UTHM
Universiti Tun Hussein Onn Malaysia

UNIVERSITI TUN HUSSEIN ONN MALAYSIA

FINAL EXAMINATION
SEMESTER I
SESSION 2019/2020

COURSE NAME	:	AIR CONDITIONING SYSTEM DESIGN
COURSE CODE	:	BDE40103
PROGRAMME	:	BDD
EXAMINATION DATE	:	DECEMBER 2019 / JANUARY 2020
DURATION	:	3 HOURS
INSTRUCTION	:	**PART A**: ANSWER **FOUR (4)** QUESTIONS **ONLY** FROM **FIVE (5)** QUESTIONS. **PART B**: ANSWER **ALL** QUESTIONS.

THIS QUESTION PAPER CONSISTS OF **NINE (9)** PAGES

PART A

Q1 (a) Refrigerants are the working fluids in refrigeration, air conditioning, and heat-pumping systems. Describes five (5) classification of refrigerants.

(10 marks)

(b) Justify and sketch a diagram for all water system and air-water system in an air conditioning system central air conditioning systems using chilled water. Describes each components such as Return air, Air handling Unit (AHU), Pump, Chiller, Cooling Tower, Compressor, Condensor, Thermostatic expansion valve (TEV) and an evaporator of this systems

(10 marks)

Q2 (a) ASHRAE Standard 55-2004, "Thermal Environmental Conditions for Human Occupancy," is a revision of Standard 55-1992. Explains six (6) factors affecting thermal comfort are both environmental and personal.

(10 marks)

(b) According ISO 7730: 1994, the predicted mean vote (PMV) and predicted percent dissatisfied (PPD) form are therefore closely related, and both indices take the form of a U-shaped relationship for thermal comfort. Compare between PMV and PPD.

(10 marks)

Q3 (a) A room of dimensions 4 m by 6 m by 2.4 m contains an air-water vapor mixture at a total pressure of 100 kPa and a temperature of 25°C. The partial pressure of the water vapor is 1.4 kPa. Estimate:

i. Humidity ratio, W;
ii. Dew point,(T_{dp}); and
iii. Total mass of water vapor in the room, m_w.

if T_{dp} =11°C for Pressure 1.3127 kPa, T_{dp} = 12 °C for Pressure 1.4026 kPa, and gas constant, R_a = 0.287 kJ/kg.K

(5 marks)

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(b) Given room conditions of 24°C dry bulb with saturation pressure of water vapor (P_{ws}) is 2.9852 kPa, atmospheric pressure (P) is 101.325 kPa and 60% RH, justify for the air vapor mixture without using the psychrometric charts:

i. Humidity ratio, W;

ii. Enthalpy, h;

iii. Dew-point temperature, Tdp ;

iv. Specific volume, v; and

v. Degree of saturation, μ if humidity ratio (W_s).

(15 marks)

Q4 (a) Explains the cooling load and the important of cooling load.

(4 Marks)

(b) Find the value of cooling load for each problem:

i. A 30 ft by 40 ft roof of a building in UTHM building is constructed of 4 in. heavy weight concrete with 1 in. insulation and suspended ceiling. The inside temperature (t_R) is 76 °F. if cooling load temperature different (CLTD) for this roof is 29 °F, correction for latitude and month (LM) for July is 1°F, outside design dry bulb temperature (t_o) is 86 °F, and overall heat transfer coefficient for roof, wall, or glass, U is 0.128 BTU/hr-ft^2-F. Find the roof cooling load.

(8 Marks)

ii. Radiant energy from the sun passes through transparent material such as glass and becomes a heat gain to the room. A building wall facing southwest has a window area of 240 ft^2. The glass is ¼ inch single clear glass with light-coloured interior venetian blinds with shading coefficient (SC) is 0.67. The building is of medium construction and is located at 40°N latitude, solar time is 3 PM at August with solar heat gain factor (SHGF) is 196 BTU/hr.ft^2 and cooling load factor (CLF) is 0.83. Find the solar cooling load.

(8 Marks)

Q5 (a) A fan is a gas flow producing machine which operates on the same basic principles as a centrifugal pump or compressor. A fan is similar to a centrifugal pump or compressor, which is convert mechanical rotated energy, applied at the shafts, to gas (or fluid) energy. Explain, the following terms for fan performance:

i. fan Volume;
ii. fan Outlet Velocity;
iii. fan Velocity Pressure; and
iv. fan Total Pressure.

(8 marks)

(b) The piping system shown in **Figure 5(b)** is to delivery water from basement to the roof storage tank, 55 m above. The friction loss in the piping, valves, and fitting is 4 m. The water inters the pump at a pressure gage of 3 m and is delivered at atmospheric pressure (all value is gage pressure). The velocity at the pump suction is 0.7 m/s and at the piping exit is 3 m/s. Evaluate the required pump pressure for this problem.

(12 marks)

PART B

Q6 (a) According ASHRAE 2009, HVAC equipment for a building is one of the major sources of interior noise, and its effect on the acoustical environment. Further, noise from equipment located outdoors often propagates to the community. Explain the characteristics of noise:

i. Sound;
ii. Cycle period;
iii. The frequency;
iv. Wavelength; and
v. Decibel.

(5 marks)

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

(15 marks)

END OF QUESTION

FINAL EXAMINATION

SEMESTER / SESSION : SEM 1 / 2019-2020 PROGRAMME : 4 BDD
COURSE : AIR CONDITIONING SYSTEM DESIGN COURSE CODE : BDE40103

List of equations

Psychometrics

Dry air: $P_aV = n_a RT$

Water vapor: $P_wV = n_w RT$

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

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

$h = t + W(2501 + 1.805t)$(kJ/kg dry air)

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

$h = 0.24t + W(1061 + 0.4444\ t)$(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 vapor

V = total mixture volume

n_a = number of moles of dry air

n_w = number of moles of water vapor

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 vapor

P_{ws} = saturation pressure of water vapor 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

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

FINAL EXAMINATION

SEMESTER / SESSION : SEM 1 / 2019-2020 PROGRAMME : 4 BDD
COURSE : AIR CONDITIONING SYSTEM DESIGN COURSE CODE : BDE40103

List of equations

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

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

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).

FINAL EXAMINATION

SEMESTER / SESSION : SEM 1 / 2019-2020 PROGRAMME : 4 BDD
COURSE : AIR CONDITIONING SYSTEM DESIGN COURSE CODE : BDE40103

List of equations

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

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 percent dissatisfied (PPD)

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

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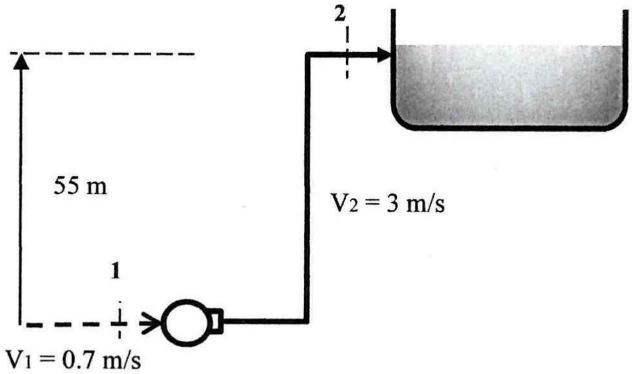

Figure 5(b)

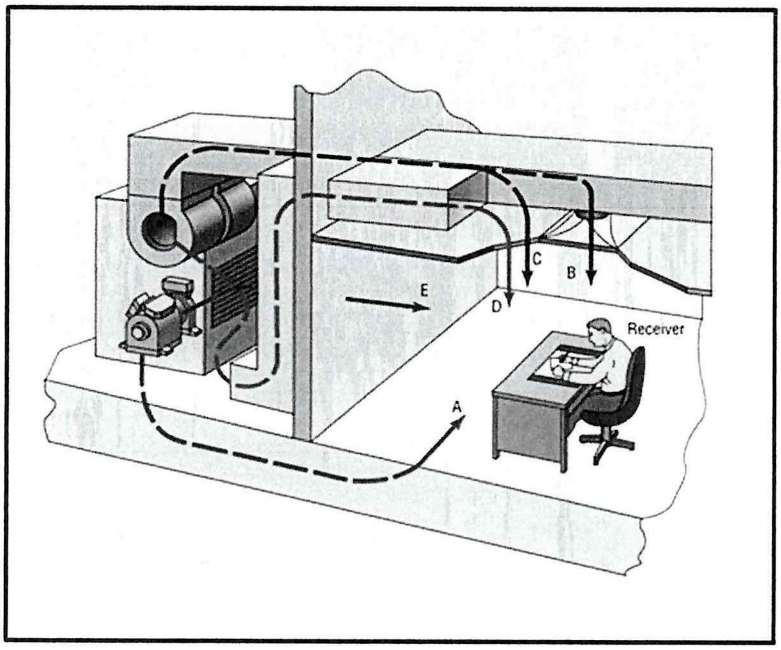

Figure Q6 (b) Typical Paths of Noise and Vibration Propagation in HVAC Systems (ASHRAE, 1995)