Indoor Tennis Facility Comfort Calculations

Given Information

- Original comfortable thermostat setting: $68°F$ (with nearly still air, i.e. negligible air velocity).
- Mean radiant temperature (MRT) = air temperature initially.
- Desired same level of comfort after increasing air velocities.
- Scenario 1 (Problem 4-5): Increase average air velocity at player level from 0 to 100 fpm.
- Scenario 2 (Problem 4-6): Increase average air velocity at player level from 0 to 200 fpm.
- After determining the new comfort temperature for 200 fpm when MRT = air temperature, recalculate for the case where MRT is $9°F$ above the air temperature.

Background and Approach

Human thermal comfort depends on several factors: air temperature, mean radiant temperature, humidity, clothing, metabolic rate, and air velocity. Increasing the air velocity around occupants generally increases the convective heat loss from the skin, making them feel cooler. Thus, to maintain the *same level* of comfort when air speed increases, the thermostat can be set to a higher temperature.

A common rule-of-thumb in comfort engineering is: - Increasing indoor air speed by about 50 fpm can provide a cooling effect roughly equivalent to decreasing the temperature by $1°F$.

This means: - Increasing the air speed from 0 to 100 fpm can allow the thermostat setpoint to be raised by approximately $2°F$ for the same comfort. - Increasing the air speed from 0 to 200 fpm can allow the thermostat setpoint to be raised by approximately $4°F$ for the same comfort.

Note: These are approximate guidelines often used in practice. Actual comfort adjustments can vary based on exact conditions and personal preferences, but this rule-of-thumb is commonly used for quick estimates.

When the mean radiant temperature differs from the air temperature, the operative temperature is often approximated as the average of the air temperature (T_a) and the mean radiant temperature (T_r). Assuming equal weighting, the operative temperature T_o:

$$T_o \approx \frac{T_a + T_r}{2}.$$

If T_r is higher than T_a, the space will feel warmer than if $T_r = T_a$. In that case, to maintain the same comfort level (same operative temperature), one must lower the air temperature. The increased air velocity can partially compensate for this increase in radiant load by allowing a higher air temperature than would be possible without it.

Problem 4-5: Air Velocity from 0 to 100 fpm

Original Condition

- Original comfort is at $68°F$ with negligible air speed. - Mean radiant temperature = Air temperature = $68°F$. - Comfort operative temperature = $68°F$.

New Condition at 100 fpm

With an increase in air speed of 100 fpm: - For every 50 fpm increment, we gain about $1°F$ worth of "cooling" feel. - At 100 fpm, that is approximately $2°F$ of "cooling" effect.

To maintain the same comfort, we can thus increase the thermostat setting by approximately $2°F$.

$$T_{\text{new setpoint}} = 68°F + 2°F = 70°F.$$

Answer for Problem 4-5: The new thermostat setting is about $70°F$.

Problem 4-6: Air Velocity from 0 to 200 fpm

First, we compute the setpoint assuming the same MRT as the air temperature, then we consider the effect of a higher MRT.

Case 1: MRT = Air Temperature

At 200 fpm: - 200 fpm is about 4×50 fpm, which gives about $4°F$ equivalent cooling effect. - Starting from $68°F$ at still air, we can raise the thermostat by $4°F$ to achieve the same comfort.

$$T_{\text{new setpoint}} = 68°F + 4°F = 72°F.$$

Without any difference between MRT and air temperature, the new thermostat setting for 200 fpm would be about $72°F$.

Case 2: MRT = $T_a + 9^\circ F$

Now, let's assume the MRT is $9^\circ F$ above the air temperature. If the air temperature is T_a, then:

$$T_r = T_a + 9^\circ F.$$

The operative temperature T_o is approximately:

$$T_o = \frac{T_a + T_r}{2} = \frac{T_a + (T_a + 9)}{2} = \frac{2T_a + 9}{2} = T_a + 4.5^\circ F.$$

Originally, with $T_a = T_r = 68^\circ F$, the operative temperature was $68^\circ F$. We want the same comfort operative temperature as originally, which is $68^\circ F$.

Set the operative temperature equal to $68^\circ F$:

$$T_a + 4.5 = 68 \implies T_a = 68 - 4.5 = 63.5^\circ F.$$

So, if there was no air motion change and MRT is $9^\circ F$ above air temperature, you would have to lower the air temperature to $63.5^\circ F$ to maintain the same operative temperature (and thus the same comfort).

However, we now have an air speed of 200 fpm, which provides a $4^\circ F$ offset (cooling effect). This means we can add back about $4^\circ F$ to the air temperature while maintaining comfort.

$$T_{\text{new setpoint}} = 63.5^\circ F + 4^\circ F = 67.5^\circ F \approx 68^\circ F.$$

Answer for Problem 4-6: - With MRT = Air temperature: $T_{\text{setpoint}} \approx 72^\circ F$. - With MRT = $T_a + 9^\circ F$: $T_{\text{setpoint}} \approx 68^\circ F$.

Summary of Results

1. **Problem 4-5 (100 fpm, MRT = Air Temp):**

$$T_{\text{setpoint}} \approx 70^\circ F.$$

2. **Problem 4-6 (200 fpm, MRT = Air Temp):**

$$T_{\text{setpoint}} \approx 72^\circ F.$$

3. **Problem 4-6 (200 fpm, MRT = $T_a + 9^\circ F$):**

$$T_{\text{setpoint}} \approx 68^\circ F.$$

These results illustrate how increased air velocity can allow a higher thermostat setting while maintaining comfort, and how an increase in mean radiant temperature forces a lower air temperature to achieve the same comfort, partially compensated by increased air movement.

Problem 4-17

Given:

- CO_2 generation rate in the space: $G = 0.25$ cfm (of CO_2).
- Outdoor air supply rate: Q cfm (unknown).
- Outdoor CO_2 concentration: $C_{\text{out}} = 220$ ppm.
- Assume complete mixing and steady-state conditions.

Analysis:

At steady state, the total CO_2 leaving the space equals the total CO_2 entering (assuming no storage). The space is ventilated with outdoor air at rate Q cfm and CO_2 is being generated internally at G cfm of pure CO_2.

$$\text{Inflow of CO}_2 = Q \times C_{\text{out (fraction)}} + G$$

Let C = indoor CO_2 concentration in ppm. We must express everything consistently. One way is to convert ppm to a fraction by dividing by 1,000,000, since 1 ppm $= 10^{-6}$ fraction by volume.

- Outdoor CO_2 fraction:

$$C_{\text{out (fraction)}} = \frac{220}{1{,}000{,}000}.$$

- Indoor CO_2 fraction:

$$C_{\text{(fraction)}} = \frac{C}{1{,}000{,}000}.$$

At steady state, the outflow of CO_2 is:

$$\text{Outflow of CO}_2 = Q \times C_{\text{(fraction)}}.$$

Setting inflow equal to outflow:

$$Q \times \frac{220}{1{,}000{,}000} + G = Q \times \frac{C}{1{,}000{,}000}.$$

Multiply through by 1,000,000:

$$Q \times 220 + 1{,}000{,}000 \times G = Q \times C.$$

We know $G = 0.25$ cfm of pure CO_2. Pure CO_2 can be considered as 1,000,000 ppm. Thus:

$$1{,}000{,}000 \times 0.25 = 250{,}000.$$

So:

$$220Q + 250{,}000 = Q \times C.$$

Solve for C:

$$C = \frac{220Q + 250{,}000}{Q} = 220 + \frac{250{,}000}{Q}.$$

Result:

The steady-state indoor CO_2 concentration is:

$$C(\text{ppm}) = 220 + \frac{250{,}000}{Q}.$$

If the outdoor air flow rate Q is known, substitute and find C. Without Q, this is the general relationship.

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Problem 4-24

Given:

- Laboratory with computed total cooling load: 3 tons.
- Sensible heat factor (SHF) = 0.7.
- Indoor design: $T_{\text{in}} = 78°F$ db, $RH = 40\%$.
- Outdoor design: $T_{\text{out}} = 95°F$ db, $RH = 50\%$.
- Direct expansion (DX) equipment fixed airflow: 350 cfm/ton.

Analysis:

We need to check if using 100% outdoor air (OA) is feasible for this 3-ton system. With 3 tons:

$$\text{Total Airflow} = 3 \times 350 = 1050 \text{ cfm}.$$

If all 1050 cfm is from outdoors at $95°F$ and 50% RH, we must cool and dehumidify this air down to $78°F$, 40% RH.

Approximate Psychrometric Values:

- At $95°F$, 50% RH (Outdoor): - Humidity ratio $W_{\text{out}} \approx 0.0203\ \text{lb}_{\text{water}}/\text{lb}_{\text{dry air}}$. - Enthalpy $h_{\text{out}} \approx 41$ Btu/lb (approximate typical value).

- At $78°F$, 40% RH (Indoor): - Humidity ratio $W_{\text{in}} \approx 0.0120\ \text{lb}_{\text{water}}/\text{lb}_{\text{dry air}}$. - Enthalpy $h_{\text{in}} \approx 28$ Btu/lb (approximate typical value).

Calculate Cooling Load for 100% OA:

Air density approximation: 1 cfm $\approx 4.5\ \text{lb}_{\text{da}}/\text{hr}$. Thus:

$$\dot{m} = 1050 \text{ cfm} \times 4.5 = 4725\ \text{lb}_{\text{da}}/\text{hr}.$$

The required cooling to bring the outdoor air to the indoor condition:

$$\Delta h = h_{\text{out}} - h_{\text{in}} = 41 - 28 = 13 \text{ Btu/lb}.$$

Total cooling:

$$Q_{\text{cool}} = \dot{m} \times \Delta h = 4725 \times 13 = 61{,}425 \text{ Btu/hr}.$$

Convert to tons (1 ton = 12,000 Btu/hr):

$$\frac{61{,}425}{12{,}000} \approx 5.12 \text{ tons}.$$

But the available capacity is only 3 tons. This is significantly more than the system's capacity.

Conclusion:

Using 100% outdoor air under these design conditions is not feasible with only a 3-ton system. The load imposed by the hot, humid outdoor air far exceeds the cooling capacity available.

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Answers:

1. **Problem 4-17:**

$$C(\text{ppm}) = 220 + \frac{250{,}000}{Q} \quad \text{(where Q is in cfm)}$$

2. **Problem 4-24:** Using 100% outdoor air is not feasible. The load requires about 5.12 tons of cooling for 1050 cfm of outdoor air, which exceeds the available 3 tons.