

01	☐ A	☐ B	☐ C	☐ D	☒ E
02	☒ A	☐ B	☐ C	☐ D	☐ E
03	☒ A	☐ B	☐ C	☐ D	☐ E
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38	☒ A	☐ B	☐ C	☐ D	☐ E
39	☒ A	☐ B	☐ C	☐ D	☐ E

PRACTICE QUESTIONS

Saudi Council of Engineers

PE006

Practice of Engineering (PE) - HVAC

EDITION

Demo

QUESTIONS

9

FORMAT

Limited Edition PDF

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QUESTION 1

MULTIPLE CHOICE

An air-handling unit supplies air at 13 °C DB (dry-bulb) and 11 °C WB (wet-bulb) to a room with a sensible heat ratio (SHR) of 0.80. Which two of the following statements about the supply air state and its relation to the room condition are correct? (Choose two.)

- ☐ **A** The supply air dew point equals the dew point of air at 13 °C DB / 50% RH, approximately 2.7 °C.
- ☒ **B** Because the room SHR equals 0.80, the supply air temperature lies on the room sensible heat ratio line drawn between the room condition and the apparatus dew point.
- ☐ **C** A supply air state of 13 °C DB / 11 °C WB implies a relative humidity of 100% at the coil surface.
- ☒ **D** The enthalpy of the supply air is lower than the enthalpy of the room return air.
- ☐ **E** Lowering the leaving-air temperature from the coil without changing the coil selection would necessarily decrease the supply air enthalpy.

ANSWER B - D

WHY Option B is correct: in HVAC cooling design the supply air state must lie on the SHR line connecting room condition to the apparatus dew point (ADP) of the cooling coil; this is the fundamental room/SHR/ADP relationship. Option D is correct: cooling and dehumidification remove both sensible and latent heat, so the supply air enthalpy is lower than the return (room) air enthalpy. Option A is wrong: a supply condition of 13 °C DB / 11 °C WB does not imply 50% RH — WB and DB together define the state, and 11 °C WB gives a much higher RH. Option C is wrong: the coil surface approaches saturation but does not reach it; the apparatus dew point corresponds to air at $\approx 90\text{--}95\%$ RH, not 100%. Option E is wrong: leaving-air temperature is a function of coil selection and bypass factor; lowering the leaving-air temperature typically increases the coil surface area/rows and decreases bypass factor, but the relationship to supply enthalpy depends on the resulting humidity as well.

QUESTION 2

MULTIPLE CHOICE

An engineer is evaluating a chilled-water system for an office building. Which two of the following statements correctly describe standard design practice for primary/secondary chilled-water pumping? (Choose two.)

- ☒ **A** A primary–secondary pumping arrangement uses a decoupled common pipe to prevent the chiller and the building loop from interfering hydraulically.
- ☐ **B** In a primary-only (variable-primary-flow) system, the chilled-water flow through each chiller is intentionally varied with building load.
- ☐ **C** The three-way valve at each air-handling unit is typically used in a variable-primary-flow chilled-water plant.
- ☐ **D** Constant primary flow through each chiller is required to maintain stable chiller evaporator tube velocity and avoid oil return problems.
- ☒ **E** Two-way valves at the air-handling unit coils are typical for a primary–secondary pumping arrangement so that the secondary loop is variable-flow.

ANSWER A • E

WHY Option A is correct: in a primary–secondary system the common pipe decouples the primary (constant-flow-through-chillers) loop from the secondary (variable-flow-to-coils) loop, preventing hydraulic interaction. Option E is correct: secondary-side air-handling units use two-way control valves so the secondary loop can vary flow with load while the primary remains constant. Option B is wrong: in primary-only (VPF) designs the chiller flow itself is varied, but minimum flow must still be maintained across each operating chiller — this is a known operational challenge, not an intentional design feature. Option C is wrong: three-way valves are used in constant-secondary-flow systems; variable-primary-flow systems use two-way valves at the coils. Option D is partially true (minimum chiller flow is needed for oil return and tube velocity) but overstates it — it is the minimum, not 'constant' primary flow that is required, so the statement as written is not a correct description of standard design practice.

QUESTION 3

SINGLE CHOICE

A finned-tube heating coil has an overall heat-transfer coefficient $U = 50 \text{ W/m}^2\cdot\text{K}$ and a total external surface area $A = 20 \text{ m}^2$. Air enters at 20°C with a mass flow of 1.5 kg/s ($c_p = 1.005 \text{ kJ/kg}\cdot\text{K}$) and hot water flows inside the tubes such that the LMTD is 25 K . Using the effectiveness–NTU method with $C_{\min}/C_{\max} = 0.85$, which statement best describes the coil performance?

- ☒ **A** The coil heat duty is approximately 25.0 kW and the air-side temperature rise is 16.6°C .
- ☐ **B** The coil heat duty is approximately 12.5 kW and the air outlet temperature is 28.3°C .
- ☐ **C** The coil heat duty is approximately 25.0 kW and the air outlet temperature is 36.6°C .
- ☐ **D** The coil heat duty is approximately 50.0 kW and the air outlet temperature is 53.2°C .

ANSWER A

WHY Using the simple LMTD estimate, $Q \approx U \cdot A \cdot \text{LMTD} = 50 \times 20 \times 25 = 25,000 \text{ W} = 25 \text{ kW}$. Air temperature rise $\Delta T = Q / (\dot{m} \cdot c_p) = 25,000 / (1.5 \times 1005) \approx 16.6^\circ\text{C}$, giving an outlet temperature of about 36.6°C , not 28.3°C or 53.2°C . Option B is wrong because 12.5 kW understates the duty; C is wrong because 36.6°C is the outlet but the option pairs it with the same 25 kW and a wrong outlet; A correctly states both the duty (25 kW) and the temperature rise (16.6°C). The $C_{\min}/C_{\max}$ value of 0.85 is provided as context; for a coil of this size the effectiveness is high enough that the LMTD-based estimate closely matches the NTU result, which is why 25 kW / 16.6°C remains the best answer.

QUESTION 4

SINGLE CHOICE

According to widely adopted energy-efficiency codes (e.g., ASHRAE Standard 90.1 / IECC) for commercial buildings, an economizer on a packaged rooftop unit is generally required when the unit's cooling capacity exceeds which threshold?

- ☐ A 1.5 tons (≈ 5.3 kW)
- ☒ B 5 tons (≈ 17.6 kW)
- ☐ C 10 tons (≈ 35.2 kW)
- ☐ D 30 tons (≈ 105.5 kW)

ANSWER B

WHY Under ASHRAE Standard 90.1 (and the equivalent IECC commercial provisions), air-side or water-side economizers are required on cooling systems generally starting at a cooling capacity of about 5 tons (≈ 17.6 kW) for many climate zones and equipment types. Options A (1.5 t), C (10 t), and D (30 t) do not match the threshold used in the published commercial energy standard, so B is the correct choice.

QUESTION 5

SINGLE CHOICE

In a cooling load calculation using the CLTD/SHF method, the Solar Heat Gain Factor (SHGF) depends primarily on which parameter?

- ☐ A The U-value of the wall or roof assembly
- ☒ B The latitude, month, hour of the day and orientation of the surface
- ☐ C The indoor set-point dry-bulb temperature
- ☐ D The outdoor–indoor temperature difference for the design day

ANSWER B

WHY The SHGF (Solar Heat Gain Factor) tabulated in the ASHRAE cooling load references is the total solar radiation striking a surface for a given latitude, month, hour, and orientation (N, NE, E, SE, S, SW, W, NW, horizontal). It does not depend on the wall U-value, the indoor set-point, or the design outdoor–indoor temperature difference — those are accounted for separately in the calculation through the CLTD and the room air temperature. Therefore B is the correct answer.

QUESTION 6

SINGLE CHOICE

In a low-pressure steel duct system (≤ 2 in. w.g. static pressure class), what is the maximum recommended air velocity in the supply duct main directly upstream of the air-handling unit, per the standard equal-friction design guideline of 0.10 in. w.g. per 100 ft?

- ☐ A ≈ 700 fpm (3.6 m/s)
- ☐ B $\approx 1,000$ fpm (5.1 m/s)
- ☒ C $\approx 1,500$ fpm (7.6 m/s)
- ☐ D $\approx 2,500$ fpm (12.7 m/s)

ANSWER C

WHY Using the equal-friction method with a design friction rate of 0.10 in. w.g. per 100 ft (≈ 0.08 Pa/m) for round equivalents, typical main-duct velocities in commercial low-pressure ductwork fall in the 1,200–1,600 fpm range, with about 1,500 fpm (≈ 7.6 m/s) being a commonly used design velocity at the AHU discharge. Velocities of 700 fpm (too low, oversized duct), 1,000 fpm (typical for branch runs rather than mains), and 2,500 fpm (high-pressure ductwork, not low-pressure) are inconsistent with the stated low-pressure, 0.10 in. w.g./100 ft basis, so C is correct.

QUESTION 7

SINGLE CHOICE

During commissioning of a vapour-compression chiller, the refrigerant is observed boiling in the liquid line at the TXV (thermostatic expansion valve) inlet. The most likely cause is:

- ☐ A The TXV sensing bulb has lost its charge and is reading high, causing the valve to overfeed.
- ☐ B The condenser is oversized, lowering the condensing pressure below design.
- ☒ C The liquid line has an excessive pressure drop due to a clogged filter-drier or a line restriction.
- ☐ D The evaporator fan is operating at a higher speed than designed, increasing airflow across the coil.

ANSWER C

WHY Flashing (boiling) of refrigerant in the liquid line just upstream of the TXV indicates that the subcooled liquid has lost enough pressure to drop below its saturation temperature at that pressure — i.e., the pressure drop across the filter-drier, liquid-line solenoid, or piping is too high. This causes a loss of subcooling and a reduction in net refrigeration effect. Option A would cause a different symptom (starved evaporator, low superheat, possible flooding). Option B (oversized condenser) would actually raise, not lower, subcooling. Option D affects superheat but does not cause flashing in the liquid line. Therefore C is the correct answer.

QUESTION 8

SINGLE CHOICE

For a typical office space, ASHRAE Standard 62.1 prescribes a ventilation rate that combines outdoor air per unit area with outdoor air per person. The minimum outdoor-air intake flow required for a 1,000 ft² (≈ 93 m²) conference room with a default occupant density of 50 people/1,000 ft² is closest to:

- ☐ A 30 cfm (≈ 14 L/s)
- ☐ B 75 cfm (≈ 35 L/s)
- ☒ C 150 cfm (≈ 71 L/s)
- ☐ D 300 cfm (≈ 142 L/s)

ANSWER C

WHY Using the ASHRAE 62.1 IAQ procedure for an office-type space (default occupant density ≈ 50 people per 1,000 ft² $\rightarrow$ 50 people total), the per-person component is typically in the range of 5–7.5 cfm/person for a conference/break-room-like occupancy, and the per-area component is on the order of 0.06 cfm/ft². With about 50 people $\times$ 5 cfm/person $\approx$ 250 cfm, plus area credit, the calculated total exceeds 150 cfm but is well below 300 cfm after rounding to standard design values. Of the options given, 150 cfm (≈ 71 L/s) is the closest reasonable design figure for a small conference room; 30 cfm and 75 cfm are too low, while 300 cfm is excessive for the stated occupancy. Hence C is the best answer.

QUESTION 9

SINGLE CHOICE

Which statement about fire/smoke dampers in a commercial HVAC duct system is correct with respect to standard NFPA-based practice?

- ☐ **A** Fire dampers are required at every duct penetration of a fire-rated wall, regardless of duct size or rating.
- ☐ **B** Smoke dampers must be installed in every supply, return and exhaust duct, regardless of the presence of a smoke-control system.
- ☐ **C** Combination fire/smoke dampers are not permitted when a smoke-control system is provided.
- ☒ **D** Fire dampers are required at fire-rated wall penetrations only where the duct is not protected by a listed fire-rated enclosure.

ANSWER D

WHY Per NFPA 90A and NFPA 92, fire dampers are required at fire-rated wall/floor-ceiling penetrations by ducts — but NFPA 90A lists specific exceptions, including cases where the duct is enclosed in a fire-rated shaft or where the wall has a fire-resistance rating that does not require a damper. Option A is too absolute ('regardless of duct size or rating'). Option B is wrong — smoke dampers are required only where a smoke-control or smoke-management system calls for them. Option C is wrong — combination fire/smoke dampers are explicitly recognized and used. Therefore D is the correct answer.