, and consideration of the locale, industries, raw materials, cleaners, and consumer products usually provides some guidance regarding probable contaminants. Material safety data sheets (MSDS) provide information on potentially harmful chemicals that a product contains, but the information is often incomplete. Once a potential contaminant has been identified, the Merck Index (Budavi 1996), the Toxic Substances Control Act Chemical Substance Inventory (EPA 1979), Dangerous Properties of Industrial Materials (Sax and Lewis 1988), and Handbook of Environmental Data on Organic Chemicals (Verschueren 1996) are all useful in identifying and gathering information on contaminant properties, including some known by trade names only. Chemical and physical properties can be found in reference books such as the Handbook of Chemistry and Physics (Lide 1996). Note that a single chemical compound, especially an organic one, may have several scientific names. To reduce confusion, the Chemical Abstracts Service (CAS) assigns each chemical a unique five- to nine-digit identifier number. Table 6 shows CAS numbers and some physical properties for selected gaseous contaminants. The volatility designation for organic chemicals (VVOC, VOC, SVOC) is explained in the section on Volatile Organic Compounds. Volatilities, expressed more exactly in boiling point and saturated vapor pressure data, are important in predicting airborne concentrations of gaseous contaminants in cases of spillage or leakage of liquids. For example, because of its much higher volatility, ammonia requires more rigorous safety precautions than ethylene glycol when used as a heat exchange fluid. In laboratories where several acids are stored, hydrochloric acid (hydrogen chloride) usually causes more corrosion than sulfuric or nitric acids because its greater gaseous concentration results in escape of more chemical. Additional chemical and physical properties for some of the chemicals in Tables 5 and 6 can be found in Chapter 33. Harmful Effects of Gaseous Contaminants Harmful effects may be divided into four categories: toxicity, irritation, odor, and material damage. Toxicity. The harmful effects of gaseous pollutants on a person depend on both short-term peak concentrations and the timeintegrated exposure received by the person. Toxic effects are generally considered to be proportional to the exposure dose, although individual response variation can obscure the relationship. The allowable concentration for short exposures is higher than that for long exposures. Safe exposure limits have been set for a number of common gaseous contaminants in industrial settings. This topic where 2017 ASHRAE Handbook—Fundamentals (SI) is covered in more detail in the section on Industrial Air Contaminants and in Chapter 10. A few gaseous contaminants are also capable of causing cancer. Formaldehyde has recently been declared a known human carcinogen by the U.S. National Toxicology Program (NTP 2011), based on an earlier report issued by the International Agency for Research in Cancer (IARC 2004). The NTP also stated that styrene is "reasonably anticipated to be a human carcinogen" (NTP 2011). Gaseous contaminants can also be responsible for chronic health effects when exposure to low levels occurs over a long period of time. Acetaldehyde, acrolein, benzene, 1,3-butadiene, 1,4-dichlorobenzene, formaldehyde, naphthalene, and nitrogen dioxide have recently been identified as priority chronic hazards in U.S. homes (Logue et al. 2011). More information on health effects of gaseous contaminants can be found in Chapter 10. Irritation. Although gaseous pollutants may have no discernible continuing health effects, exposure may cause physical irritation to building occupants. This phenomenon has been studied principally in laboratories and nonindustrial work environments, and is discussed in more detail in the section on Nonindustrial Indoor Air Contaminants and in Chapter 10. Odors. Gaseous contaminant problems often appear as complaints about odors, and these usually are the result of concentrations considerably below industrial exposure limits. Odors are discussed in more detail in Chapter 12. Note that controlling gaseous contaminants because they constitute a nuisance odor is fundamentally different from controlling a contaminant because it has a demonstrated health effect. Odor control frequently can use limited-capacity "peak-shaving" technology to drop peaks of odorous compounds below the odor threshold. Later reemission at a low rate is neither harmful nor noticed. Such an approach may not be acceptable for control of toxic materials. Damage to Materials. Material damage from gaseous pollutants includes corrosion, embrittlement, and discoloration. Because these effects usually involve chemical reactions that need water, material damage from air pollutants is less severe in the relatively dry indoor environment than outdoors, even at similar gaseous contaminant concentrations. Contaminants that can corrode HVAC systems include seawater, acid gases (chlorine, hydrogen fluoride, hydrogen sulfide, nitrogen oxides and sulfur oxides), ammonia, and ozone. Corrosion from these gases can also cause electrical, electronic, and telephone switching systems to malfunction (ISA 1985). Some dry materials can be significantly damaged. These effects are most serious in museums, because any loss of color or texture changes the essence of the object. Libraries and archives are also vulnerable, as are pipe organs and textiles. Consult Chapter 23 in the 2015 ASHRAE Handbook—HVAC Applications for additional information and an exhaustive reference list. Units of Measurement Concentrations of gaseous contaminants are usually expressed in the following units: ppm = parts of contaminant by volume per million parts of air by volume ppb = parts of contaminant by volume per billion parts of air by volume 1000 ppb = 1 ppm mg/m3 = milligrams of contaminant per cubic metre of air µg/m3 = micrograms of contaminant per cubic metre of air Conversions between ppm and mg/m3 are ppm = [8.314(273.15 + t)/Mp] (mg/m3) (1) mg/m3 = [0.1203(Mp)/(273.15 + t)] (ppm) (2) M = relative molar mass of contaminant This file is licensed to John Murray (). Publication Date: 6/1/2017 Air Contaminants 11.11 Table 6 Characteristics of Selected Gaseous Air Contaminants Chemical and Physical Properties Licensed for single user. © 2017 ASHRAE, Inc. Contaminant Table 5 CASa Family number Volatilityb Inorganic Contaminants Ammonia 5 Arsine 6 Carbon dioxide 4 Carbon monoxide 3 Chlorine 1 Hydrogen chloride 4 Hydrogen fluoride 4 Hydrogen sulfide 4 Mercury 1 Nitric acid 4 Nitric oxide 5 Nitrogen dioxide 2 Ozone 2 Sulfur dioxide 4 Organic Contaminants 1,1,1-trichloroethane 11 1,2,4-trimethylben19 zene 2-butanone (MEK) 16 Mc Chemical and Physical Properties Usage 7664-41-7 7784-42-1 124-38-9 630-08-0 7782-50-5 7647-01-0 7664-39-3 7783-06-4 7439-97-6 7697-37-2 10102-43-9 10102-44-0 10028-15-6 7446-09-5 Gas Gas Gas Gas Gas Gas Gas Vapor Gas Vapor Vapor Gas Vapor Gas Gas 17 145.1,d 145.2e 78 44 145.2 28 145.2 71 145.1, 145.2 37 145.2 20 34 145.1, 145.2 201 63 30 145.1 46 145.1, 145.2 48 145.1, 145.2 64 145.1, 145.2 71-55-6 95-63-6 VOC VOC 133 120 IAQf IAQ 78-93-3 VVOC 72 2-butoxyethanol 13, 14 111-76-2 VOC 4-phenyl 9, 19 4994-16-5 SVOC cyclohexene -pinene 20 127-91-3 VOC Acetaldehyde 15 75-07-0 VVOC Acetic acid 26 64-19-7 VOC Acetone 16 67-64-1 VVOC Acrolein 15 107-02-8 VOC Benzene 19 71-43-2 VOC Butyl acetate 17 123-86-4 VOC Carbon disulfide 25 75-15-0 VVOC Carbon tetrachloride 11 56-23-5 VOC Chloroform 11 67-66-3 VVOC Cyclohexane 9 110-82-7 VVOC Cyclohexylamine 9, 23 108-91-8 VOC Cyclopentane 9 287-92-3 VVOC Dichlorodifluoro10 75-71-8 VVOC methane Dichloromethane 12 75-09-2 VVOC 118 158 145.1, 145.2, IAQ IAQ IAQ Dimethyl disulfide Dimethylmethylphosphate Ethanol Ethyl acetate 136 44 60 58 56 78 116 76 154 119 84 99 70 121 IAQ 145.1, 145.2 85 145.2, IAQ 145.2, IAQ IAQ IAQ IAQ 145.2 25 22 624-92-0 756-79-6 VVOC VOC 94 124 13 17 64-17-5 141-78-6 VVOC VVOC 46 88 145.2, IAQ IAQ = Chemical Abstracts Services. of organic chemicals complies with Table 9. VVOC adopted from the list produced by Salthammer (2016). Volatility of inorganic chemicals is gas if boiling point is less than 20°C, and vapor if boiling point is greater than 20°C. cM = molar mass. bVolatility Mc 13 21 15 107-21-1 75-21-8 50-00-0 VOC VVOC VVOC 62 44 30 15 18 8 13 13 20 17, 22 7 13 16 66-25-1 74-90-8 75-28-5 78-83-1 67-63-0 5989-27-5 121-75-5 74-82-8 67-56-1 108-10-1 VOC VVOC VVOC VOC VVOC VOC VOC Gas VVOC VOC 100 27 58 74 60 136 330 16 32 100 14 1634-04-4 VVOC 88 21, 23 19 7 7 7 7 21 16, 21 110-91-8 91-20-3 124-18-5 112-40-3 110-54-3 142-82-5 54-11-5 872-50-4 VOC VOC VOC VOC VVOC VOC SVOC VOC 87 128 142 170 86 100 162 99 7 7 7 11, 19 13 27 7 27 9, 19 11 111-84-2 111-65-9 1120-21-4 106-46-7 108-95-2 75-44-5 74-98-6 Various 100-42-5 127-18-4 VOC VOC VOC VOC VOC VVOC VVOC VOC VOC VOC Toluene 19 108-88-3 VOC Toluene diisocyanate Trichloroethylene Trichlorofluoromethane Vinyl chloride monomer 18 584-84-9 SVOC 11 10 79-01-6 75-69-4 VOC VVOC 131 137 24 75-01-4 VVOC 63 Hexanal Hydrogen cyanide Isobutane Isobutanol Isopropanol Limonene Malathion Methane Methanol Methyl isobutyl ketone Methyl tertiary butyl ether Morpholine Naphthalene n-decane n-dodecane n-hexane n-heptane Nicotine N-methylpyrrholidone n-nonane n-octane n-undecane p-dichlorobenzene Phenol Phosgene Propane Siloxanes Styrene Tetrachloroethylene Usage 145.1, 145.2, IAQ 145.2 IAQ 145.1, 145.2 145.2, IAQ IAQ IAQ IAQ IAQ IAQ 145.1, 145.2 145.2 128 IAQ 114 IAQ 156 IAQ 147 IAQ 94 IAQ 90 44 IAQ various IAQ 104 IAQ 166 145.1, 145.2, IAQ 92 145.1, 145.2, IAQ 174 IAQ IAQ dListed as a challenge gas for laboratory testing of gas-phase filter granular media using ASHRAE Standard 145.1. eListed as a challenge gas for testing full-size gas-phase filters using ASHRAE Standard 145.2. fCommonly found in buildings and may impact indoor air quality (IAQ) (taken from list in Table 10). p = mixture pressure, kPa t = mixture temperature, °C Concentration data are often reduced to standard temperature and pressure (i.e., 25°C and 101.325 kPa), in which case, ppm = (24.46/M) (mg/m3) Ethylene glycol Ethylene oxide Formaldehyde 145.2 145.1, 145.2, IAQ IAQ 145.2 aCAS Contaminant Table 5 CASa Family number Volatilityb (3) Using the 21°C standard temperature more familiar to engineers results in a conversion factor between ppm and mg/m3 of 24.14 in Equation (3). A temperature of 0°C gives a corresponding conversion factor of 22.41. These calculations show that variations in indoor temperature are likely to impact conversion factors by 1% or less, and can probably be ignored. However, outdoor temperatures may result in conversion factors in Equations (1) and (2) that differ by 10% from indoor ones, so that indoor and outdoor data may need to be converted separately using the appropriate factors. The differences in the conversion factors are caused by the fact that gases contract and become denser as temperatures decrease. Concentrations expressed in ppm are temperature independent, because This file is licensed to John Murray (). Publication Date: 6/1/2017 11.12 2017 ASHRAE Handbook—Fundamentals (SI) Table 7 Gaseous Contaminant Sample Collection Techniques Technique* Advantages Disadvantages Real-time readout, continuous monitoring possible Several pollutants possible with one sample (when coupled with chromatograph, spectroscope, or multiple detectors) Average concentration must be determined by integration No preconcentration possible before detector; sensitivity may be inadequate On-site equipment often complicated, expensive, intrusive, and requires skilled operator 2. Capture by pumped flow through colorimetric detector tubes, papers, or tapes Very simple, relatively inexpensive equipment and materials Immediate readout Integration over time One pollutant per sample Relatively high detection limit Poor precision Requires multiple tubes, papers, or tapes for high concentrations or long-term measurements 3. Capture by pumped flow through solid adsorbent; subsequent desorption for concentration measurement On-site sampling equipment relatively simple and inexpensive Preconcentration and integration over time inherent in method Several pollutants possible with one sample Sampling media and desorption techniques are compound-specific Interaction between captured compounds and between compounds and sampling media; bias may result Gives only average over sampling period, no peaks Subsequent concentration measurement required 4. Collection in evacuated containers Very simple on-site equipment No pump (silent) Several pollutants possible with one sample Subsequent concentration measurement required Gives average over sampling period; no peaks Finite volume requires multiple containers for long-term or continuous measurement 5. Collection in nonrigid containers (specialized, commercially available sampling bags) Simple, inexpensive on-site equipment (pumps required) Several pollutants possible with one sample Cannot hold some pollutants Subsequent concentration measurement required Gives average over sampling period; no peaks Finite volume requires multiple containers for long-term or continuous measurement 6. Cryogenic condensation Wide variety of organic pollutants can be Water vapor interference captured Subsequent concentration measurement required Minimal problems with interferences and media Gives average over sampling period; no peaks interaction Several pollutants possible with one sample 7. Liquid impingers (bubblers) Integration over time Several pollutants possible with one sample if appropriate liquid chosen May be noisy Subsequent concentration measurement required Gives average over sampling period; no peaks Immediate readout possible Simple, unobtrusive, inexpensive No pumps, mobile; may be worn by occupants to determine average exposure One pollutant per sample Relatively high detection limit Poor precision May require multiple badges for higher concentrations or long-term measurement Simple, unobtrusive, inexpensive No pumps, mobile; may be worn by occupants to determine average exposure Subsequent concentration measurement required Gives average over sampling period; no peaks Poor precision Licensed for single user. © 2017 ASHRAE, Inc. Active Methods 1. Direct flow to detectors Passive Methods 8. Passive colorimetric badges 9. Passive diffusional samplers Sources: ATC (1990), Lodge (1988), NIOSH (1977, 1994), and Taylor et al. (1977). *All techniques except 1, 2, and 8 require laboratory work after completion of field sampling. Only first technique is adaptable to continuous monitoring and able to detect shortterm excursions. both the contaminant gas and the diluting air contract. However, concentrations expressed in mg/m3 increase as temperature decreases, leading to lower ratios of ppm to mg/m3. Equations (1) to (3) are strictly true only for ideal gases, but generally are acceptable for dilute vaporous contaminants dispersed in ambient air. Measurement of Gaseous Contaminants The concentration of contaminants in air must be measured to determine whether indoor air quality conforms to occupational health standards (in industrial environments) and is acceptable (in nonindustrial environments). Measurement methods for airborne chemicals that are important industrially have been published by several organizations, including NIOSH (1994) and OSHA (1995). Methods typically involve sampling air with pumps for several hours to capture contaminants on a filter or in an adsorbent tube, followed by laboratory analysis for detection and determination of contaminant concentration. Concentrations measured in this way can usefully be compared to 8 h industrial exposure limits. Measurement of gaseous contaminants at the lower levels acceptable for indoor air is not always as straightforward. Relatively costly analytical equipment may be needed, and it must be calibrated and operated by experienced personnel. Currently available sample collection techniques are listed in Table 7, with information about their advantages and disadvantages. Analytical measurement techniques are shown in Table 8, with information on the types of contaminants to which they apply. Tables 7 and 8 provide an overview of gaseous contaminant sampling and analysis, with the intent of allowing informed interaction with specialists. Techniques 1, 2, and 8 in Table 7 combine sampling and analysis in one piece of equipment and give immediate, on-site results. The other sampling methods require laboratory analysis after the field work. Equipment using the first technique can be coupled with a This file is licensed to John Murray (). Publication Date: 6/1/2017 Air Contaminants 11.13 Table 8 Method Analytical Methods to Measure Gaseous Contaminant Concentration Description Typical Application (Family) Gas chromatography Separation of gas mixtures by time of passage down absorption column (using the following detectors) Flame ionization Change in flame electrical resistance caused by ions of pollutant Volatile, nonpolar organics (7-27) Flame photometry Measures light produced when pollutant is ionized by a flame Sulfur (25), phosphorous (22) compounds Most organics (7-27), except methane Halogenated organics (11, 12) Nitrogenated organics (18, 23) Photoionization Measures ion current for ions created by ultraviolet light Electron capture Radioactively generated electrons attach to pollutant atoms; current measured Mass spectroscopy Pollutant molecules are charged, passed through electrostatic magnetic fields in Volatile organics (7-27 with boiling point vacuum; path curvature depends on mass of molecule, allowing separation

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