UNIVERSITI TUN HUSSEIN ONN MALAYSIA

FINAL EXAMINATION
SEMESTER II
SESSION 2021/2022

COURSE NAME	:	AIR CONDITIONING SYSTEM DESIGN
COURSE CODE	:	BDE40103
PROGRAMME CODE	:	BDD
EXAMINATION DATE	:	JULY 2022
DURATION	:	3 HOURS
INSTRUCTION	:	1. PART A: ANSWER **FOUR (4)** QUESTIONS ONLY FROM **FIVE (5)** QUESTIONS. PART B: ANSWER **ALL** QUESTIONS. 2. THIS FINAL EXAMINATION IS CONDUCTED VIA **CLOSED BOOK.** 3. STUDENTS ARE **PROHIBITED** TO CONSULT THEIR OWN MATERIAL OR AN EXTERNAL RESOURCES DURING THE EXAMINATION CONDUCTED VIA CLOSED BOOK.

THIS QUESTION PAPER CONSISTS OF **FOURTEEN (14)** PAGES

PART A

Q1 (a) State the purpose of the chilled water system.

(5 marks)

(b) **Figure Q1(b)** shows the difference between individual and centralized air conditioning systems. Individual air conditioners (Traditional air conditioning) are used in Malaysia, primarily for residents. In addition, a centralized air conditioning system that may be used both for cooling and heating has been used outside of Malaysia.

Based on these statements, what solution would you suggest for a large building in Malaysia and why?

(15 marks)

Q2 (a) Thermal comfort is "the condition of mind that expresses satisfaction with the thermal environment and is assessed by subjective evaluation" (ASHRAE Standard 55). Based on these statements, explain about airflow requirement in the building.

(8 marks)

(b) Determine the predicted mean vote (PMV) and predicted per cent dissatisfied (PPD), based on the information given below:

(i) Metabolic level, M= 80 W/m^2;
(ii) Clothing insulation, I_{cl}, = 0.1 m^2 kW;
(iii) Air temperature, t_a, = 25 °C;
(iv) Mean radiant temperature, t_r, = 20 °C;
(v) Relative air velocity, v_{ar}, = 0.5 m/s;
(vi) Water vapour partial pressure, pa, = 1500 Pa;
(vii) The effective mechanical power, W = 0 W/m^2;
(viii) The clothing surface area factor, f_{cl} = 1.1145;
(ix) Convection heat transfer coefficient, hc = 5.48 W(m^2.K);
(x) The clothing surface temperature, tcl = 34.7 (°C); and
(xi) Plot PMV and PPD in **Figure Q2 (b)** and justify the thermal comfort?

(12 marks)

Q5 (a) The total pressure (H_t) of a flowing define as,

$$H_t = H_s + H_v$$

where, H_t = total pressure,

H_s = Static pressure

H_v = Velocity pressure,

Sketch and explain the manometer arrangement to read static, total, and velocity pressure

(6 Marks)

(b) The Cast iron piping systems are shown in **Figure Q5(b)(i),** installed in a chilled water system with 300 GPM of water flowing. Find the pressure drop through a 3 ½ inch with 90° standard elbow in which water flows from point 1 if the velocity at point 2 is 500 ft. /min. Use **Figure Q5(b)(ii)** to solve this problem.

Hint: 1 ft^3 = 7.48 gallon

1ft^3 = 1728 in^3

(7 marks)

(c) Pressure loss in duct fittings when the air flows through duct fittings such as elbows, tees, etc., is called loss coefficient. These pressure losses, called dynamic losses, are due to turbulence and direction changes. This method is expressed as follows:

$$H_f = C \times H_v = C \times \left(\frac{V}{4000}\right)^2$$

Where H_f = total pressure loss through the fitting, inch w.

C = a loss coefficient

H_v = velocity pressure at the fitting, in w.

V = velocity

Figure Q5(c)(i) shows that the ducting system was installed with the inlet connection with the fan. The fan inlet velocity is 3000 ft. /min. What is the pressure loss to the fan for case A and case B? Which case is better? Defend your answer by giving three (3) reasons.

(7 marks)

PART B

Q6 (a) Explain in your own words what is meant by sound, noise, frequency, decibel, and wavelength?

(5 marks)

(b) Duct-borne noise is noise traveling through the ductwork and out of the diffusers, and return grills are a problem that should be addressed at the design stage. **Figure 6(b)** shows the Typical Paths of Noise and Vibration Propagation in HVAC Systems (ASHRAE, 1995). Describe five (5) types of duct bone noise based on **Figure 6(b).**

(15 marks)

- END OF QUESTION -

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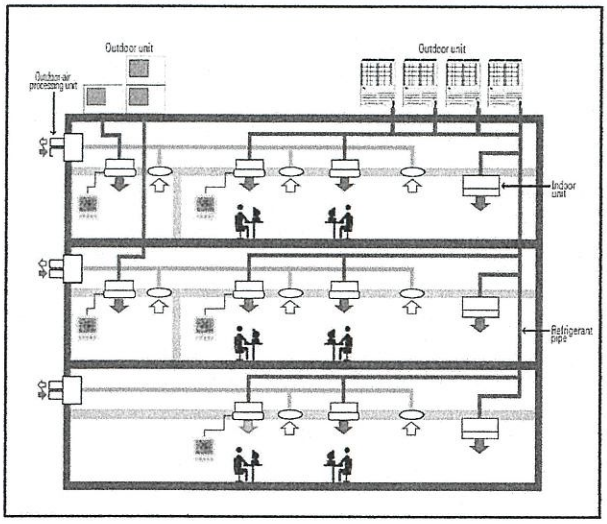
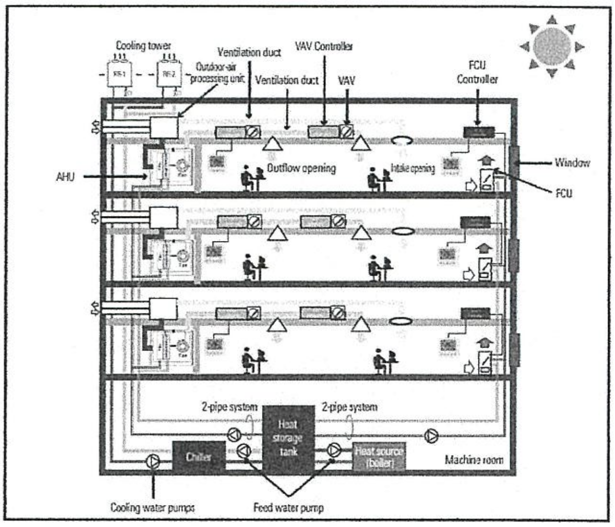


(i)



(ii)

Figure Q1(b): (i) Individual Air Conditioning System, and (ii) Central Air Conditioning System (Sources: www.renesas.com)

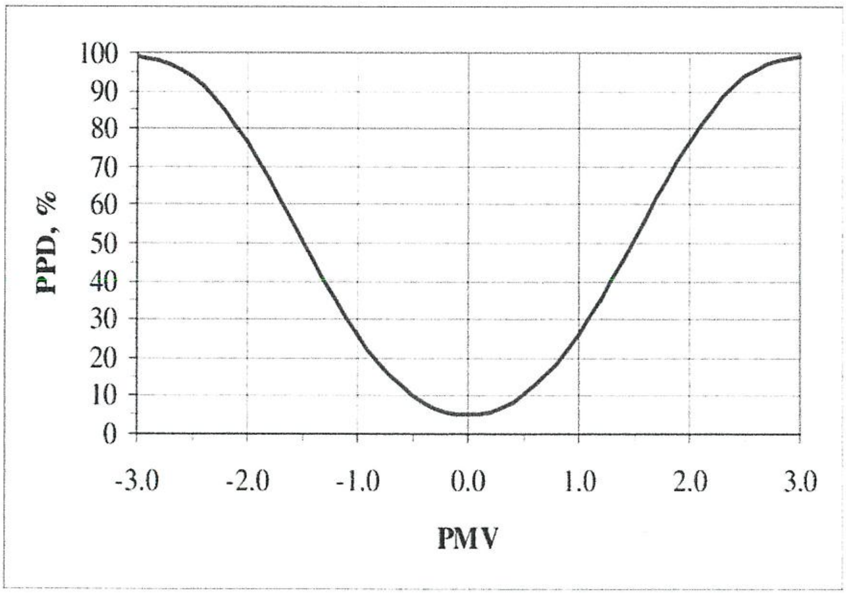


Figure Q2(b): Index PPD and PMV variation Thermal

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Table Q3(b)(ii): Thermodynamic Properties of Water

(Table 3, Chapter 1, 2017 ASHRAE Handbook—Fundamentals)

Temp., °F	Absolute Pressure		Specific Volume, ft³/lb$_w$			Specific Enthalpy, Btu/lb$_w$			Specific Entropy, Btu/lb$_w$·°F			Temp., °F
t	p, psi	p, in. Hg	Sat. Solid/Liq. v_i/v_f	Evap. v_{ig}	Sat. Vapor v_g	Sat. Solid/Liq. h_i/h_f	Evap. h_{ig}	Sat. Vapor h_g	Sat. Solid/Liq. s_i/s_f	Evap. s_{ig}	Sat. Vapor s_g	t
45	0.14755	0.30042	0.01602	2035.91	2035.92	13.05	1067.79	1080.84	0.0262	2.1158	2.1420	45
50	0.17811	0.36263	0.01602	1703.18	1703.20	18.06	1064.96	1083.03	0.0361	2.0895	2.1256	50
55	0.21410	0.43591	0.01603	1430.61	1430.62	23.07	1062.14	1085.21	0.0459	2.0637	2.1096	55
60	0.25635	0.52192	0.01604	1206.30	1206.32	28.07	1059.32	1087.39	0.0555	2.0385	2.094	60
65	0.30574	0.62249	0.01604	1020.98	1021.00	33.07	1056.5	1089.57	0.0651	2.0136	2.0787	65
70	0.36328	0.73964	0.01605	867.34	867.36	38.07	1053.68	1091.75	0.0746	1.9893	2.0639	70
75	0.43008	0.87564	0.01606	739.42	739.44	43.06	1050.85	1093.92	0.084	1.9654	2.0494	75
80	0.50736	1.03298	0.01607	632.54	632.56	48.06	1048.03	1096.08	0.0933	1.942	2.0352	80
85	0.59647	1.21442	0.01609	542.93	542.94	53.05	1045.19	1098.24	0.1025	1.9189	2.0214	85
90	0.69889	1.42295	0.0161	467.52	467.53	58.04	1042.36	1100.4	0.1116	1.8963	2.0079	90

Table Q4(b)(i): Solar heat gain factor (SHGF)

40° N. Lat

	N (Shade)	NNE/ NNW	NE/ NW	ENE/ WNW	E/ W	ESE/ WSW	SE/ SW	SSE/ SSW	S	HOR
Jan.	20	20	20	74	154	205	241	252	254	133
Feb.	24	24	50	129	186	234	246	244	241	180
Mar.	29	29	93	169	218	238	236	216	206	223
Apr.	34	71	140	190	224	223	203	170	154	252
May	37	102	165	202	220	208	175	133	113	265
June	48	113	172	205	216	199	161	116	95	267
July	38	102	163	198	216	203	170	129	109	262
Aug.	35	71	135	185	216	214	196	165	149	247
Sep.	30	30	87	160	203	227	226	209	200	215
Oct.	25	25	49	123	180	225	238	236	234	177
Nov.	20	20	20	73	151	201	237	248	250	132
Dec.	18	18	18	60	135	188	232	249	253	113

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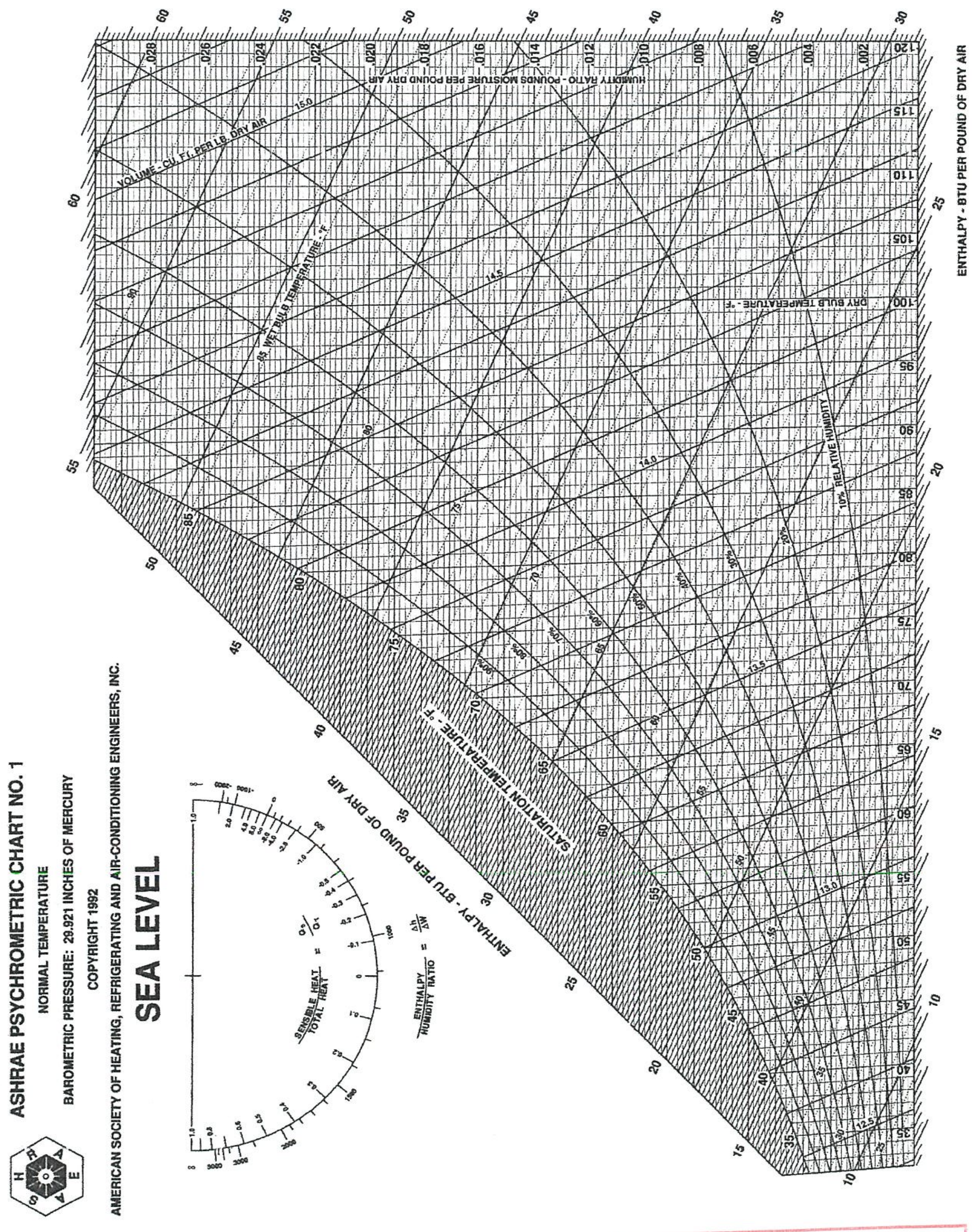

Figure Q3(b)(i): English unit ASHRAE Psychometric Chart

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Table Q4(b)(ii): Shading coefficients (SC) for Glass without or with interior shading devices

Type of Glazing	Nominal Thickness, in (Each light)	Without Shading	With Interior Shading: Venetian Blinds: Medium	Venetian Blinds: Light	Roller Shades: Opaque: Dark	Roller Shades: Opaque: Light	Roller Shades: Translucent: Light
Single glass							
Clear	¼	0.94	0.74	0.67	0.81	0.39	0.44
Heat absorbing	¼	0.69	0.57	0.53	0.45	0.30	0.36
Double glass							
Clear	¼	0.81	0.62	0.58	0.71	0.35	0.40
Heat absorbing	¼	0.55	0.39	0.36	0.40	0.22	0.30

Note: Venetian blinds are assumed set at a 45° position. Adapted with permission from the *1993 ASHRAE Handbook—Fundamentals*.

Table Q4(b)(iii): Cooling load factors (CLF)for glas ith interior shading, North latitudes

Fenestration Facing	Solar Time, h: 0100	0200	0300	0400	0500	0600	0700	0800	0900	1000	1100	1200	1300	1400	1500	1600	1700	1800	1900	2000	2100	2200	2300	2400
N	0.08	0.07	0.06	0.06	0.07	0.73	0.66	0.65	0.73	0.80	0.86	0.89	0.89	0.86	0.82	0.75	0.78	0.91	0.24	0.18	0.15	0.13	0.11	0.10
NNE	0.03	0.03	0.02	0.02	0.03	0.64	0.77	0.62	0.42	0.37	0.37	0.37	0.36	0.35	0.32	0.28	0.23	0.17	0.08	0.07	0.06	0.05	0.04	0.04
NE	0.03	0.02	0.02	0.02	0.02	0.56	0.76	0.74	0.58	0.37	0.29	0.27	0.26	0.24	0.22	0.20	0.16	0.12	0.06	0.05	0.04	0.04	0.03	0.03
ENE	0.03	0.02	0.02	0.02	0.02	0.52	0.76	0.80	0.71	0.52	0.31	0.26	0.24	0.22	0.20	0.18	0.15	0.11	0.06	0.05	0.04	0.04	0.03	0.03
E	0.03	0.02	0.02	0.02	0.02	0.47	0.72	0.80	0.76	0.62	0.41	0.27	0.24	0.22	0.20	0.17	0.14	0.11	0.06	0.05	0.05	0.04	0.03	0.03
ESE	0.03	0.03	0.02	0.02	0.02	0.41	0.67	0.79	0.80	0.72	0.54	0.34	0.27	0.24	0.21	0.19	0.15	0.12	0.07	0.06	0.05	0.04	0.04	0.03
SE	0.03	0.03	0.02	0.02	0.02	0.30	0.57	0.74	0.81	0.79	0.68	0.49	0.33	0.28	0.25	0.22	0.18	0.13	0.08	0.07	0.06	0.05	0.04	0.04
SSE	0.04	0.03	0.03	0.03	0.02	0.12	0.31	0.54	0.72	0.81	0.81	0.71	0.54	0.38	0.32	0.27	0.22	0.16	0.09	0.08	0.07	0.06	0.05	0.04
S	0.04	0.04	0.03	0.03	0.03	0.09	0.16	0.23	0.38	0.58	0.75	0.83	0.80	0.68	0.50	0.35	0.27	0.19	0.11	0.09	0.08	0.07	0.06	0.05
SSW	0.05	0.04	0.04	0.03	0.03	0.09	0.14	0.18	0.22	0.27	0.43	0.63	0.78	0.84	0.80	0.66	0.46	0.25	0.13	0.11	0.09	0.08	0.07	0.06
SW	0.05	0.05	0.04	0.04	0.03	0.07	0.11	0.14	0.16	0.19	0.22	0.38	0.59	0.75	0.83	0.81	0.69	0.45	0.16	0.12	0.10	0.09	0.07	0.06
WSW	0.05	0.05	0.04	0.04	0.03	0.07	0.10	0.12	0.14	0.16	0.17	0.23	0.44	0.64	0.78	0.84	0.78	0.55	0.16	0.12	0.10	0.09	0.07	0.06
W	0.05	0.05	0.04	0.04	0.03	0.06	0.09	0.11	0.13	0.15	0.16	0.17	0.31	0.53	0.72	0.82	0.81	0.61	0.16	0.12	0.10	0.08	0.07	0.06
WNW	0.05	0.05	0.04	0.03	0.03	0.07	0.10	0.12	0.14	0.16	0.17	0.18	0.22	0.43	0.65	0.80	0.84	0.66	0.16	0.12	0.10	0.08	0.07	0.06
NW	0.05	0.04	0.04	0.03	0.03	0.07	0.11	0.14	0.17	0.19	0.20	0.21	0.22	0.30	0.52	0.73	0.82	0.69	0.16	0.12	0.10	0.08	0.07	0.06
NNW	0.05	0.05	0.04	0.03	0.03	0.11	0.17	0.22	0.26	0.30	0.32	0.33	0.34	0.34	0.39	0.61	0.82	0.76	0.17	0.12	0.10	0.08	0.07	0.06
HOR.	0.06	0.05	0.04	0.04	0.03	0.12	0.27	0.44	0.59	0.72	0.81	0.85	0.85	0.81	0.71	0.58	0.42	0.25	0.14	0.12	0.10	0.08	0.07	0.06

Reprinted with permission from the *1989 ASHRAE Handbook—Fundamentals*.

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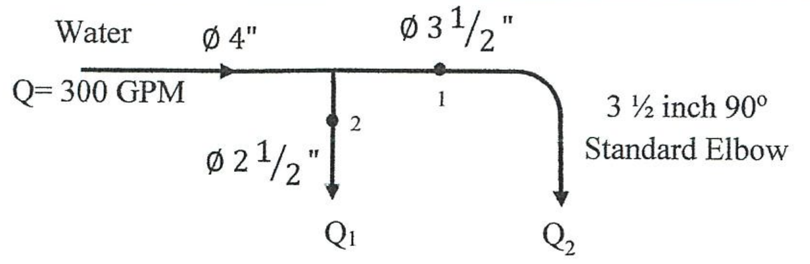

<u>Given: Equivalent length (EL) feet of 90° Elbow Standard</u>

Pipe 3inch = 8 (EL)

Pipe 4 inch = 11 (EL)

Figure Q5(b)(i): Piping system

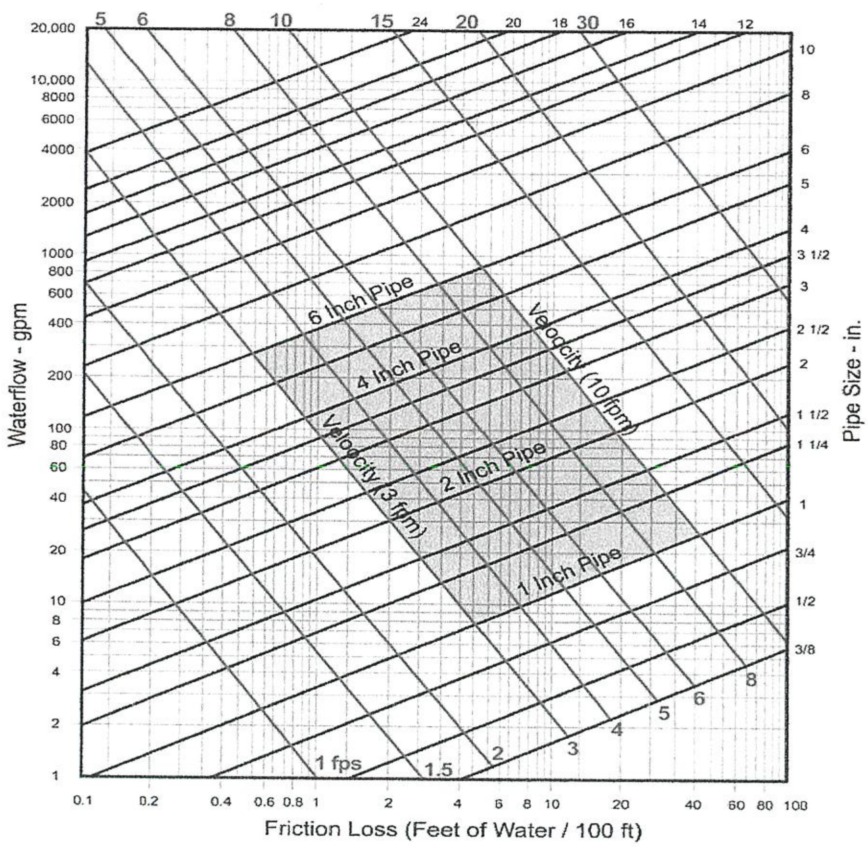

Figure Q5(b)(ii): Friction loss of water Sch. 40 steel pipe – Close system (Sources: Carrier Corporation)

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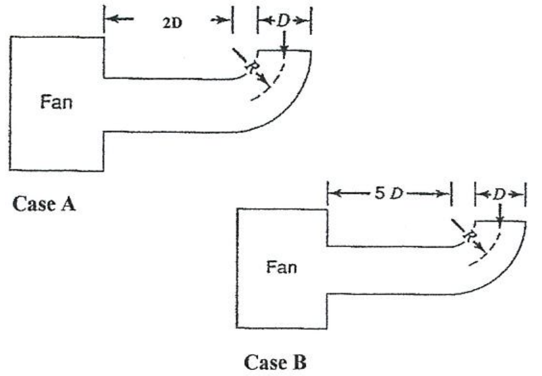

R/D = 1

Loss coefficient C

R/D	Length of Inlet in Diameters: 0	2D	5D.
0.75	1.6	0.8	0.4
1.0	1.3	0.7	0.3
2.0	1.2	0.5	0.25
3.0	0.7	0.4	0.20

Sources from the SMACNA HVAC system – Duct design manual. Second edition, 1981

Figure Q5(c)(i): Ducting system

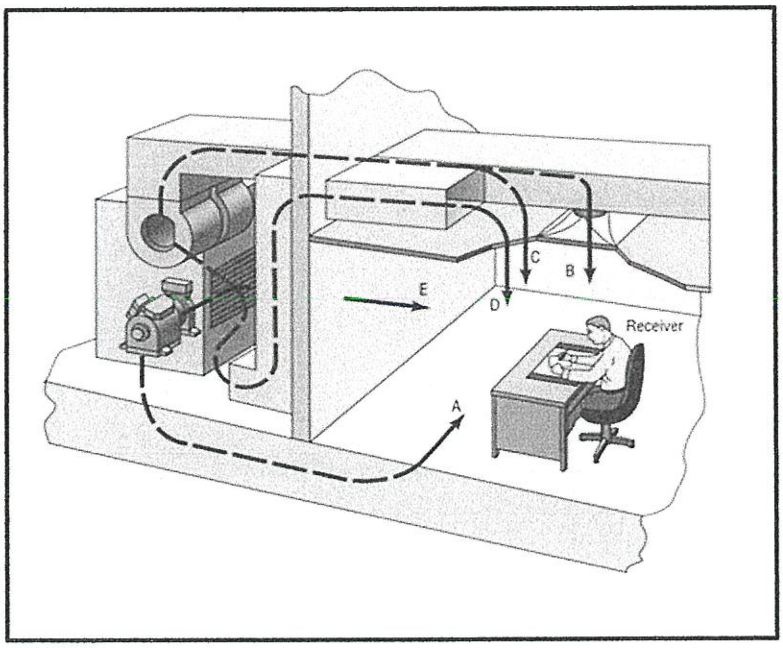

Figure Q6 (b) Typical Paths of Noise and Vibration Propagation in HVAC Systems (ASHRAE, 1995)

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List of equations

Psychometrics

Dry air: $P_aV = n_a RT$

Water vapour: $P_wV = n_w RT$

Humidity Ratio, $W = \frac{m_w}{m_a} = 0.622 \frac{P_w}{P - P_w}$

Humidity Ratio at saturated

at web bulb temperature, $W^* = \frac{m_{ws}}{m_a}$

$$= 0.622 \frac{P_{ws}}{P - P_{ws}}$$

In SI unit

$$W = \frac{(2501 - 2.38\, t^*)W^* - (t - t^*)}{2501 + 1.805\, t - 4.186\, t^*}$$

Where $t\ (dry - bulb)\ and\ t^*(wet - bulb)$ are in °C

In inch-pound units

$$W = \frac{(1093 - 0.556\, t^*)W^* - 0.240(t - t^*)}{1093 + 0.444\, t - t^*}$$

Where $t\ (dry - bulb)\ and\ t^*\ (wet - bulb)$ are in °F

The term W_S^* indicates the humidity ratio if saturated at the wet − bulb temperature

Relative Humidity, $\phi = \frac{P_W}{P_{WS}}$

Degree of Saturation: $\mu = \frac{W}{W_s}$

Dew − point temperature, t_d

For 0°C to 70°C

$$t_d = -35.957 - 1.8726a + 1.1689a^2$$

For −60°C to 0°C

$$t_d = -60.45 + 7.0322a + 0.37a^2$$

with t_d in°C and $a = \ln P_w$ with P_w in pascals.

For 32°F to 150°F

$$t_d = 79.047 + 30.579a + 1.8893a^2$$

For below 32°F

$$t_d = 71.98 + 24.873a + 0.8927a^2$$

with t_d in°F and $a = \ln P_w$ with P_w in (inHg).

Dalton Laws

$$P = P_a + P_w$$

Volume, v of a moist air mixture,

$$v = {R_aT}/{(P - P_w)}$$

Where, $R_a = 0.287$ kJ/kg.K

Enthalpy

$$h = t + W(2501 + 1.805t)(\text{kJ/kg dry air})$$

t is dry − bulb temperature °C (S − I unit)

$$h = 0.24t + W(1061 + 0.4444\, t)(\text{Btu/lb dry air})$$

t is dry − bulb temperature °F (I − P unit)

Where

P = Total of air pressure

P_a = Partial pressure of dry air

P_w = Partial pressure of water vapour

V = total mixture volume

n_a = number of moles of dry air

n_w = number of moles of water vapour

R = universal gas constant (8.3144 J/g − mol.K or 1545.32 ft.lb_f/1b − mol.R)

T=absolute temperature

m_a = mass of dry air

m_w = mass of water vapour

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P_{ws} = saturation pressure of water vapour at the given temperature

W_s = humidity ratio at saturated at web bulb temperature

Cooling load cause by conduction heat gain through Exterior structure (roof, Walls and glass)

$Q = U \times A \times CLTD_c$

Where

Q = cooling load for roof, wall, or glass, BTU/hr
U = overall heat transfer coefficient for roof, wall or glass, $BTU/hr - ft^2 - F$

Conduction through Interior Structure (Partitions, Floors and ceiling)

$Q = U \times A \times TD$

Where,
Q = Heat gain (Cooling load)through partition, floor or ceiling, BTU/hr
U = overall heat transfer coefficient for partition, floor or ceiling, $BTU/hr - ft^2 - F$
TD = temperature difference between uncoditioned and conditioned space, F
A = area of partition, floor or ceiling, ft^2

Solar Radiation through glass

$Q = SHGF \times A \times SC \times CLF$

Where,
Q = Solar radiation cooling load for glass, BTU/hr
$SHGF$ = maximum solar heat gain factor, $BTU/_{hr - ft^2}$
A = area of glass, ft^2
SC = shading coefficient
CLF = cooling load factor for glass

Cooling Load from Lighting

$Q = 3.4 \times W \times BF \times CLF$

Where,
Q = Heat gain (Cooling load) from lighting, BTU/hr
W = lighting capacity, watts
BF = ballast factor
CLF = cooling load factor for lighting

A = area of roof, wall or glass, ft^2
$CLTD_c$ = corrected cooling load temperature difference, F

$$CLTD_c = CLTD + LM + (78 - t_R) + (t_a - 85)$$

$CLTD$ = cooling load temperature difference, F
LM = correction for latitude and month, F
t_R = room temperature difference, F
t_a = average outside temperature on a design day, F

$$t_a = t_o - (DR/2)$$

t_o = outside design dry bulb temperature, F
DR = daily temperature range, F

Cooling Load from People

$Q_s = q_s \times n \times CLF$

$Q_l = q_l \times n$

Where,
Q_s, Q_l = sensible and latent heat gains, (loads)
q_s, q_l = sensible and latent heat gains per person
n = number of people
CLF = cooling load factor for people

TERBUKA

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THERMAL COMFORT

Predicted mean vote (PMV)

$$PMV = (0.303e^{-0.036M} + 0.028)L$$

Where:

M is Metabolic rate, in Watt per square meter (W/m^2);

L is thermal load

$$L = [(M - W) - H - E_C - C_{res} - E_{res}]$$

Where:

W = the effective mechanical power, in Watt per square meter (W/m^2);

H = the sensitive heat losses,

E_c = the heat exchange by evaporation on the skin;

C_{res} = heat exchange by convection in breathing;

E_{res} = the evaporative heat exchange in breathing.

$$H = 3.96 * 10^{-8} * f_{cl} * [(t_{cl} + 273)^4 - (t_r + 273)^4] - f_{cl} * h_c * (t_{cl} - t_a)$$

$$E_c = 3.05 * 10^{-3} * [5733 - 6.99 * (M-W) - p_a] - 0.42 * [(M-W) - 58.15]$$

$$C_{res} = 0.0014 * M * (34 - t_a)$$

$$E_{res} = 1.7 * 10^{-5} * M * (5867 - p_a)$$

where:

I_{cl} = the clothing insulation, in square meters Kelvin per watt (m^2 K/W);

f_{cl} = the clothing surface area factor;

t_a = the air temperature, in degrees Celsius (°C);

t_r = the mean radiant temperature, in degrees Celsius (°C);

v_{ar} = the relative air velocity, in meters per second (m/s);

pa = the water vapor partial pressure, in Pascal (Pa);

t_{cl} = the clothing surface temperature, in degrees Celsius (°C).

$$(t_{sk} - t_{cl}) / I_{cl} = 3.96 * 10^{-8} * f_{cl} * [(t_{cl} + 273)4 - (t_r + 273)4] + f_{cl} * h_c * (t_{cl} - t_a)$$

from which we can express t_{cl}:

$$t_{cl} = t_{sk} - I_{cl} * 3.96 * 10^{-8} * f_{cl} * [(t_{cl} + 273)4 - (t_{eq} + 273)4] - I_{cl} * f_{cl} * h_c * (t_{cl} - t_{eq}),$$

where t_{sk} is the skin external temperature,

calculated from: $t_{sk} = 35.7 - 0.028\ (M-W)$.

and,

fcl = the clothing surface area factor;

for $I_{cl} \leq 0.078\ m^2.K/W$

$$f_{cl} = 1.00 + 1.290 I_{cl}$$

for $I_{cl} > 0.078\ m^2.K/W$

$$f_{cl} = 1.05 + 0.645 I_{cl}$$

Predicted per cent dissatisfied (PPD)

$$PPD = 100 - 95 * e^{-(0.03353.PMV^4 + 0.2179.PMV^2)}$$