CXT EXAM

2025

ENGINEERING HVAC EQUATIONS

Commissioning Specialist Curriculum

BEYOND SMART CITIES

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The commissioning specialist can utilize HVAC design and engineering formulas to address issues in both new and existing buildings.

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BEYOND SMART CITIES

HVAC and Engineering Equations

Specialists in commissioning services can use engineering and HVAC design formulas to implement energy analysis, HVAC design, lighting design, heat gain design, measurement, and verification processes in both new and existing buildings, as well as to diagnose problems.

GEOMETRY

- Area of circle = $\Pi * r^2$
- Circumference of circle = $\Pi * D$
- Area of rectangle = L * W
- Perimeter of rectangle = 2*L + 2*W

ELECTRICAL

- V = I * R   Ohm's Law   (single phase)
- $P = I^2 * R$ Ohm's law for power   (single phase)
- kW = (V × A × 1.73 × PF)/1000   (3 phase) PF = Cos phi
- kW = kVA * PF   PF = Cos phi
- kVA = 1.73 x kV x A (3 phase)

LIGHTING

- RCR = (2.5 × cavity height × perimeter)/(area)   General
- RCR = (5 x height x (L + W))/(L*W) Rectangular room
- Number of lamps = (LUXdesired × area)/(Lumens-per-lamp × Cu × L1 × L2)
- Lux = Lumens/d^2
- Efficacy = Lumens/watt

HEAT FLOW

- $q = u \times A \times \Delta t$ W
- $q = M \times Cp \times \Delta T$ W   sensible heat only
- $q = M \times \Delta h$ W   sensible and latent heat
- Q cooling = U × A × 24 × CDD   Wh/yr
- Q heating= U × A × 24 × HDD   Wh/yr

MOTORS

- Percent load = (NLRPM-RPM)/(NLRPM - FLRPM)
- Synchronous speed = (frequency × 50)/(number of pole pairs)
- kW × Percent load = (V × A × 1.73 × PF × Efficiency)   (3 phase)
- PF = kW/kVA = Cos phi

FANS AND FAN LAWS

- LPSn/LPSo = (RPMn/RPMo)   n=new; o=old
- SPn /SPo = $(RPMn /RPMo)^2$
- kWn/kWo = $(RPMn /RPMo)^3$
- RPM-driven × diameter-driven = RPM-driver × diameter-driver

- Percent OA = (RAT - MAT)/(RAT - OAT)

MISCELLANEOUS

- $q = LPS \times 1.2 \times \Delta T$ W (sensible heat for air)
- $q = LPS \times 4.2 \times \Delta t$ kW (sensible heat for water)
- POU energy cost = (output/efficiency) x (cost per unit)
- Density of dry air at standard conditions 1.204 kg/m^3
- Specific heat of dry air at standard conditions 1.006 kJ/ kg°C
- Density of water at standard conditions 1 kg/L
- Specific heat of water at standard conditions 4.2 kJ/kg°C
- Density of water 1000 kg/m^3

COOLING AND HEATING EQUATIONS

For SI:

- **Sensible Heat**

  $HS = cp\ \rho\ q\ \Delta T$
- **Latent Heat**

  $HL = c1 \rho q \Delta W$
- **Total Heat**

  $HT = \rho q \Delta h$
- **Sensible Heat Ratio:**

  $SHR = HS/HT$

Where:

H_S = Sensible Heat (kW)

H_L = Latent Heat (kW)

H_T = Total Heat (kW)

ΔT = Temperature Difference (°K) q = Air Volume Flow (m_3/s)

ρ = Density of Air (1.202 kg/m_3)

*C_p = **Specific Heat** of Air (1.0 kJ/kg.K)*

C_1 = Air Latent Factor (a typical value 3010)

*ΔW = Humidity Ratio Difference (kg water/kg dry air) Δh = **Enthalpy** Difference (kJ/kg)*

SHR = Sensible Heat Ratio

For I-P:

- **Sensible Heat**
  $H_S = 1.085 \times CFM \times \Delta T$
- **Latent Heat**
  $H_L = 0.68 \times CFM \times \Delta WGR = 4840 \times CFM \times \Delta WLB$
- **Total Heat**

$H_T = 4.5 \times CFM \times \Delta h$

• Sensible Heat Ratio

$SHR = H_S / H_T$

Where:
H_S = Sensible Heat (Btu/hr)
H_L = Latent Heat (Btu/hr)
H_T = Total Heat (Btu/hr)
ΔT = Temperature Difference (°F)
ΔWGR. = Humidity Ratio Difference (Gr.H_2O/Lb.DA)

ΔWLB. = Humidity Ratio Difference (Lb.H_2O/Lb.DA)

Δh = Enthalpy Difference (Btu/Lb.DA)

CFM = Air Flow Rate (Cubic Feet per Minute) SHR = Sensible Heat Ratio

Thermal Resistance **R-Values/U-Values**

A. Thermal Value (R-Value)

$R = t / k$

B. Thermal Transmittance (U-Value)

$U = 1 / \Sigma R$

Where:

For SI:

k = Thermal Conductivity (W/(m·K))
R = Thermal Resistance ((m2·K)/W)
U = Thermal Transmittance (W/(m2·K)) t = Thickness (m)
ΣR = Sum of the Individual R-Values

For I-P:

k = Thermal Conductivity (Btu./hr. ft. °F.)
R = Thermal Resistance (hr. ft2. °F./Btu.)
U = Thermal Transmittance (Btu./hr. ft2. °F.) t = Thickness (ft)

WATER SYSTEM EQUATIONS

For SI:

H= ρ•q•cp•ΔT

q (Evap) = H (ρ•cp•ΔT)

Where:
H = Total Heat (kW)
q = Water Flow Rate (m^3/s)
ρ = Density of Water (997 kg/m^3)

cp = Specific Heat of Water (4.187 kJ/kg.K) ΔT = Temperature Difference (°K)

For I-P:

H = (GPM • ΔT) / 24

GPM (Evap) = (H • 24) /ΔT

GPM (Cond) = (H • 30) /ΔT

Where:
H = Total Heat (Tons of Refrigerant)
ΔT = Temperature Difference (°F)
GPM = Water Flow Rate (Gallons per Minute)
GPM (Evap). = Evaporator Water Flow Rate (Gallons per Minute)

GPM (Cond). = Condenser Water Flow Rate (Gallons per Minute)

AIR CHANGE RATE EQUATIONS

For SI:

ACH = (q • 3600) / V

Where:
ACH. = Air Change Rate per Hour

q = Air Volume Flow (m_3/s)
V = Space Volume (m_3)

For I-P:

ACH = (CFM • 60) / V

Where:
ACH. = Air Change Rate per Hour
CFM = Air Volume Flow (cubic feet per minute)

V = Space Volume (ft3)

Mixed Air Temperature

$$T_{MA} = \left(T_{RA} \cdot \left(\frac{Q_{RA}}{Q_{SA}}\right)\right) + T_{OA} \cdot \left(\frac{Q_{OA}}{Q_{SA}}\right)$$

Where:

For SI:

Q_{SA}= Supply Air (L/s)
Q_{RA} = Return Air (L/s)
Q_{OA} = Outside Air (L/s)
T_{MA} = Mixed Air Temperature (°C) T_{RA} = Return Air Temperature (°C) T_{OA} = Outside Air Temperature (°C)

For I-P:

Q_{SA}= Supply Air (CFM)
Q_{RA} = Return Air (CFM)
Q_{OA} = Outside Air (CFM)
T_{MA} = Mixed Air Temperature (°F) T_{RA} = Return Air Temperature (°F) T_{OA} = Outside Air Temperature (°F)

DUCTWORK EQUATIONS

• Total Pressure

$p_t = p_s + p_v$ p_s = Static Pressure (Pa)

Where:
p_t = Total Pressure (Pa)

p_v = Velocity Pressure (Pa)

• Velocity

V=Q / A

Where:

For SI:

V = Fluid Mean Velocity (m/s)
Q = Volumetric Flow Rate (m3/s)
A = Cross-Sectional Area of Duct (m2)

For I-P:

V = Fluid Mean Velocity (FPM)
Q = Volumetric Flow Rate (CFM)
A = Cross-Sectional Area of Duct (ft2)

Fan Affinity Laws

A. Flow Rate

$$Q_1 = Q_2 \cdot (\frac{N_1}{N_2})$$

B. Static Pressure

$$P_1 = P_2 \cdot \left(\frac{N_1}{N_2}\right)^2$$

C. Electrical Power

$$W_1 = W_2 \cdot \left(\frac{N_1}{N_2}\right)^3$$

Where:

Where:

For SI:

Q = Volumetric Flow Rate (m3/s)
N = Rotational Speed, Revolutions Per Minute (RPM) P = Static Pressure (Pa)
W = Electrical Power (W

For I-P:

Q = Volumetric Flow Rate (CFM)
N = Rotational Speed, Revolutions Per Minute (RPM) P = Static Pressure (in.wg)
W = Electrical Power (W)

Pump Affinity Laws (At Constant Pump Impeller Diameter)

Where:

- Flow Rate

$$Q_1 = Q_2 \cdot (\frac{N_1}{N_2})$$

- Pump Head

$$P_1 = P_2 \cdot \left(\frac{N_1}{N_2}\right)^2$$

- Electrical Power

$$W_1 = W_2 \cdot \left(\frac{N_1}{N_2}\right)^3$$

Where:

For SI:

Q = Volumetric Flow Rate (m3/s)
N = Rotational Speed, Revolutions Per Minute (RPM) P = Pump Head (bar)
W = Electrical Power (W)

For I-P:

Q = Volumetric Flow Rate (GPM)
N = Rotational Speed, Revolutions Per Minute (RPM) P = Static Pressure (ft.wg)
W = Electrical Power (W)

Pump Net Positive Suction Head (NPSH) Calculations

$NPSH_{AVAIL} > NPSH_{REQ'D}$

Net Positive Suction Head Available:

$NPSH_{AVAIL} = H_A \pm H_S - H_F - H_{VP}$

Where:

For SI:

$NPSH_{AVAIL}$ = Net Positive Suction Available at Pump (m)
$NPSH_{REQ'D}$ = Net Positive Suction Required at Pump (m)
H_A = Pressure at Liquid Surface (m—10.2 m for Water at Atmospheric Pressure) H_S = Height of Liquid Surface Above (+) or Below (−) Pump (m)
H_F = Friction Loss between Pump and Source (m)
H_{VP} = Absolute Pressure of Water Vapor at Liquid Temperature (m)

For I-P:

$NPSH_{AVAIL}$ = Net Positive Suction Available at Pump (ft)
$NPSH_{REQ'D}$ = Net Positive Suction Required at Pump (ft)
H_A = Pressure at Liquid Surface (ft—34 ft for Water at Atmospheric Pressure)

H_S = Height of Liquid Surface Above (+) or Below (−) Pump (ft)
H_F = Friction Loss between Pump and Source (ft)

H_{VP} = Absolute Pressure of Water Vapor at Liquid Temperature (ft)

EFFICINEECIES

For SI:

• Coefficient of Performance (COP)

COP = Total Cooling Capacity (W) /(Compressor Input Power (W) + Condenser Fan Input Power (W))

• Energy Efficiency Ratio (EER)

$$EER = \frac{\text{Net Cooling Capacity (W)} \cdot 3.413}{\text{Total Input Power (W)}}$$

For I-P:

• Coefficient of Performance (COP)

$$COP = \frac{\text{Total Cooling Capacity (BTU/h)}}{(\text{Compressor (W)} + \text{Condenser Fan (W)}) \cdot 3.413}$$

• Energy Efficiency Ratio (EER)

EER = Net Cooling Capacity (BTU/h) / Total Input Power (W)

Cooling Towers and Heat Exchangers

APPROACH (COOLING TOWER) = LWT – AWB

APPROACH (HEAT EXCHANGER) = EWT_{HS} – LWT_{CS}

RANGE = EWT – LWT

Where:

For SI:

EWT = Entering Water Temperature (°C) LWT = Leaving Water Temperature (°C) AWB = Ambient Wet Bulb Temperature (°C) HS = Hot Side

CS = Cold Side

For I-P:

EWT = Entering Water Temperature (°F) LWT = Leaving Water Temperature (°F) AWB = Ambient Wet Bulb Temperature (°F) HS = Hot Side

CS = Cold Side

External Heat Gain

Fenestration Transmission

For SI:

$$q_c = UA(T_{out} - T_{in})$$

where

q = fenestration transmission heat gain, W

U = overall U-factor, including frame and mounting orientation from Table 4 of Chapter 15, $W/(m^2 \cdot K)$

A = window area, m^2

T_{in} = indoor temperature, °C

T_{out} = outdoor temperature, °C

For I-P:

$$q_c = UA(T_{out} - T_{in})$$

where

q = fenestration transmission heat gain, Btu/h

U = overall U-factor, including frame and mounting orientation from Table 4 of Chapter 15, $Btu/h \cdot ft^2 \cdot °F$

A = window area, ft^2

T_{in} = indoor temperature, °F

T_{out} = outdoor temperature, °F

Internal Heat Gain

Occupants

For SI:

$$q_s = q_{s,per} N$$

$$q_l = q_{l,per} N$$

where

q_s	=	occupant sensible heat gain, W
q_l	=	occupant latent heat gain, W
$q_{l,per}$	=	latent heat gain per person, W/person; see Table 1
N	=	number of occupants

For I-P:

$$q_s = q_{s,per} N$$

$$q_l = q_{l,per} N$$

where

q_s	=	occupant sensible heat gain, Btu/h
q_l	=	occupant latent heat gain, Btu/h
$q_{l,per}$	=	latent heat gain per person, Btu/h·person; see Table 1
N	=	number of occupants

Lighting

For SI:

$$q_{el} = WF_{ul} F_{sa}$$

where

q_{el} = heat gain, W
W = total light wattage, W
F_{ul} = lighting use factor
F_{sa} = lighting special allowance factor

For I-P:

$$q_{el} = 3.41 WF_{ul} F_{sa}$$

where

q_{el} = heat gain, Btu/h
W = total light wattage, W
F_{ul} = lighting use factor
F_{sa} = lighting special allowance factor
3.41 = conversion factor

Ventilation and Infiltration Air Heat Gain

For SI:

$$q_s = 1230 Q_s \Delta t$$

$$q_l = 1.20 \times 2500 Q_s \, \Delta W = 3000 Qs \, \Delta W$$

where

q_s = sensible heat gain due to infiltration, W
q_l = latent heat gain due to infiltration, W
Q_s = infiltration airflow at standard air conditions, L/s
t_o = outdoor air temperature, °C
t_i = indoor air temperature, °C
W_o = outdoor air humidity ratio, kg/kg
W_i = indoor air humidity ratio, kg/kg
1230 = air sensible heat factor at standard air conditions, (W·s)/(m^2·K)
3000 = air latent heat factor at standard air conditions, (W·s)/(m^2·K)

For I-P:

$$q_s = 1.10 Q_s \,\Delta t$$

$$q_l = 60 \times 0.075 \times 1076 Q_s \,\Delta W = 4840 Q_s \,\Delta W$$

where

q_s = sensible heat gain due to infiltration, Btu/h

q_l = latent heat gain due to infiltration, Btu/h

Q_s = infiltration airflow at standard air conditions, cfm

t_o = outdoor air temperature, °F

t_i = indoor air temperature, °F

W_o = outdoor air humidity ratio, lb/lb

W_i = indoor air humidity ratio, lb/lb

1.10 = air sensible heat factor at standard air conditions, Btu/h·cfm(

4840 = air latent heat factor at standard air conditions, Btu/h·cfm(

GHG emissions =

Activity data × Emission factor

$$\textbf{Inventory data} = \frac{\text{Factor}_{\text{Inventory data}}}{\text{Factor}_{\text{Available data}}} \times \text{Available data}$$

Available data	Activity (or emissions) data available which needs to be scaled to align with the inventory boundary
Inventory data	Activity (or emissions) data total for the city
$\text{Factor}_{\text{Inventory}}$	Scaling factor data point for the inventory
$\text{Factor}_{\text{Available data}}$	Scaling factor data point for the original data