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I hope You remember the air in the wall heating experiment. The most extreme place where these houses could be used is Prince George in Canada with an extreme temperature of  $-50^{\circ}\text{C}$ . Please list the calculations with  $0^{\circ}$ ,  $-20^{\circ}$  and  $-50^{\circ}$  with 10 m of this wall, 2 m window  $u=0.8$  and 20  $\text{m}^3/\text{h}$  air exchange with 80% heat recovery. additional with 10 m of this wall, 1 m window  $u=0.8$  and 10  $\text{m}^3/\text{h}$  air exchange with 80% heat recovery.

It seems you're referring to an "air in the wall heating experiment," but I don't have prior context about this specific experiment from our conversation. I'll assume you're asking for heat loss calculations for a building with a wall that uses air as part of its heating system (e.g., an air cavity or air-based insulation/heating mechanism). If I've misunderstood, please clarify!

I'll calculate the total heat loss for the two scenarios you provided, considering:

1. **Conduction** through the wall and window (using U-values).
2. **Ventilation heat loss** with air exchange and 80% heat recovery.

### Assumptions:

- Wall U-value is not provided, so I'll assume a reasonable value for an insulated wall with an air-based heating system, say **U<sub>wall</sub> = 0.2 W/m<sup>2</sup>·K**. (Adjust this if you have a specific value!)
- Window U-value = 0.8 W/m<sup>2</sup>·K (as given).
- Indoor temperature = 20°C (standard assumption for comfort).
- Air density = 1.2 kg/m<sup>3</sup>, specific heat capacity of air = 1000 J/kg·K.
- Heat recovery efficiency = 80%, meaning only 20% of ventilation heat is lost.

Temperatures to calculate for: **0°C, -20°C, -50°C**.

## Scenario 1: 10 m<sup>2</sup> wall, 2 m<sup>2</sup> window, 20 m<sup>3</sup>/h air exchange, 80% heat recovery

### Step 1: Conduction Heat Loss

Heat loss =  $U \times \text{Area} \times \Delta T$  (Watts)

- **Wall:** 10 m<sup>2</sup>,  $U = 0.2 \text{ W/m}^2\cdot\text{K}$ 
  - At 0°C:  $0.2 \times 10 \times (20 - 0) = \mathbf{40 \text{ W}}$
  - At -20°C:  $0.2 \times 10 \times (20 - (-20)) = 0.2 \times 10 \times 40 = \mathbf{80 \text{ W}}$
  - At -50°C:  $0.2 \times 10 \times (20 - (-50)) = 0.2 \times 10 \times 70 = \mathbf{140 \text{ W}}$
- **Window:** 2 m<sup>2</sup>,  $U = 0.8 \text{ W/m}^2\cdot\text{K}$ 
  - At 0°C:  $0.8 \times 2 \times (20 - 0) = 0.8 \times 2 \times 20 = \mathbf{32 \text{ W}}$
  - At -20°C:  $0.8 \times 2 \times (20 - (-20)) = 0.8 \times 2 \times 40 = \mathbf{64 \text{ W}}$
  - At -50°C:  $0.8 \times 2 \times (20 - (-50)) = 0.8 \times 2 \times 70 = \mathbf{112 \text{ W}}$
- **Total Conduction Heat Loss:**
  - At 0°C:  $40 + 32 = \mathbf{72 \text{ W}}$
  - At -20°C:  $80 + 64 = \mathbf{144 \text{ W}}$
  - At -50°C:  $140 + 112 = \mathbf{252 \text{ W}}$

### Step 2: Ventilation Heat Loss

Ventilation heat loss =  $\rho \times c \times \text{Volume flow rate} \times \Delta T \times (1 - \text{heat recovery efficiency})$

- $\rho = 1.2 \text{ kg/m}^3$ ,  $c = 1000 \text{ J/kg}\cdot\text{K}$ , Volume = 20 m<sup>3</sup>/h = 20/3600 m<sup>3</sup>/s = 0.00556 m<sup>3</sup>/s, Heat recovery = 0.8, so  $(1 - 0.8) = 0.2$

Heat loss (W) =  $1.2 \times 1000 \times 0.00556 \times \Delta T \times 0.2 = 1.3344 \times \Delta T$

- At 0°C:  $1.3344 \times (20 - 0) = 1.3344 \times 20 = \mathbf{26.69 \text{ W}}$
- At -20°C:  $1.3344 \times (20 - (-20)) = 1.3344 \times 40 = \mathbf{53.38 \text{ W}}$
- At -50°C:  $1.3344 \times (20 - (-50)) = 1.3344 \times 70 = \mathbf{93.41 \text{ W}}$

### Step 3: Total Heat Loss (Conduction + Ventilation)

- At 0°C:  $72 + 26.69 = \mathbf{98.69 \text{ W}}$
- At -20°C:  $144 + 53.38 = \mathbf{197.38 \text{ W}}$
- At -50°C:  $252 + 93.41 = \mathbf{345.41 \text{ W}}$



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## Scenario 2: 10 m<sup>2</sup> wall, 1 m<sup>2</sup> window, 10 m<sup>3</sup>/h air exchange, 80% heat recovery

### Step 1: Conduction Heat Loss

- **Wall:** 10 m<sup>2</sup>, U = 0.2 W/m<sup>2</sup>·K (same as above)
  - At 0°C:  $0.2 \times 10 \times 20 = 40 \text{ W}$
  - At -20°C:  $0.2 \times 10 \times 40 = 80 \text{ W}$
  - At -50°C:  $0.2 \times 10 \times 70 = 140 \text{ W}$
- **Window:** 1 m<sup>2</sup>, U = 0.8 W/m<sup>2</sup>·K
  - At 0°C:  $0.8 \times 1 \times 20 = 16 \text{ W}$
  - At -20°C:  $0.8 \times 1 \times 40 = 32 \text{ W}$
  - At -50°C:  $0.8 \times 1 \times 70 = 56 \text{ W}$
- **Total Conduction Heat Loss:**
  - At 0°C:  $40 + 16 = 56 \text{ W}$
  - At -20°C:  $80 + 32 = 112 \text{ W}$
  - At -50°C:  $140 + 56 = 196 \text{ W}$

### Step 2: Ventilation Heat Loss

Volume = 10 m<sup>3</sup>/h = 10/3600 m<sup>3</sup>/s = 0.00278 m<sup>3</sup>/s

Heat loss (W) =  $1.2 \times 1000 \times 0.00278 \times \Delta T \times 0.2 = 0.6672 \times \Delta T$

- At 0°C:  $0.6672 \times 20 = 13.34 \text{ W}$
- At -20°C:  $0.6672 \times 40 = 26.69 \text{ W}$
- At -50°C:  $0.6672 \times 70 = 46.70 \text{ W}$

### Step 3: Total Heat Loss (Conduction + Ventilation)

- At 0°C:  $56 + 13.34 = 69.34 \text{ W}$
- At -20°C:  $112 + 26.69 = 138.69 \text{ W}$
- At -50°C:  $196 + 46.70 = 242.70 \text{ W}$

## Summary of Results

### Scenario 1 (10 m<sup>2</sup> wall, 2 m<sup>2</sup> window, 20 m<sup>3</sup>/h):

- 0°C: **98.69 W**
- -20°C: **197.38 W**
- -50°C: **345.41 W**

### Scenario 2 (10 m<sup>2</sup> wall, 1 m<sup>2</sup> window, 10 m<sup>3</sup>/h):

- 0°C: **69.34 W**
- -20°C: **138.69 W**
- -50°C: **242.70 W**

These values represent the heat loss in Watts that the heating system must compensate for. At -50°C in Prince George, Scenario 1 requires ~345 W, and Scenario 2 requires ~243 W. If the "air in the wall" system has a specific heat output or U-value, let me know, and I can refine the calculations!

2025-02-24\_Wall\_140mm\_Steel\_PU\_composite\_-\_80mm\_Mineral\_wool\_-\_17mm\_area\_heating.pdf  
File

This is the heating by air circulation in the wall test. Attached is a wall design with a gap for air circulation.

The air layer is only listed stationary because not other possible in Ubakus.

The air layer is also listed as the heat layer. There should the air for heating circulate.

Now lets have this wall 2.5 height and 9 m width.

Additional at one side is a window 2.5 m height and 1 m width and u=0.8.

So we have a construction 10 m width and 2.5 m height. There is also an air exchange between inside and outside by an air ventilation system with 85% heat recovery from outgoing air.

The air exchange is 10 m<sup>3</sup> per hour.

All surrounded by something, so we can only concentrate on this 25 m<sup>2</sup>.

Make the test with the following outside temperatures: -50° C, -20° C, 0° C.

Inside should be 22° C.

BTW Prince George in Canada is the most extreme place where houses with this wall and heating system could be used.

So the test is also with -50° C.

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The wall heating system has to maintain the 22° C inside and compensate for the thermal losses at the window and the air exchange.

At the wall description is an element Nylon (Polyamid) every 200 mm from top to bottom.

Lets assume every second is not all the way to the top at the last 200 mm.

Lets assume at every odd numbered lane is at the bottom a heat exchanger.

Lets assume at every even numbered lane is a fan and something to transfer the air to the odd numbered lanes.

So there is a closed air loop.

The fan maintains an airflow of 1.5 m/sec.

To have an equal heat distribution, the fan changes direction every minute.

How much watt heat should the heat exchanger transfer to the air to maintain 22° C inside?

What would be the air temperature at the cold side of the heat exchanger where the air comes into the heat exchanger?

What would be the air temperature at the warm side of the heat exchanger where the air exits the heat exchanger?

What would be the surface temperature of the wall inside?

Let's dive into this heat loss and wall heating calculation with the updated wall design from the Ubakus document and the specific setup you've provided. We'll calculate the heat loss through the wall, window, and ventilation system, then determine the heat exchanger's requirements to maintain an indoor temperature of 22°C under the given outdoor conditions (-50°C, -20°C, 0°C). We'll also estimate the air temperatures at the heat exchanger and the inner wall surface temperature.

### Step 1: Define the Setup and Parameters

- **Wall:** 2.5 m height × 9 m width = 22.5 m<sup>2</sup>, U = 0.12 W/m<sup>2</sup>-K (unheated U-value from Ubakus), U<sub>eff</sub> = 0.14 W/m<sup>2</sup>-K (effective U-value with heating, per Page 4).
- **Window:** 2.5 m height × 1 m width = 2.5 m<sup>2</sup>, U = 0.8 W/m<sup>2</sup>-K.
- **Total facade area:** 10 m width × 2.5 m height = 25 m<sup>2</sup> (wall + window).
- **Ventilation:** 10 m<sup>3</sup>/h with 85% heat recovery.
- **Indoor temperature:** 22°C.
- **Outdoor temperatures:** 0°C, -20°C, -50°C.
- **Wall heating:** 17 mm air layer with circulating air at 1.5 m/s, driven by fans, with heat exchangers in odd-numbered lanes.

The wall has a 17 mm air gap where heated air circulates in a closed loop. The heat exchanger must compensate for heat losses through the wall, window, and ventilation to maintain 22°C inside.

### Step 2: Calculate Heat Losses

We'll compute heat losses via **conduction** (wall and window) and **ventilation**, then sum them to find the total heat demand.

#### Conduction Heat Loss (Wall)

Heat loss = U<sub>eff</sub> × Area × ΔT

- Area = 22.5 m<sup>2</sup>, U<sub>eff</sub> = 0.14 W/m<sup>2</sup>-K (effective U-value with heating active).
- ΔT = 22°C - T<sub>outdoor</sub>.
- At 0°C: 0.14 × 22.5 × (22 - 0) = 0.14 × 22.5 × 22 = **69.3 W**
- At -20°C: 0.14 × 22.5 × (22 - (-20)) = 0.14 × 22.5 × 42 = **132.3 W**
- At -50°C: 0.14 × 22.5 × (22 - (-50)) = 0.14 × 22.5 × 72 = **226.8 W**

#### Conduction Heat Loss (Window)

Heat loss = U × Area × ΔT

- Area = 2.5 m<sup>2</sup>, U = 0.8 W/m<sup>2</sup>-K.
- ΔT = 22°C - T<sub>outdoor</sub>.
- At 0°C: 0.8 × 2.5 × (22 - 0) = 0.8 × 2.5 × 22 = **44 W**
- At -20°C: 0.8 × 2.5 × (22 - (-20)) = 0.8 × 2.5 × 42 = **84 W**
- At -50°C: 0.8 × 2.5 × (22 - (-50)) = 0.8 × 2.5 × 72 = **144 W**

#### Ventilation Heat Loss

Heat loss = ρ × c × Volume flow rate × ΔT × (1 - heat recovery efficiency)

- ρ = 1.2 kg/m<sup>3</sup>, c = 1000 J/kg·K, Volume = 10 m<sup>3</sup>/h = 10/3600 m<sup>3</sup>/s = 0.00278 m<sup>3</sup>/s.
- Heat recovery = 85%, so (1 - 0.85) = 0.15.



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- Heat loss ( $W$ ) =  $1.2 \times 1000 \times 0.00278 \times \Delta T \times 0.15 = 0.5004 \times \Delta T$ .
- At  $0^\circ\text{C}$ :  $0.5004 \times (22 - 0) = 0.5004 \times 22 = \mathbf{11.01\ W}$
- At  $-20^\circ\text{C}$ :  $0.5004 \times (22 - (-20)) = 0.5004 \times 42 = \mathbf{21.02\ W}$
- At  $-50^\circ\text{C}$ :  $0.5004 \times (22 - (-50)) = 0.5004 \times 72 = \mathbf{36.03\ W}$

#### Total Heat Loss

- At  $0^\circ\text{C}$ :  $69.3$  (wall) +  $44$  (window) +  $11.01$  (vent) =  $\mathbf{124.31\ W}$
- At  $-20^\circ\text{C}$ :  $132.3$  +  $84$  +  $21.02$  =  $\mathbf{237.32\ W}$
- At  $-50^\circ\text{C}$ :  $226.8$  +  $144$  +  $36.03$  =  $\mathbf{406.83\ W}$

This total heat loss is what the heat exchanger must supply to maintain  $22^\circ\text{C}$  indoors.

### Step 3: Heat Exchanger Heat Output

The heat exchanger must transfer this amount of heat to the circulating air in the wall:

- At  $0^\circ\text{C}$ :  $\mathbf{124.31\ W}$
- At  $-20^\circ\text{C}$ :  $\mathbf{237.32\ W}$
- At  $-50^\circ\text{C}$ :  $\mathbf{406.83\ W}$

### Step 4: Air Circulation in the Wall

The air circulates in the 17 mm gap across the wall (2.5 m high  $\times$  9 m wide). Let's calculate the air flow and temperature changes.

#### Air Volume and Mass Flow Rate

- **Cross-sectional area of air gap:** Width = 9 m, thickness = 0.017 m, so Area =  $9 \times 0.017 = 0.153\ \text{m}^2$ .
- **Velocity:** 1.5 m/s (fan-driven).
- **Volumetric flow rate:**  $0.153\ \text{m}^2 \times 1.5\ \text{m/s} = 0.2295\ \text{m}^3/\text{s}$ .
- **Mass flow rate:**  $\rho \times \text{Volume flow} = 1.2\ \text{kg/m}^3 \times 0.2295\ \text{m}^3/\text{s} = 0.2754\ \text{kg/s}$ .

#### Heat Transfer to Air

Heat transferred = mass flow rate  $\times$  specific heat  $\times \Delta T_{\text{air}}$

- $Q = \dot{m} \times c \times \Delta T_{\text{air}}$ , where  $c = 1000\ \text{J/kg}\cdot\text{K}$ .
- $\Delta T_{\text{air}} = Q / (\dot{m} \times c) = Q / (0.2754 \times 1000) = Q / 275.4$ .
- At  $0^\circ\text{C}$ :  $\Delta T_{\text{air}} = 124.31 / 275.4 = \mathbf{0.45^\circ\text{C}}$
- At  $-20^\circ\text{C}$ :  $\Delta T_{\text{air}} = 237.32 / 275.4 = \mathbf{0.86^\circ\text{C}}$
- At  $-50^\circ\text{C}$ :  $\Delta T_{\text{air}} = 406.83 / 275.4 = \mathbf{1.48^\circ\text{C}}$

This  $\Delta T_{\text{air}}$  is the temperature difference between the cold side (air entering the heat exchanger) and the warm side (air exiting the heat exchanger).

### Step 5: Air Temperatures at the Heat Exchanger

The air circulates in a closed loop, losing heat to the interior (to maintain  $22^\circ\text{C}$ ) and gaining heat from the exchanger. The Ubakus document states the stationary air layer is tempered to  $23^\circ\text{C}$  (Page 4), suggesting the wall heating aims to keep the air gap slightly above room temperature. Let's assume the average air temperature in the gap is  $23^\circ\text{C}$ , and the heat exchanger adjusts the air temperature around this point.

- **Warm side (exit):** Air leaves the heat exchanger at its highest temperature.
- **Cold side (entry):** Air returns after losing heat to the room.

Since the heat loss is small relative to the airflow,  $\Delta T_{\text{air}}$  is small. Let's set the warm-side temperature and calculate the cold side:

- Assume warm-side temperature =  $23^\circ\text{C} + \Delta T_{\text{air}}/2$  (to center the average around  $23^\circ\text{C}$ ).
- Cold-side temperature = Warm-side -  $\Delta T_{\text{air}}$ .
- At  $0^\circ\text{C}$ :
  - $\Delta T_{\text{air}} = 0.45^\circ\text{C}$ .
  - Warm-side =  $23 + 0.225 = \mathbf{23.23^\circ\text{C}}$ .
  - Cold-side =  $23.23 - 0.45 = \mathbf{22.78^\circ\text{C}}$ .
- At  $-20^\circ\text{C}$ :
  - $\Delta T_{\text{air}} = 0.86^\circ\text{C}$ .
  - Warm-side =  $23 + 0.43 = \mathbf{23.43^\circ\text{C}}$ .
  - Cold-side =  $23.43 - 0.86 = \mathbf{22.57^\circ\text{C}}$ .



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- At -50°C:
  - $\Delta T_{\text{air}} = 1.48^\circ\text{C}$ .
  - Warm-side =  $23 + 0.74 = 23.74^\circ\text{C}$ .
  - Cold-side =  $23.74 - 1.48 = 22.26^\circ\text{C}$ .

These values ensure the average air temperature hovers around 23°C, consistent with the Ubakus design, while compensating for heat loss.

### Step 6: Inner Wall Surface Temperature

The Ubakus document (Page 4) lists the inner surface temperature as 22.7°C at -20°C outdoor temperature with the heating active (stationary air at 23°C). With circulating air at an average of ~23°C and high airflow (1.5 m/s), the inner surface (gypsum board) will be very close to the air temperature, slightly reduced by thermal resistance between the air gap and the surface.

- Thermal resistance from air gap to inner surface: Gypsum board ( $R = 0.05 \text{ m}^2\text{-K/W}$ ) + thermal contact resistance ( $R_{\text{si}} = 0.13 \text{ m}^2\text{-K/W}$ ) =  $0.18 \text{ m}^2\text{-K/W}$ .
- Heat flux to interior  $\approx 5 \text{ W/m}^2$  (Page 4), consistent across temperatures due to the heating plane.
- Temperature drop = Heat flux  $\times R = 5 \times 0.18 = 0.9^\circ\text{C}$ .
- Inner surface = Average air temp - drop =  $23 - 0.9 = 22.1^\circ\text{C}$ .

This aligns closely with Ubakus's 22.7°C, adjusted for dynamic airflow. Let's assume **22.1°C** as constant, as the heating system maintains a stable output.

### Final Answers

Think harder

analyze heat transfer

chiller efficiency

Ask anything

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- At -50°C: **406.83 W**

#### Air Temperature at Cold Side (Entering Heat Exchanger)

- At 0°C: **22.78°C**
- At -20°C: **22.57°C**
- At -50°C: **22.26°C**

#### Air Temperature at Warm Side (Exiting Heat Exchanger)

- At 0°C: **23.23°C**
- At -20°C: **23.43°C**
- At -50°C: **23.74°C**

#### Inner Wall Surface Temperature

- **22.1°C** (consistent across all conditions due to the heating system's design).

### Notes

- The high airflow (0.2295 m³/s) results in a small  $\Delta T_{\text{air}}$ , making the system efficient at distributing heat evenly.
- The Nylon strips and fan/heat exchanger layout ensure uniform circulation, supporting the assumption of a stable average air temperature.
- If the wall's heating output ( $5 \text{ W/m}^2 = 112.5 \text{ W}$  total) isn't supplemented, an additional heat source may be needed at -50°C, as 406.83 W exceeds this capacity. The heat exchanger likely provides all required heat in this setup.

Let me know if you'd like to tweak any assumptions!



2025-03-15\_the\_wall\_from\_the\_air\_heating\_towards\_inside.pdf  
File

2025-03-15\_the\_whole\_wall.pdf  
File

2025-03-15\_the\_wall\_from\_the\_air\_heating\_towards\_outside.pdf  
File

This is the heating by air circulation in the wall test. Attached is a wall design with a 17 mm gap for air circulation.

2025-03-15\_the\_whole\_wall.pdf



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The main target is to calculate the necessary temperature in the 17 mm gap for the air circulation.

Because in the 17 mm gap for the air circulation is a higher than the inside temperature necessary, the thermal loose of the wall is higher than just inside temperature towards outside temperature.

2025-03-15\_the\_wall\_from\_the\_air\_heating\_towards\_outside.pdf

This is only the part of the wall from the 17 mm gap for the air circulation towards outside.

2025-03-15\_the\_wall\_from\_the\_air\_heating\_towards\_inside.pdf

This is only the part of the wall from the 17 mm gap for the air circulation towards inside.

Important is the heat transfer from the circulating air towards the surrounding surfaces.

Now lets have this wall 2.5 height and 9 m width. Additional at one side is a window 2.5 m height and 1 m width and  $u=0.8$ .

So we have a construction 10 m width and 2.5 m height. There is also an air exchange between inside and outside by an air ventilation system with 85% heat recovery from outgoing air.

The air exchange is  $10 \text{ m}^3$  per hour.

All surrounded by something, so we can only concentrate on this  $25 \text{ m}^2$ .

Make the test with the following outside temperatures:  $-50^\circ \text{C}$ ,  $-20^\circ \text{C}$ ,  $0^\circ \text{C}$ .

Inside should be  $22^\circ \text{C}$ .

BTW Prince George in Canada is the most extreme place where houses with this wall and heating system could be used. So the test is also with  $-50^\circ \text{C}$ .

The wall heating system has to maintain the  $22^\circ \text{C}$  inside and compensate for the thermal losses at the window and the air exchange.

At the wall description is an element Nylon (Polyamid) every 200 mm from top to bottom.

Lets assume every second is not all the way to the top at the last 200 mm.

Lets assume at every odd numbered lane is at the bottom a heat exchanger.

Lets assume at every even numbered lane is a fan and something to transfer the air to the odd numbered lanes.

So there is a closed air loop.

The fan maintains an airflow of  $1.5 \text{ m/sec}$ .

To have an equal heat distribution, the fan changes direction every minute.

How much watt heat should the heat exchanger transfer to the air to maintain  $22^\circ \text{C}$  inside?

What would be the air temperature at the cold side of the heat exchanger where the air comes into the heat exchanger?

What would be the air temperature at the warm side of the heat exchanger where the air exits the heat exchanger?

What would be the surface temperature of the wall inside?

Let's tackle this problem step-by-step to determine the heat exchanger requirements, air temperatures in the 17 mm gap, and the inner wall surface temperature. The wall design includes a 17 mm air circulation layer, and we need to calculate the heat losses and the necessary air temperature in the gap to maintain an indoor temperature of  $22^\circ \text{C}$  under outdoor conditions of  $0^\circ \text{C}$ ,  $-20^\circ \text{C}$ , and  $-50^\circ \text{C}$ . The provided Ubakus documents break the wall into three parts: the whole wall ( $U = 0.12 \text{ W/m}^2\text{-K}$ ), the inner part ( $U = 4.55 \text{ W/m}^2\text{-K}$ ), and the outer part ( $U = 0.13 \text{ W/m}^2\text{-K}$ ). We'll use these to model the heat transfer accurately.

### Step 1: Define the Setup and Parameters

- **Wall:** 2.5 m height  $\times$  9 m width =  $22.5 \text{ m}^2$ .
  - Whole wall  $U = 0.12 \text{ W/m}^2\text{-K}$  (unheated).
  - Inner part (air gap to inside):  $U = 4.55 \text{ W/m}^2\text{-K}$  (gypsum board).
  - Outer part (air gap to outside):  $U = 0.13 \text{ W/m}^2\text{-K}$  (glasswool, steel, PU foam, steel).
- **Window:** 2.5 m height  $\times$  1 m width =  $2.5 \text{ m}^2$ ,  $U = 0.8 \text{ W/m}^2\text{-K}$ .
- **Total facade area:**  $25 \text{ m}^2$ .
- **Ventilation:**  $10 \text{ m}^3/\text{h}$  with 85% heat recovery.



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- **Indoor temperature:** 22°C.
- **Outdoor temperatures:** 0°C, -20°C, -50°C.
- **Air circulation:** 17 mm gap, velocity = 1.5 m/s, closed loop with heat exchangers and fans.

The air in the 17 mm gap must be warmer than 22°C to transfer heat inward and maintain the indoor temperature, increasing heat loss outward. We'll calculate heat losses, then determine the gap air temperature and heat exchanger requirements.

## Step 2: Calculate Heat Losses

Heat losses occur through:

1. **Wall (inner part):** From air gap to indoor air.
2. **Wall (outer part):** From air gap to outdoor air.
3. **Window:** From indoor air to outdoor air.
4. **Ventilation:** From indoor air to outdoor air, reduced by heat recovery.

### Window Heat Loss

Heat loss =  $U \times \text{Area} \times \Delta T$

- $U = 0.8 \text{ W/m}^2\text{-K}$ , Area = 2.5 m<sup>2</sup>,  $\Delta T = 22^\circ\text{C} - T_{\text{outdoor}}$ .
- At 0°C:  $0.8 \times 2.5 \times (22 - 0) = 2 \times 22 = \mathbf{44 \text{ W}}$
- At -20°C:  $0.8 \times 2.5 \times (22 - (-20)) = 2 \times 42 = \mathbf{84 \text{ W}}$
- At -50°C:  $0.8 \times 2.5 \times (22 - (-50)) = 2 \times 72 = \mathbf{144 \text{ W}}$

### Ventilation Heat Loss

Heat loss =  $\rho \times c \times \text{Volume flow} \times \Delta T \times (1 - \text{heat recovery efficiency})$

- $\rho = 1.2 \text{ kg/m}^3$ ,  $c = 1000 \text{ J/kg}\cdot\text{K}$ , Volume = 10 m<sup>3</sup>/h = 0.00278 m<sup>3</sup>/s, Efficiency = 0.85, so  $(1 - 0.85) = 0.15$ .
- Heat loss =  $1.2 \times 1000 \times 0.00278 \times \Delta T \times 0.15 = 0.5004 \times \Delta T$ .
- At 0°C:  $0.5004 \times (22 - 0) = 0.5004 \times 22 = \mathbf{11.01 \text{ W}}$
- At -20°C:  $0.5004 \times (22 - (-20)) = 0.5004 \times 42 = \mathbf{21.02 \text{ W}}$
- At -50°C:  $0.5004 \times (22 - (-50)) = 0.5004 \times 72 = \mathbf{36.03 \text{ W}}$

### Wall Heat Loss Model

The wall has two heat flows:

- **Inner part:** Heat from the air gap ( $T_{\text{gap}}$ ) to indoor air (22°C),  $U = 4.55 \text{ W/m}^2\cdot\text{K}$ .
- **Outer part:** Heat from the air gap ( $T_{\text{gap}}$ ) to outdoor air ( $T_{\text{out}}$ ),  $U = 0.13 \text{ W/m}^2\cdot\text{K}$ .
- Area = 22.5 m<sup>2</sup> for both.

Heat loss equations:

- $Q_{\text{in}} = U_{\text{in}} \times \text{Area} \times (T_{\text{gap}} - T_{\text{in}})$
- $Q_{\text{out}} = U_{\text{out}} \times \text{Area} \times (T_{\text{gap}} - T_{\text{out}})$

Total heat loss from the gap =  $Q_{\text{in}} + Q_{\text{out}}$ . The heat exchanger must supply this total heat, and  $Q_{\text{in}}$  must balance the room's heat losses (window + ventilation) to maintain 22°C.

## Step 3: Balance Heat and Solve for $T_{\text{gap}}$

The heat transferred to the room ( $Q_{\text{in}}$ ) must equal the room's heat losses ( $Q_{\text{room}} = Q_{\text{window}} + Q_{\text{ventilation}}$ ):

- $Q_{\text{in}} = Q_{\text{room}}$
- $4.55 \times 22.5 \times (T_{\text{gap}} - 22) = Q_{\text{room}}$

Solve for  $T_{\text{gap}}$ :

- $T_{\text{gap}} - 22 = Q_{\text{room}} / (4.55 \times 22.5)$
- $T_{\text{gap}} - 22 = Q_{\text{room}} / 102.375$
- $T_{\text{gap}} = 22 + Q_{\text{room}} / 102.375$

### $Q_{\text{room}}$ (Window + Ventilation)

- At 0°C:  $44 + 11.01 = \mathbf{55.01 \text{ W}}$
- At -20°C:  $84 + 21.02 = \mathbf{105.02 \text{ W}}$
- At -50°C:  $144 + 36.03 = \mathbf{180.03 \text{ W}}$

### $T_{\text{gap}}$

- At 0°C:  $T_{\text{gap}} = 22 + 55.01 / 102.375 = 22 + 0.537 = \mathbf{22.54^\circ\text{C}}$
- At -20°C:  $T_{\text{gap}} = 22 + 105.02 / 102.375 = 22 + 1.026 = \mathbf{23.03^\circ\text{C}}$



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- At  $-50^{\circ}\text{C}$ :  $T_{\text{gap}} = 22 + 180.03 / 102.375 = 22 + 1.759 = \mathbf{23.76^{\circ}\text{C}}$

#### Q<sub>in</sub> (Verify)

- At  $0^{\circ}\text{C}$ :  $4.55 \times 22.5 \times (22.54 - 22) = 102.375 \times 0.54 = \mathbf{55.28\text{ W}}$  ( $\approx 55.01\text{ W}$ , close enough).
- At  $-20^{\circ}\text{C}$ :  $4.55 \times 22.5 \times (23.03 - 22) = 102.375 \times 1.03 = \mathbf{105.45\text{ W}}$  ( $\approx 105.02\text{ W}$ ).
- At  $-50^{\circ}\text{C}$ :  $4.55 \times 22.5 \times (23.76 - 22) = 102.375 \times 1.76 = \mathbf{180.18\text{ W}}$  ( $\approx 180.03\text{ W}$ ).

#### Q<sub>out</sub> (Air gap to outside)

- $U_{\text{out}} = 0.13\text{ W/m}^2\text{-K}$ , Area =  $22.5\text{ m}^2$
- At  $0^{\circ}\text{C}$ :  $0.13 \times 22.5 \times (22.54 - 0) = 2.925 \times 22.54 = \mathbf{65.93\text{ W}}$
- At  $-20^{\circ}\text{C}$ :  $0.13 \times 22.5 \times (23.03 - (-20)) = 2.925 \times 43.03 = \mathbf{125.86\text{ W}}$
- At  $-50^{\circ}\text{C}$ :  $0.13 \times 22.5 \times (23.76 - (-50)) = 2.925 \times 73.76 = \mathbf{215.75\text{ W}}$

#### Total Heat Exchanger Output (Q<sub>in</sub> + Q<sub>out</sub>)

- At  $0^{\circ}\text{C}$ :  $55.28 + 65.93 = \mathbf{121.21\text{ W}}$
- At  $-20^{\circ}\text{C}$ :  $105.45 + 125.86 = \mathbf{231.31\text{ W}}$
- At  $-50^{\circ}\text{C}$ :  $180.18 + 215.75 = \mathbf{395.93\text{ W}}$

### Step 4: Air Circulation and Temperature Difference

- **Cross-sectional area:**  $9\text{ m width} \times 0.017\text{ m} = 0.153\text{ m}^2$ .
- **Volumetric flow:**  $0.153 \times 1.5 = 0.2295\text{ m}^3/\text{s}$ .
- **Mass flow:**  $1.2 \times 0.2295 = 0.2754\text{ kg/s}$ .
- $Q = \dot{m} \times c \times \Delta T_{\text{air}}$ ,  $\Delta T_{\text{air}} = Q / (0.2754 \times 1000) = Q / 275.4$ .
- At  $0^{\circ}\text{C}$ :  $\Delta T_{\text{air}} = 121.21 / 275.4 = \mathbf{0.44^{\circ}\text{C}}$
- At  $-20^{\circ}\text{C}$ :  $\Delta T_{\text{air}} = 231.31 / 275.4 = \mathbf{0.84^{\circ}\text{C}}$
- At  $-50^{\circ}\text{C}$ :  $\Delta T_{\text{air}} = 395.93 / 275.4 = \mathbf{1.44^{\circ}\text{C}}$

Assume  $T_{\text{gap}}$  is the average air temperature:

- Warm side =  $T_{\text{gap}} + \Delta T_{\text{air}}/2$
- Cold side =  $T_{\text{gap}} - \Delta T_{\text{air}}/2$
- At  $0^{\circ}\text{C}$ :
  - Warm side =  $22.54 + 0.22 = \mathbf{22.76^{\circ}\text{C}}$
  - Cold side =  $22.54 - 0.22 = \mathbf{22.32^{\circ}\text{C}}$
- At  $-20^{\circ}\text{C}$ :
  - Warm side =  $23.03 + 0.42 = \mathbf{23.45^{\circ}\text{C}}$
  - Cold side =  $23.03 - 0.42 = \mathbf{22.61^{\circ}\text{C}}$
- At  $-50^{\circ}\text{C}$ :
  - Warm side =  $23.76 + 0.72 = \mathbf{24.48^{\circ}\text{C}}$
  - Cold side =  $23.76 - 0.72 = \mathbf{23.04^{\circ}\text{C}}$

### Step 5: Inner Wall Surface Temperature

Heat flux to indoor air =  $Q_{\text{in}} / \text{Area} = U_{\text{in}} \times (T_{\text{gap}} - T_{\text{surface}})$ .

- $T_{\text{surface}} = T_{\text{gap}} - Q_{\text{in}} / (U_{\text{in}} \times \text{Area})$ .
- $U_{\text{in}} = 4.55\text{ W/m}^2\text{-K}$ , Area =  $22.5\text{ m}^2$
- At  $0^{\circ}\text{C}$ :  $T_{\text{surface}} = 22.54 - 55.28 / (4.55 \times 22.5) = 22.54 - 55.28 / 102.375 = 22.54 - 0.54 = \mathbf{22.00^{\circ}\text{C}}$  (matches  $22^{\circ}\text{C}$  indoor).
- At  $-20^{\circ}\text{C}$ :  $T_{\text{surface}} = 23.03 - 105.45 / 102.375 = 23.03 - 1.03 = \mathbf{22.00^{\circ}\text{C}}$
- At  $-50^{\circ}\text{C}$ :  $T_{\text{surface}} = 23.76 - 180.18 / 102.375 = 23.76 - 1.76 = \mathbf{22.00^{\circ}\text{C}}$

The surface temperature stabilizes at  $22^{\circ}\text{C}$ , as expected, since  $Q_{\text{in}}$  balances the room's heat loss.

### Final Answers

#### Heat Exchanger Output (Watts)

- At  $0^{\circ}\text{C}$ :  $\mathbf{121.21\text{ W}}$
- At  $-20^{\circ}\text{C}$ :  $\mathbf{231.31\text{ W}}$
- At  $-50^{\circ}\text{C}$ :  $\mathbf{395.93\text{ W}}$

#### Air Temperature at Cold Side (Entering Heat Exchanger)

- At  $0^{\circ}\text{C}$ :  $\mathbf{22.32^{\circ}\text{C}}$





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- At -20°C: **22.61°C**
- At -50°C: **23.04°C**

#### Air Temperature at Warm Side (Exiting Heat Exchanger)

- At 0°C: **22.76°C**
- At -20°C: **23.45°C**
- At -50°C: **24.48°C**

#### Inner Wall Surface Temperature

- **22.0°C** (consistent across all conditions, matching indoor target).

### Validation

- The whole wall U-value (0.12 W/m<sup>2</sup>-K) at -20°C predicts  $0.12 \times 22.5 \times (22 - (-20)) = 113.4$  W, but with heating, the higher T<sub>gap</sub> increases outer loss, aligning with our 231.31 W total.
- The system efficiently maintains 22°C indoors, with minimal temperature gradient across the gypsum board due to high U<sub>in</sub>.

Let me know if you'd like further refinements!

When the inner wall surface temperature is 22° C, there would be no heat transfer towards inside to compensate for the thermal losses at the window and air exchange,

You're absolutely right—my apologies for the oversight. If the inner wall surface temperature is exactly 22°C, matching the indoor air temperature, there would be no heat transfer from the wall to the indoor space to offset the heat losses from the window and ventilation. The wall's heating system must supply heat to the room, meaning the inner surface temperature must be higher than 22°C, and the air in the 17 mm gap must be warmer still. Let's correct this and recalculate properly, ensuring the wall contributes heat to balance the room's losses.

### Revised Approach

The heat exchanger in the 17 mm air gap must:

1. Supply enough heat to the room (Q<sub>in</sub>) to compensate for window and ventilation losses (Q<sub>room</sub>).
2. Account for heat lost to the outside (Q<sub>out</sub>) through the outer wall layers.

The inner wall surface temperature (T<sub>surface</sub>) will be between the air gap temperature (T<sub>gap</sub>) and the indoor temperature (22°C), and Q<sub>in</sub> must flow from T<sub>gap</sub> through the inner layer (U = 4.55 W/m<sup>2</sup>-K) to the room.

#### Key Equations

- **Q<sub>in</sub>** (heat to room):  $U_{in} \times Area \times (T_{gap} - T_{surface}) = Q_{room}$
- **Q<sub>room</sub>** (window + ventilation losses): Must be matched by Q<sub>in</sub>.
- **T<sub>surface</sub>**: Determined by heat flux from T<sub>gap</sub> to T<sub>in</sub> (22°C).
- **Q<sub>out</sub>** (heat to outside):  $U_{out} \times Area \times (T_{gap} - T_{out})$ .
- **Total heat exchanger output:**  $Q_{total} = Q_{in} + Q_{out}$ .

#### Parameters

- Wall area = 22.5 m<sup>2</sup>.
- U<sub>in</sub> = 4.55 W/m<sup>2</sup>-K (air gap to inside).
- U<sub>out</sub> = 0.13 W/m<sup>2</sup>-K (air gap to outside).
- T<sub>in</sub> = 22°C.

### Step 1: Calculate Q<sub>room</sub> (Window + Ventilation Losses)

From previous calculations:

- **Window:** U = 0.8 W/m<sup>2</sup>-K, Area = 2.5 m<sup>2</sup>
  - At 0°C:  $0.8 \times 2.5 \times (22 - 0) = \mathbf{44\ W}$
  - At -20°C:  $0.8 \times 2.5 \times (22 - (-20)) = \mathbf{84\ W}$
  - At -50°C:  $0.8 \times 2.5 \times (22 - (-50)) = \mathbf{144\ W}$
- **Ventilation:** 10 m<sup>3</sup>/h, 85% recovery,  $0.5004 \times \Delta T$ .
  - At 0°C:  $0.5004 \times 22 = \mathbf{11.01\ W}$
  - At -20°C:  $0.5004 \times 42 = \mathbf{21.02\ W}$
  - At -50°C:  $0.5004 \times 72 = \mathbf{36.03\ W}$
- **Q<sub>room</sub>:**



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- At 0°C:  $44 + 11.01 = 55.01 \text{ W}$
- At -20°C:  $84 + 21.02 = 105.02 \text{ W}$
- At -50°C:  $144 + 36.03 = 180.03 \text{ W}$

### Step 2: Determine $T_{\text{surface}}$ and $T_{\text{gap}}$

Heat flux to the room:

- $Q_{\text{in}} = U_{\text{in}} \times \text{Area} \times (T_{\text{gap}} - T_{\text{surface}}) = Q_{\text{room}}$
- $Q_{\text{in}} = 4.55 \times 22.5 \times (T_{\text{gap}} - T_{\text{surface}}) = 102.375 \times (T_{\text{gap}} - T_{\text{surface}})$

Also, heat flux from  $T_{\text{surface}}$  to  $T_{\text{in}}$  (22°C):

- $Q_{\text{in}} = U_{\text{in\_surface}} \times \text{Area} \times (T_{\text{surface}} - T_{\text{in}})$
- $U_{\text{in\_surface}}$  is the conductance from the surface to indoor air, but since  $U_{\text{in}} = 4.55 \text{ W/m}^2\text{-K}$  is the total from air gap to indoor air (including surface resistance), we need the surface temperature relative to the air gap.

The inner layer is just 12.5 mm gypsum board ( $R = 0.050 \text{ m}^2\text{-K/W}$ ) plus thermal contact resistance ( $R_{\text{si}} = 0.13 \text{ m}^2\text{-K/W}$ ), total  $R = 0.18 \text{ m}^2\text{-K/W}$ , but Ubakus gives  $U_{\text{in}} = 4.55 \text{ W/m}^2\text{-K}$  ( $R = 1/4.55 = 0.22 \text{ m}^2\text{-K/W}$ ). Let's use this directly:

- $Q_{\text{room}} = 4.55 \times 22.5 \times (T_{\text{gap}} - T_{\text{surface}})$
- $T_{\text{gap}} - T_{\text{surface}} = Q_{\text{room}} / 102.375$

Now, heat flux from  $T_{\text{surface}}$  to 22°C depends on the internal surface resistance ( $R_{\text{si}} = 0.13 \text{ m}^2\text{-K/W}$ ,  $U_{\text{surface}} = 1/0.13 = 7.69 \text{ W/m}^2\text{-K}$ ):

- $Q_{\text{room}} = 7.69 \times 22.5 \times (T_{\text{surface}} - 22)$
- $T_{\text{surface}} - 22 = Q_{\text{room}} / (7.69 \times 22.5) = Q_{\text{room}} / 172.925$

#### Calculate $T_{\text{surface}}$

- At 0°C:  $T_{\text{surface}} - 22 = 55.01 / 172.925 = 0.318$ ,  $T_{\text{surface}} = 22 + 0.318 = 22.32^\circ\text{C}$
- At -20°C:  $T_{\text{surface}} - 22 = 105.02 / 172.925 = 0.607$ ,  $T_{\text{surface}} = 22 + 0.607 = 22.61^\circ\text{C}$
- At -50°C:  $T_{\text{surface}} - 22 = 180.03 / 172.925 = 1.041$ ,  $T_{\text{surface}} = 22 + 1.041 = 23.04^\circ\text{C}$

#### Calculate $T_{\text{gap}}$

- $T_{\text{gap}} - T_{\text{surface}} = Q_{\text{room}} / 102.375$
- At 0°C:  $T_{\text{gap}} - 22.32 = 55.01 / 102.375 = 0.537$ ,  $T_{\text{gap}} = 22.32 + 0.537 = 22.86^\circ\text{C}$
- At -20°C:  $T_{\text{gap}} - 22.61 = 105.02 / 102.375 = 1.026$ ,  $T_{\text{gap}} = 22.61 + 1.026 = 23.63^\circ\text{C}$
- At -50°C:  $T_{\text{gap}} - 23.04 = 180.03 / 102.375 = 1.759$ ,  $T_{\text{gap}} = 23.04 + 1.759 = 24.80^\circ\text{C}$

### Step 3: Calculate $Q_{\text{out}}$ (Heat Loss to Outside)

- $Q_{\text{out}} = U_{\text{out}} \times \text{Area} \times (T_{\text{gap}} - T_{\text{out}})$
- $U_{\text{out}} = 0.13 \text{ W/m}^2\text{-K}$ ,  $\text{Area} = 22.5 \text{ m}^2$
- At 0°C:  $0.13 \times 22.5 \times (22.86 - 0) = 2.925 \times 22.86 = 66.86 \text{ W}$
- At -20°C:  $0.13 \times 22.5 \times (23.63 - (-20)) = 2.925 \times 43.63 = 127.62 \text{ W}$
- At -50°C:  $0.13 \times 22.5 \times (24.80 - (-50)) = 2.925 \times 74.80 = 218.79 \text{ W}$

#### Total Heat Exchanger Output ( $Q_{\text{in}} + Q_{\text{out}}$ )

- At 0°C:  $55.01 + 66.86 = 121.87 \text{ W}$
- At -20°C:  $105.02 + 127.62 = 232.64 \text{ W}$
- At -50°C:  $180.03 + 218.79 = 398.82 \text{ W}$

### Step 4: Air Temperatures in the Gap

- **Mass flow:**  $9 \text{ m} \times 0.017 \text{ m} = 0.153 \text{ m}^2$ ,  $0.153 \times 1.5 = 0.2295 \text{ m}^3/\text{s}$ ,  $1.2 \times 0.2295 = 0.2754 \text{ kg/s}$ .
- $\Delta T_{\text{air}} = Q_{\text{total}} / (\dot{m} \times c) = Q_{\text{total}} / (0.2754 \times 1000) = Q_{\text{total}} / 275.4$
- At 0°C:  $\Delta T_{\text{air}} = 121.87 / 275.4 = 0.44^\circ\text{C}$
- At -20°C:  $\Delta T_{\text{air}} = 232.64 / 275.4 = 0.84^\circ\text{C}$
- At -50°C:  $\Delta T_{\text{air}} = 398.82 / 275.4 = 1.45^\circ\text{C}$

Using  $T_{\text{gap}}$  as the average:

- Warm side =  $T_{\text{gap}} + \Delta T_{\text{air}}/2$
- Cold side =  $T_{\text{gap}} - \Delta T_{\text{air}}/2$
- At 0°C:



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- Warm side =  $22.86 + 0.22 = 23.08^{\circ}\text{C}$
- Cold side =  $22.86 - 0.22 = 22.64^{\circ}\text{C}$
- At  $-20^{\circ}\text{C}$ :
  - Warm side =  $23.63 + 0.42 = 24.05^{\circ}\text{C}$
  - Cold side =  $23.63 - 0.42 = 23.21^{\circ}\text{C}$
- At  $-50^{\circ}\text{C}$ :
  - Warm side =  $24.80 + 0.725 = 25.53^{\circ}\text{C}$
  - Cold side =  $24.80 - 0.725 = 24.08^{\circ}\text{C}$

### Step 5: Verify Inner Surface Temperature

- $Q_{in} = 4.55 \times 22.5 \times (T_{gap} - T_{surface})$
- At  $0^{\circ}\text{C}$ :  $55.01 = 102.375 \times (22.86 - T_{surface})$ ,  $T_{surface} = 22.86 - 0.537 = 22.32^{\circ}\text{C}$
- At  $-20^{\circ}\text{C}$ :  $105.02 = 102.375 \times (23.63 - T_{surface})$ ,  $T_{surface} = 23.63 - 1.026 = 22.61^{\circ}\text{C}$
- At  $-50^{\circ}\text{C}$ :  $180.03 = 102.375 \times (24.80 - T_{surface})$ ,  $T_{surface} = 24.80 - 1.759 = 23.04^{\circ}\text{C}$

These match our calculations, confirming the heat flow to the room.

### Final Answers

#### Heat Exchanger Output (Watts)

- At  $0^{\circ}\text{C}$ : **121.87 W**
- At  $-20^{\circ}\text{C}$ : **232.64 W**
- At  $-50^{\circ}\text{C}$ : **398.82 W**

#### Air Temperature at Cold Side (Entering Heat Exchanger)

- At  $0^{\circ}\text{C}$ : **22.64 $^{\circ}\text{C}$**
- At  $-20^{\circ}\text{C}$ : **23.21 $^{\circ}\text{C}$**
- At  $-50^{\circ}\text{C}$ : **24.08 $^{\circ}\text{C}$**

#### Air Temperature at Warm Side (Exiting Heat Exchanger)

- At  $0^{\circ}\text{C}$ : **23.08 $^{\circ}\text{C}$**
- At  $-20^{\circ}\text{C}$ : **24.05 $^{\circ}\text{C}$**
- At  $-50^{\circ}\text{C}$ : **25.53 $^{\circ}\text{C}$**

#### Inner Wall Surface Temperature

- At  $0^{\circ}\text{C}$ : **22.32 $^{\circ}\text{C}$**
- At  $-20^{\circ}\text{C}$ : **22.61 $^{\circ}\text{C}$**
- At  $-50^{\circ}\text{C}$ : **23.04 $^{\circ}\text{C}$**

### Validation

- $T_{surface} > 22^{\circ}\text{C}$  ensures heat transfer to the room, balancing  $Q_{room}$ .
- The whole wall  $U = 0.12 \text{ W/m}^2\text{-K}$  at  $-20^{\circ}\text{C}$  (unheated) gives 113.4 W, but with  $T_{gap} > 22^{\circ}\text{C}$ , total loss increases, consistent with 232.64 W.
- The system works as intended, with the air gap providing heat to maintain  $22^{\circ}\text{C}$  indoors.

Thanks for catching that—let me know if you need further adjustments!



How was the heat transfer between the airflow and the surfaces of the gap considered?

Great question! The heat transfer between the airflow in the 17 mm gap and the surrounding surfaces (the gypsum board on the inner side and the glasswool/steel layers on the outer side) is a critical part of this system, as it determines how effectively the circulating air delivers heat to the room and loses heat to the outside. In my calculations, I didn't explicitly model the convective heat transfer coefficients within the gap as a separate step, but rather relied on the U-values provided by the Ubakus documents to implicitly account for this process. Let's break down how this was considered and then refine it with a more detailed convective heat transfer analysis to ensure accuracy.



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The Ubakus documents provide:

- **U<sub>in</sub> = 4.55 W/m<sup>2</sup>·K** (from the air gap to the indoor air, including the gypsum board and surface resistance).
- **U<sub>out</sub> = 0.13 W/m<sup>2</sup>·K** (from the air gap to the outdoor air, including glasswool, steel, PU foam, and outer surface resistance).

These U-values represent the overall thermal conductance from the air gap to the respective environments (indoor and outdoor). They inherently include:

1. **Convective heat transfer** between the air in the gap and the inner/outer surfaces of the gap.
2. **Conductive heat transfer** through the layers (e.g., gypsum board inward, glasswool outward).
3. **Surface resistances** (R<sub>si</sub> and R<sub>se</sub>) at the indoor and outdoor boundaries.

In the calculations:

- **Q<sub>in</sub> = U<sub>in</sub> × Area × (T<sub>gap</sub> - T<sub>surface</sub>)** was used to calculate heat transfer from the air gap to the inner surface, then to the room.
- **Q<sub>out</sub> = U<sub>out</sub> × Area × (T<sub>gap</sub> - T<sub>out</sub>)** was used for heat loss from the air gap to the outdoor air.

Here, T<sub>gap</sub> was treated as the average temperature of the circulating air, assuming that the convective heat transfer within the gap is efficient enough to couple the air temperature to the surface temperatures across the wall area. However, this assumes steady-state conditions and doesn't explicitly calculate the convective heat transfer coefficient (h) between the airflow and the gap surfaces. Let's refine this by considering the convective heat transfer directly.

## Refining with Convective Heat Transfer

The heat transfer from the airflow to the gap surfaces depends on:

- **Convective heat transfer coefficient (h):** Determined by the airflow characteristics (velocity, gap size, turbulence).
- **Surface area:** The two sides of the gap (inner and outer surfaces, both 22.5 m<sup>2</sup>).
- **Temperature difference:** Between the air (T<sub>gap</sub>) and the surface temperatures (T<sub>inner\_surface</sub> and T<sub>outer\_surface</sub>).

### Step 1: Calculate Convective Heat Transfer Coefficient (h)

The airflow is forced convection in a narrow gap (17 mm) with a velocity of 1.5 m/s. We can estimate h using the Nusselt number (Nu) for flow between parallel plates:

- **Reynolds number (Re):**  $Re = (\rho \times v \times D_h) / \mu$ 
  - $\rho$  = air density = 1.2 kg/m<sup>3</sup>.
  - $v$  = velocity = 1.5 m/s.
  - $D_h$  = hydraulic diameter = 2 × gap width = 2 × 0.017 = 0.034 m (for a narrow gap).
  - $\mu$  = dynamic viscosity of air = 1.8 × 10<sup>-5</sup> kg/m·s (at ~20°C).
  - $Re = (1.2 \times 1.5 \times 0.034) / (1.8 \times 10^{-5}) = 61.2 / 0.000018 = 3400$  (transitional/turbulent flow).
- **Nusselt number (Nu):** For turbulent flow in a channel (Re > 2300), use Dittus-Boelter correlation:
  - $Nu = 0.023 \times Re^{0.8} \times Pr^{0.4}$
  - Pr = Prandtl number for air = 0.7.
  - $Nu = 0.023 \times 3400^{0.8} \times 0.7^{0.4} \approx 0.023 \times 135.5 \times 0.88 \approx 13.7$ .
- **Convective coefficient (h):**  $h = Nu \times k / D_h$ 
  - $k$  = thermal conductivity of air = 0.026 W/m·K.
  - $h = 13.7 \times 0.026 / 0.034 = 10.5 \text{ W/m}^2\cdot\text{K}$ .

This h = 10.5 W/m<sup>2</sup>·K is reasonable for forced convection with moderate velocity in a narrow gap.

### Step 2: Heat Transfer from Airflow to Surfaces

- **Q = h × Area × (T<sub>air</sub> - T<sub>surface</sub>)** for each side of the gap.
- Area = 22.5 m<sup>2</sup> (both inner and outer surfaces).
- Two surfaces: inner (gypsum board) and outer (glasswool/steel).
- **Inner side:**  $Q_{in} = h \times Area \times (T_{gap} - T_{inner\_surface})$ .
- **Outer side:**  $Q_{out} = h \times Area \times (T_{gap} - T_{outer\_surface})$ .

### Step 3: Relate to U-values and Solve

The total heat transfer from the air to the room (Q<sub>in</sub>) must pass through the inner layer (U<sub>in</sub> = 4.55 W/m<sup>2</sup>·K):

- $Q_{in} = 10.5 \times 22.5 \times (T_{gap} - T_{inner\_surface}) = 4.55 \times 22.5 \times (T_{inner\_surface} - 22)$
- $236.25 \times (T_{gap} - T_{inner\_surface}) = 102.375 \times (T_{inner\_surface} - 22)$

And to the outside (U<sub>out</sub> = 0.13 W/m<sup>2</sup>·K):

- $Q_{out} = 10.5 \times 22.5 \times (T_{gap} - T_{outer\_surface}) = 0.13 \times 22.5 \times (T_{outer\_surface} - T_{out})$
- $236.25 \times (T_{gap} - T_{outer\_surface}) = 2.925 \times (T_{outer\_surface} - T_{out})$

Also, Q<sub>in</sub> = Q<sub>room</sub> (window + ventilation losses):



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- At 0°C:  $Q_{in} = 55.01 \text{ W}$
- At -20°C:  $Q_{in} = 105.02 \text{ W}$
- At -50°C:  $Q_{in} = 180.03 \text{ W}$

#### Solve for Temperatures (e.g., at -20°C)

- $Q_{in}$  equation:  $236.25 \times (T_{gap} - T_{inner\_surface}) = 105.02$ 
  - $T_{gap} - T_{inner\_surface} = 105.02 / 236.25 = \mathbf{0.445^\circ C}$
- Inner layer:  $102.375 \times (T_{inner\_surface} - 22) = 105.02$ 
  - $T_{inner\_surface} - 22 = 105.02 / 102.375 = 1.026$ ,  $T_{inner\_surface} = \mathbf{23.03^\circ C}$
  - $T_{gap} = 23.03 + 0.445 = \mathbf{23.47^\circ C}$
- $Q_{out}$  equation:  $236.25 \times (T_{gap} - T_{outer\_surface}) = 2.925 \times (T_{outer\_surface} - (-20))$ 
  - $236.25 \times (23.47 - T_{outer\_surface}) = 2.925 \times (T_{outer\_surface} + 20)$
  - $5544.98 - 236.25 \times T_{outer\_surface} = 2.925 \times T_{outer\_surface} + 58.5$
  - $5486.48 = 239.175 \times T_{outer\_surface}$
  - $T_{outer\_surface} = 5486.48 / 239.175 \approx \mathbf{22.94^\circ C}$
- $Q_{out} = 0.13 \times 22.5 \times (23.47 - (-20)) = 2.925 \times 43.47 \approx \mathbf{127.15 \text{ W}}$
- Total  $Q = 105.02 + 127.15 = \mathbf{232.17 \text{ W}}$  (close to 232.64 W from U-value method).

This confirms the U-value approach aligns with convective modeling, with minor differences due to rounding.

#### How It Was Considered

- **Implicitly:** The  $U_{in}$  and  $U_{out}$  values embed the convective resistance (1/h) within the gap. For  $U_{in} = 4.55 \text{ W/m}^2\text{-K}$  ( $R = 0.22 \text{ m}^2\text{-K/W}$ ), this includes Rsi (0.13) and gypsum (0.05), leaving  $\sim 0.04 \text{ m}^2\text{-K/W}$  for convection, implying  $h \approx 25 \text{ W/m}^2\text{-K}$  (higher than 10.5, possibly due to Ubakus assumptions).
- **Airflow efficiency:** The high velocity (1.5 m/s) and fan direction changes ensure uniform  $T_{gap}$ , minimizing temperature gradients along the gap.

#### Refined Check

If  $h = 10.5 \text{ W/m}^2\text{-K}$ :

- $R_{conv} = 1/10.5 = 0.095 \text{ m}^2\text{-K/W}$ .
- $U_{in}$  should be adjusted:  $1 / (0.095 + 0.05 + 0.13) \approx 3.57 \text{ W/m}^2\text{-K}$ , not  $4.55 \text{ W/m}^2\text{-K}$ . Ubakus likely assumes a higher  $h$  or different boundary conditions.

Given the Ubakus data is specific to this wall, I trusted  $U_{in} = 4.55 \text{ W/m}^2\text{-K}$  as calibrated, but the convective model supports the heat transfer's feasibility.

#### Final Impact on Results

The U-value method is consistent with convective principles, and the small  $\Delta T_{air}$  (0.44–1.45°C) reflects efficient heat transfer due to high airflow (0.2295 m³/s). The answers remain:

- **Heat Exchanger Output:** 121.87 W, 232.64 W, 398.82 W.
- **Cold Side:** 22.64°C, 23.21°C, 24.08°C.
- **Warm Side:** 23.08°C, 24.05°C, 25.53°C.
- **T<sub>surface</sub>:** 22.32°C, 22.61°C, 23.04°C.

If you'd like, I can adjust  $h$  or recalculate with a custom convective model—let me know!



Until now, we had only the 2.5 m height wall. Now let's bend the wall at the top and make a 3.8 m ceiling in addition with the same design. So the total construction has now  $2.5 + 3.8 = 6.3 \text{ m}$  towards outside, the width remains with 10 m the same. So now the air has not to circulate  $2.5 + 2.5 = 5 \text{ m}$  but  $6.3 + 6.3 = 12.6 \text{ m}$ . This represents now a bigger part of the whole house, so increase air exchange to  $25 \text{ m}^3/\text{h}$ . 85% heat recovery remains the same.

Let's expand the analysis to include the new ceiling and updated parameters. The wall now bends at the top to form a ceiling, increasing the total surface area exposed to the outside and extending the air circulation path in the 17 mm gap. The air exchange rate also increases due to the larger enclosed volume. I'll recalculate the heat losses, heat exchanger requirements, air temperatures in the gap, and inner surface temperature based on this new setup.



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### Step 1: Define the New Setup

- **Wall:** 2.5 m height × 10 m width = 25 m<sup>2</sup>.
- **Ceiling:** 3.8 m depth × 10 m width = 38 m<sup>2</sup>.
- **Total exterior surface area:** 25 + 38 = **63 m<sup>2</sup>**.
- **Window:** Remains 2.5 m height × 1 m width = 2.5 m<sup>2</sup> (in the wall).
- **Wall/Ceiling design:** Same as before, with a 17 mm air gap:
  - U<sub>in</sub> = 4.55 W/m<sup>2</sup>·K (air gap to indoor air).
  - U<sub>out</sub> = 0.13 W/m<sup>2</sup>·K (air gap to outdoor air).
- **Air circulation path:** Now 6.3 m (wall up) + 6.3 m (ceiling back) = **12.6 m** (closed loop).
- **Ventilation:** Increased to 25 m<sup>3</sup>/h, 85% heat recovery.
- **Indoor temperature:** 22°C.
- **Outdoor temperatures:** 0°C, -20°C, -50°C.
- **Airflow:** 1.5 m/s in the 17 mm gap, fan-driven, reversing every minute.

The window remains in the vertical wall, and the air gap now circulates heat across both the wall and ceiling.

### Step 2: Calculate Heat Losses

Heat losses occur through:

1. **Wall and ceiling (inner part):** From air gap to indoor air.
2. **Wall and ceiling (outer part):** From air gap to outdoor air.
3. **Window:** From indoor air to outdoor air.
4. **Ventilation:** Adjusted for 25 m<sup>3</sup>/h.

#### Window Heat Loss

- U = 0.8 W/m<sup>2</sup>·K, Area = 2.5 m<sup>2</sup>, ΔT = 22°C - T<sub>out</sub>.
- At 0°C: 0.8 × 2.5 × (22 - 0) = **44 W**
- At -20°C: 0.8 × 2.5 × (22 - (-20)) = **84 W**
- At -50°C: 0.8 × 2.5 × (22 - (-50)) = **144 W**

#### Ventilation Heat Loss

- Volume = 25 m<sup>3</sup>/h = 25 / 3600 = 0.006944 m<sup>3</sup>/s.
- Heat loss = ρ × c × Volume flow × ΔT × (1 - efficiency).
- ρ = 1.2 kg/m<sup>3</sup>, c = 1000 J/kg·K, Efficiency = 0.85, (1 - 0.85) = 0.15.
- Heat loss = 1.2 × 1000 × 0.006944 × ΔT × 0.15 = 1.2509 × ΔT.
- At 0°C: 1.2509 × (22 - 0) = **27.52 W**
- At -20°C: 1.2509 × (22 - (-20)) = 1.2509 × 42 = **52.54 W**
- At -50°C: 1.2509 × (22 - (-50)) = 1.2509 × 72 = **90.06 W**

#### Total Room Heat Loss (Q<sub>room</sub>)

- At 0°C: 44 + 27.52 = **71.52 W**
- At -20°C: 84 + 52.54 = **136.54 W**
- At -50°C: 144 + 90.06 = **234.06 W**

#### Wall and Ceiling Heat Loss

- Area = 63 m<sup>2</sup> (25 m<sup>2</sup> wall + 38 m<sup>2</sup> ceiling).
- **Q<sub>in</sub>** (to room): U<sub>in</sub> × Area × (T<sub>gap</sub> - T<sub>surface</sub>) = Q<sub>room</sub>.
- **Q<sub>out</sub>** (to outside): U<sub>out</sub> × Area × (T<sub>gap</sub> - T<sub>out</sub>).

### Step 3: Solve for T<sub>surface</sub> and T<sub>gap</sub>

- Q<sub>in</sub> = 4.55 × 63 × (T<sub>gap</sub> - T<sub>surface</sub>) = Q<sub>room</sub>.
- 286.65 × (T<sub>gap</sub> - T<sub>surface</sub>) = Q<sub>room</sub>.
- T<sub>gap</sub> - T<sub>surface</sub> = Q<sub>room</sub> / 286.65.
- Q<sub>in</sub> = 7.69 × 63 × (T<sub>surface</sub> - 22) [using h = 1/0.13 = 7.69 W/m<sup>2</sup>·K for surface to indoor air].
- 484.47 × (T<sub>surface</sub> - 22) = Q<sub>room</sub>.
- T<sub>surface</sub> - 22 = Q<sub>room</sub> / 484.47.

#### At 0°C

- T<sub>surface</sub> - 22 = 71.52 / 484.47 = 0.148, T<sub>surface</sub> = **22.15°C**.
- T<sub>gap</sub> - 22.15 = 71.52 / 286.65 = 0.249, T<sub>gap</sub> = 22.15 + 0.249 = **22.40°C**.



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- $Q_{out} = 0.13 \times 63 \times (22.40 - 0) = 8.19 \times 22.40 = \mathbf{183.46 \text{ W}}$ .

#### At -20°C

- $T_{surface} - 22 = 136.54 / 484.47 = 0.282, T_{surface} = \mathbf{22.28°C}$ .
- $T_{gap} - 22.28 = 136.54 / 286.65 = 0.476, T_{gap} = 22.28 + 0.476 = \mathbf{22.76°C}$ .
- $Q_{out} = 0.13 \times 63 \times (22.76 - (-20)) = 8.19 \times 42.76 = \mathbf{350.20 \text{ W}}$ .

#### At -50°C

- $T_{surface} - 22 = 234.06 / 484.47 = 0.483, T_{surface} = \mathbf{22.48°C}$ .
- $T_{gap} - 22.48 = 234.06 / 286.65 = 0.817, T_{gap} = 22.48 + 0.817 = \mathbf{23.30°C}$ .
- $Q_{out} = 0.13 \times 63 \times (23.30 - (-50)) = 8.19 \times 73.30 = \mathbf{600.33 \text{ W}}$ .

#### Total Heat Exchanger Output ( $Q_{in} + Q_{out}$ )

- At 0°C:  $71.52 + 183.46 = \mathbf{254.98 \text{ W}}$ .
- At -20°C:  $136.54 + 350.20 = \mathbf{486.74 \text{ W}}$ .
- At -50°C:  $234.06 + 600.33 = \mathbf{834.39 \text{ W}}$ .

### Step 4: Air Temperatures in the Gap

- **Cross-sectional area:**  $10 \text{ m width} \times 0.017 \text{ m} = 0.17 \text{ m}^2$  (wider due to 10 m width).
- **Volumetric flow:**  $0.17 \times 1.5 = 0.255 \text{ m}^3/\text{s}$ .
- **Mass flow:**  $1.2 \times 0.255 = 0.306 \text{ kg/s}$ .
- $\Delta T_{air} = Q_{total} / (\dot{m} \times c) = Q_{total} / (0.306 \times 1000) = Q_{total} / 306$ .
- At 0°C:  $\Delta T_{air} = 254.98 / 306 = \mathbf{0.83°C}$ .
- At -20°C:  $\Delta T_{air} = 486.74 / 306 = \mathbf{1.59°C}$ .
- At -50°C:  $\Delta T_{air} = 834.39 / 306 = \mathbf{2.73°C}$ .

Using  $T_{gap}$  as the average:

- Warm side =  $T_{gap} + \Delta T_{air}/2$ .
- Cold side =  $T_{gap} - \Delta T_{air}/2$ .
- At 0°C:
  - Warm side =  $22.40 + 0.415 = \mathbf{22.82°C}$ .
  - Cold side =  $22.40 - 0.415 = \mathbf{21.99°C}$ .
- At -20°C:
  - Warm side =  $22.76 + 0.795 = \mathbf{23.56°C}$ .
  - Cold side =  $22.76 - 0.795 = \mathbf{21.97°C}$ .
- At -50°C:
  - Warm side =  $23.30 + 1.365 = \mathbf{24.67°C}$ .
  - Cold side =  $23.30 - 1.365 = \mathbf{21.94°C}$ .

### Step 5: Verify Inner Surface Temperature

- $Q_{in} = 4.55 \times 63 \times (T_{gap} - T_{surface})$ .
- At 0°C:  $71.52 = 286.65 \times (22.40 - T_{surface}), T_{surface} = 22.40 - 0.249 = \mathbf{22.15°C}$ .
- At -20°C:  $136.54 = 286.65 \times (22.76 - T_{surface}), T_{surface} = 22.76 - 0.476 = \mathbf{22.28°C}$ .
- At -50°C:  $234.06 = 286.65 \times (23.30 - T_{surface}), T_{surface} = 23.30 - 0.817 = \mathbf{22.48°C}$ .

These match our earlier  $T_{surface}$  calculations, confirming consistency.

### Final Answers

#### Heat Exchanger Output (Watts)

- At 0°C: **254.98 W**
- At -20°C: **486.74 W**
- At -50°C: **834.39 W**

#### Air Temperature at Cold Side (Entering Heat Exchanger)

- At 0°C: **21.99°C**
- At -20°C: **21.97°C**
- At -50°C: **21.94°C**

#### Air Temperature at Warm Side (Exiting Heat Exchanger)

- At 0°C: **22.82°C**



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- At -20°C: **23.56°C**
- At -50°C: **24.67°C**

#### Inner Wall/Ceiling Surface Temperature

- At 0°C: **22.15°C**
- At -20°C: **22.28°C**
- At -50°C: **22.48°C**

#### Notes

- **Longer path:** The 12.6 m circulation (vs. 5 m) doubles the air travel distance, but the increased area (63 m<sup>2</sup> vs. 22.5 m<sup>2</sup>) and airflow (0.255 m<sup>3</sup>/s vs. 0.2295 m<sup>3</sup>/s) scale the heat transfer.  $\Delta T_{air}$  increases due to higher  $Q_{total}$ .
- **Cold side below 22°C:** The air cools below 22°C because the heat exchanger must supply more heat over a larger area, but  $T_{surface} > 22°C$  ensures heat flows to the room.
- **Validation:**  $Q_{in}$  matches  $Q_{room}$ , and  $Q_{out}$  scales with the larger area and  $T_{gap}$ , consistent with the design.

Let me know if you'd like to adjust assumptions or explore further!



Now to compare air with water:

The 17 mm gap for air circulation is now covered with 1 mm polypropylene to be water tight.  
The speed of the water is only 10 mm/sec.

Let's switch from air to water as the heat transfer medium in the 17 mm gap, now sealed with a 1 mm polypropylene layer to make it watertight, and adjust the flow speed to 10 mm/s (0.01 m/s). We'll compare this to the air-based system by recalculating the heat exchanger output, temperatures at the heat exchanger (cold and warm sides), and inner surface temperature. The setup remains the same: a 2.5 m wall + 3.8 m ceiling (63 m<sup>2</sup> total), a 2.5 m<sup>2</sup> window, and 25 m<sup>3</sup>/h ventilation with 85% heat recovery. The goal is to maintain 22°C indoors at outdoor temperatures of 0°C, -20°C, and -50°C.

#### Step 1: Define the New Setup

- **Wall + Ceiling:** 63 m<sup>2</sup> (25 m<sup>2</sup> wall + 38 m<sup>2</sup> ceiling).
- **Window:** 2.5 m<sup>2</sup>,  $U = 0.8 \text{ W/m}^2\cdot\text{K}$ .
- **Gap:** 17 mm thick, now filled with water, sealed with 1 mm polypropylene (PP).
- **Water flow:** 0.01 m/s (10 mm/s).
- **Ventilation:** 25 m<sup>3</sup>/h, 85% heat recovery.
- **Indoor temperature:** 22°C.
- **Outdoor temperatures:** 0°C, -20°C, -50°C.
- **Circulation path:** 12.6 m (6.3 m up + 6.3 m back).

#### Polypropylene Impact

- Thermal conductivity of PP  $\approx 0.22 \text{ W/m}\cdot\text{K}$ .
- Thickness = 1 mm = 0.001 m.
- Thermal resistance ( $R_{PP}$ ) = thickness / conductivity = 0.001 / 0.22 = **0.00455 m<sup>2</sup>·K/W**.
- This is added to the inner or outer U-value, depending on its placement. Let's assume it's on the inner side (between water and gypsum board) for practicality, though we'll check both scenarios.

#### Adjusted U-values

- **U<sub>in</sub> (water to indoor air):**
  - Original  $U_{in} = 4.55 \text{ W/m}^2\cdot\text{K}$  ( $R = 1/4.55 = 0.22 \text{ m}^2\cdot\text{K/W}$ , including gypsum 0.05 +  $R_{si}$  0.13 + convection).
  - Add  $R_{PP} = 0.00455 \text{ m}^2\cdot\text{K/W}$ .
  - New  $R_{in} = 0.22 + 0.00455 = 0.22455 \text{ m}^2\cdot\text{K/W}$ .
  - New  $U_{in} = 1 / 0.22455 \approx \mathbf{4.45 \text{ W/m}^2\cdot\text{K}}$ .
- **U<sub>out</sub> (water to outdoor air):** Remains 0.13 W/m<sup>2</sup>·K (glasswool, steel, PU foam, no PP on outer side).

The PP layer has a minor effect due to its thinness and relatively high conductivity compared to other layers.

#### Step 2: Calculate Heat Losses

##### Window Heat Loss





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- Same as before:
  - At 0°C: **44 W.**
  - At -20°C: **84 W.**
  - At -50°C: **144 W.**

#### Ventilation Heat Loss

- 25 m³/h,  $1.2509 \times \Delta T$  (from previous calculation):
  - At 0°C:  $1.2509 \times 22 = \mathbf{27.52 W.}$
  - At -20°C:  $1.2509 \times 42 = \mathbf{52.54 W.}$
  - At -50°C:  $1.2509 \times 72 = \mathbf{90.06 W.}$

#### Total Room Heat Loss (Q<sub>room</sub>)

- At 0°C:  $44 + 27.52 = \mathbf{71.52 W.}$
- At -20°C:  $84 + 52.54 = \mathbf{136.54 W.}$
- At -50°C:  $144 + 90.06 = \mathbf{234.06 W.}$

### Step 3: Solve for T<sub>surface</sub> and T<sub>water</sub> (Gap Temperature)

- **Q<sub>in</sub>:**  $U_{in} \times \text{Area} \times (T_{water} - T_{surface}) = Q_{room}.$
- $U_{in} = 4.45 \text{ W/m}^2\text{-K}, \text{Area} = 63 \text{ m}^2.$
- $Q_{in} = 4.45 \times 63 \times (T_{water} - T_{surface}) = 280.35 \times (T_{water} - T_{surface}).$
- **Surface to indoor air:**  $Q_{in} = h \times \text{Area} \times (T_{surface} - 22), h = 7.69 \text{ W/m}^2\text{-K} (1/0.13), 7.69 \times 63 = 484.47.$
- $484.47 \times (T_{surface} - 22) = Q_{room}.$

#### At 0°C

- $T_{surface} - 22 = 71.52 / 484.47 = 0.148, T_{surface} = \mathbf{22.15°C.}$
- $280.35 \times (T_{water} - 22.15) = 71.52.$
- $T_{water} - 22.15 = 71.52 / 280.35 = 0.255, T_{water} = 22.15 + 0.255 = \mathbf{22.40°C.}$
- $Q_{out} = 0.13 \times 63 \times (22.40 - 0) = 8.19 \times 22.40 = \mathbf{183.46 W.}$

#### At -20°C

- $T_{surface} - 22 = 136.54 / 484.47 = 0.282, T_{surface} = \mathbf{22.28°C.}$
- $280.35 \times (T_{water} - 22.28) = 136.54.$
- $T_{water} - 22.28 = 136.54 / 280.35 = 0.487, T_{water} = 22.28 + 0.487 = \mathbf{22.77°C.}$
- $Q_{out} = 0.13 \times 63 \times (22.77 - (-20)) = 8.19 \times 42.77 = \mathbf{350.28 W.}$

#### At -50°C

- $T_{surface} - 22 = 234.06 / 484.47 = 0.483, T_{surface} = \mathbf{22.48°C.}$
- $280.35 \times (T_{water} - 22.48) = 234.06.$
- $T_{water} - 22.48 = 234.06 / 280.35 = 0.835, T_{water} = 22.48 + 0.835 = \mathbf{23.32°C.}$
- $Q_{out} = 0.13 \times 63 \times (23.32 - (-50)) = 8.19 \times 73.32 = \mathbf{600.49 W.}$

#### Total Heat Exchanger Output (Q<sub>in</sub> + Q<sub>out</sub>)

- At 0°C:  $71.52 + 183.46 = \mathbf{254.98 W.}$
- At -20°C:  $136.54 + 350.28 = \mathbf{486.82 W.}$
- At -50°C:  $234.06 + 600.49 = \mathbf{834.55 W.}$

### Step 4: Water Temperatures in the Gap

- **Cross-sectional area:**  $10 \text{ m} \times 0.017 \text{ m} = 0.17 \text{ m}^2.$
- **Volumetric flow:**  $0.17 \times 0.01 = 0.0017 \text{ m}^3/\text{s} = 1.7 \text{ L/s.}$
- **Mass flow:** Water density  $\approx 1000 \text{ kg/m}^3, \dot{m} = 1000 \times 0.0017 = 1.7 \text{ kg/s.}$
- **Specific heat of water:**  $c = 4186 \text{ J/kg}\cdot\text{K.}$
- $\Delta T_{water} = Q_{total} / (\dot{m} \times c) = Q_{total} / (1.7 \times 4186) = Q_{total} / 7116.2.$
- At 0°C:  $\Delta T_{water} = 254.98 / 7116.2 = \mathbf{0.0358°C.}$
- At -20°C:  $\Delta T_{water} = 486.82 / 7116.2 = \mathbf{0.0684°C.}$
- At -50°C:  $\Delta T_{water} = 834.55 / 7116.2 = \mathbf{0.1173°C.}$

Using T<sub>water</sub> as the average:

- Warm side =  $T_{water} + \Delta T_{water}/2.$
- Cold side =  $T_{water} - \Delta T_{water}/2.$
- At 0°C:
  - Warm side =  $22.40 + 0.0179 = \mathbf{22.42°C.}$
  - Cold side =  $22.40 - 0.0179 = \mathbf{22.38°C.}$



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- At -20°C:
  - Warm side =  $22.77 + 0.0342 = 22.80^\circ\text{C}$ .
  - Cold side =  $22.77 - 0.0342 = 22.74^\circ\text{C}$ .
- At -50°C:
  - Warm side =  $23.32 + 0.0587 = 23.38^\circ\text{C}$ .
  - Cold side =  $23.32 - 0.0587 = 23.26^\circ\text{C}$ .

### Step 5: Verify Inner Surface Temperature

- $Q_{in} = 4.45 \times 63 \times (T_{water} - T_{surface})$ .
- At 0°C:  $71.52 = 280.35 \times (22.40 - T_{surface})$ ,  $T_{surface} = 22.40 - 0.255 = 22.15^\circ\text{C}$ .
- At -20°C:  $136.54 = 280.35 \times (22.77 - T_{surface})$ ,  $T_{surface} = 22.77 - 0.487 = 22.28^\circ\text{C}$ .
- At -50°C:  $234.06 = 280.35 \times (23.32 - T_{surface})$ ,  $T_{surface} = 23.32 - 0.835 = 22.48^\circ\text{C}$ .

Matches perfectly.

### Final Answers (Water System)

#### Heat Exchanger Output (Watts)

- At 0°C: **254.98 W** (same as air).
- At -20°C: **486.82 W** (vs. 486.74 W air).
- At -50°C: **834.55 W** (vs. 834.39 W air).

#### Water Temperature at Cold Side (Entering Heat Exchanger)

- At 0°C: **22.38°C**.
- At -20°C: **22.74°C**.
- At -50°C: **23.26°C**.

#### Water Temperature at Warm Side (Exiting Heat Exchanger)

- At 0°C: **22.42°C**.
- At -20°C: **22.80°C**.
- At -50°C: **23.38°C**.

#### Inner Wall/Ceiling Surface Temperature

- At 0°C: **22.15°C** (same as air).
- At -20°C: **22.28°C** (same as air).
- At -50°C: **22.48°C** (same as air).

### Comparison: Air vs. Water

#### Heat Exchanger Output

- **Air:** 254.98 W, 486.74 W, 834.39 W.
- **Water:** 254.98 W, 486.82 W, 834.55 W.
- **Difference:** Negligible (within rounding). The PP layer's small resistance doesn't significantly alter total heat loss, as  $Q_{room}$  and  $Q_{out}$  dominate.

#### Temperature Differences (Warm - Cold)

- **Air:**  $\Delta T_{air} = 0.83^\circ\text{C}$ ,  $1.59^\circ\text{C}$ ,  $2.73^\circ\text{C}$ .
  - Higher due to lower specific heat (1000 J/kg·K) and lower mass flow (0.306 kg/s).
- **Water:**  $\Delta T_{water} = 0.0358^\circ\text{C}$ ,  $0.0684^\circ\text{C}$ ,  $0.1173^\circ\text{C}$ .
  - Much lower due to higher specific heat (4186 J/kg·K) and higher mass flow (1.7 kg/s).

#### Cold Side Temperatures

- **Air:** 21.99°C, 21.97°C, 21.94°C (drops below 22°C, less ideal).
- **Water:** 22.38°C, 22.74°C, 23.26°C (stays above 22°C, more consistent).

#### Warm Side Temperatures

- **Air:** 22.82°C, 23.56°C, 24.67°C.
- **Water:** 22.42°C, 22.80°C, 23.38°C.
- **Water runs cooler** due to higher heat capacity, requiring less temperature elevation.

#### Practical Implications

- **Water:** Smaller  $\Delta T$  means more uniform temperature across the 12.6 m loop, easier to control, and less risk of cold spots. Higher density and heat capacity (4186 vs. 1000 J/kg·K) make it more efficient per unit volume, despite lower velocity (0.01 m/s vs. 1.5 m/s).
- **Air:** Larger  $\Delta T$  and lower cold-side temperatures indicate less thermal stability over the longer path. Higher velocity requires more fan power, though air is simpler to implement (no sealing).

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### Energy Input

- **Air:** Fan power for 0.255 m<sup>3</sup>/s at 1.5 m/s over 12.6 m is significant (pressure drop considerations).
- **Water:** Pump power for 1.7 L/s at 0.01 m/s is lower due to slower flow, though piping and sealing add complexity.

### Conclusion

Water provides tighter temperature control ( $\Delta T \sim 0.1^\circ\text{C}$  vs.  $\sim 2^\circ\text{C}$ ) and keeps the cold side above  $22^\circ\text{C}$ , enhancing comfort and efficiency. Total heat demand is nearly identical, as it's dictated by  $Q_{\text{room}}$  and  $U_{\text{out}}$ , not the medium. Let me know if you'd like to explore pump/fan power or other factors!



Now we return to the air system.  
What is the pressure decrease in the air loop?  
Energy for ventilation?  
Efficiency of usual fans to do this job?  
How many watts input into the fans?

Let's dive into the air circulation system for the 2.5 m wall + 3.8 m ceiling setup (63 m<sup>2</sup> total exterior surface) and calculate the pressure drop in the air loop, the energy required for ventilation, the efficiency of typical fans, and the power input needed for the fans. We're working with the air-based system from earlier: a 17 mm gap, 10 m width, airflow velocity of 1.5 m/s, and a total loop length of 12.6 m (6.3 m up + 6.3 m back).

### Step 1: Pressure Drop in the Air Loop

The pressure drop ( $\Delta P$ ) in the air loop depends on friction losses in the channel, calculated using the Darcy-Weisbach equation:

- $\Delta P = f \times (L / D_h) \times (\rho \times v^2 / 2)$ ,  
where:
- $f$  = friction factor (depends on Reynolds number and channel roughness),
- $L$  = length of the loop = 12.6 m,
- $D_h$  = hydraulic diameter,
- $\rho$  = air density  $\approx 1.2 \text{ kg/m}^3$ ,
- $v$  = velocity = 1.5 m/s.

#### Hydraulic Diameter ( $D_h$ )

For a rectangular channel (width  $\gg$  height):

- Cross-sectional area =  $10 \text{ m} \times 0.017 \text{ m} = 0.17 \text{ m}^2$ .
- Perimeter =  $2 \times (10 + 0.017) \approx 20.034 \text{ m}$  (width dominates).
- $D_h = 4 \times \text{Area} / \text{Perimeter} = 4 \times 0.17 / 20.034 \approx \mathbf{0.03397 \text{ m}}$  (close to  $2 \times$  gap width = 0.034 m).

#### Reynolds Number ( $Re$ )

- $Re = (\rho \times v \times D_h) / \mu$ .
- $\mu$  = viscosity of air  $\approx 1.8 \times 10^{-5} \text{ kg/m}\cdot\text{s}$ .
- $Re = (1.2 \times 1.5 \times 0.034) / (1.8 \times 10^{-5}) = 0.0612 / 0.000018 \approx \mathbf{3400}$ .
- Flow is transitional ( $Re > 2300$ ), trending toward turbulent.

#### Friction Factor ( $f$ )

For turbulent flow in a smooth channel, use the Blasius approximation:

- $f = 0.316 / Re^{0.25}$ .
- $Re^{0.25} = 3400^{0.25} \approx 6.74$ .
- $f = 0.316 / 6.74 \approx \mathbf{0.0469}$ .  
(Assuming smooth surfaces; roughness could increase  $f$  slightly, but polypropylene or similar linings suggest smoothness.)

#### Pressure Drop Calculation

- $L / D_h = 12.6 / 0.034 \approx 370.59$ .
- Dynamic pressure =  $\rho \times v^2 / 2 = 1.2 \times 1.5^2 / 2 = 1.2 \times 2.25 / 2 = \mathbf{1.35 \text{ Pa}}$ .
- $\Delta P = f \times (L / D_h) \times (\rho \times v^2 / 2) = 0.0469 \times 370.59 \times 1.35 \approx \mathbf{23.47 \text{ Pa}}$ .

So, the pressure drop in the loop is approximately **23.5 Pa**.

### Step 2: Energy for Ventilation (Power to Move Air)

The mechanical power required to overcome this pressure drop is:



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- **Power (P) = Volume flow rate  $\times$   $\Delta P$ .**

• Volume flow rate = Cross-sectional area  $\times$  velocity =  $0.17 \text{ m}^2 \times 1.5 \text{ m/s} = \mathbf{0.255 \text{ m}^3/\text{s}}$ .

•  $P = 0.255 \times 23.47 = \mathbf{5.98 \text{ W}}$ .

This is the theoretical energy needed to maintain airflow against friction losses, assuming no additional losses (e.g., bends, fittings).

#### Additional Losses

The loop includes bends (e.g., at the wall-to-ceiling transition) and fittings (e.g., fan inlets, heat exchanger). For simplicity:

• Assume  $2 \times 90^\circ$  bends + minor fittings  $\approx 10\text{-}20\%$  additional  $\Delta P$ .

• Adjusted  $\Delta P \approx 23.5 \times 1.15 = \mathbf{27 \text{ Pa}}$ .

• Adjusted Power =  $0.255 \times 27 \approx \mathbf{6.89 \text{ W}}$ .

Let's use **6.9 W** as a conservative estimate for the fluid power required.

---

### Step 3: Efficiency of Typical Fans

Fan efficiency ( $\eta$ ) varies by type and operating condition:

- **Axial fans:** Common for high flow, low pressure (e.g., 20-50 Pa).
  - Efficiency: 60-80% (high-end models, e.g., computer cooling or HVAC fans).
- **Centrifugal fans:** Better for higher pressure but less suited here.
  - Efficiency: 50-70%.
- Given the low  $\Delta P$  (27 Pa) and high flow (0.255 m<sup>3</sup>/s), an **axial fan** is likely used.

Assume a typical efficiency of **70%** (mid-range for good-quality axial fans at this operating point).

---

### Step 4: Power Input to the Fans

- **Electrical power input = Fluid power / Efficiency.**

• Fluid power = 6.9 W.

•  $\eta = 0.7$ .

• Power input =  $6.9 / 0.7 \approx \mathbf{9.86 \text{ W}}$ .

#### Multiple Fans

The system has fans in "every even-numbered lane" (Nylon strips every 200 mm across 10 m width = 50 lanes, 25 even lanes). If each fan handles a portion of the flow:

• Flow per lane =  $0.255 \text{ m}^3/\text{s} / 25 = 0.0102 \text{ m}^3/\text{s}$ .

• Area per lane =  $0.2 \text{ m} \times 0.017 \text{ m} = 0.0034 \text{ m}^2$ .

• Velocity check:  $0.0102 / 0.0034 = 3 \text{ m/s}$  (suggests uneven flow or shared fans; let's stick with 1.5 m/s system-wide, implying fewer, larger fans).

Realistically, 25 small fans may be impractical. Assume **2-4 larger fans** share the load:

• Total power = 9.86 W (system-wide).

• Per fan (if 2):  $9.86 / 2 \approx \mathbf{4.93 \text{ W each}}$ .

• Per fan (if 4):  $9.86 / 4 \approx \mathbf{2.47 \text{ W each}}$ .

Let's use **10 W total** (rounded) for simplicity, assuming 2 fans at 5 W each, accounting for minor inefficiencies or fittings.

---

## Final Answers

#### Pressure Decrease in the Air Loop

- **27 Pa** (including minor losses).

#### Energy for Ventilation (Fluid Power)

- **6.9 W** (theoretical power to move air).

#### Efficiency of Usual Fans

- **70%** (typical for axial fans at this flow and pressure).

#### Watts Input into the Fans

- **10 W total** (assuming 2 fans at ~5 W each, or 4 at ~2.5 W each, rounded for practicality).

---

## Validation



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- **Comparison:** A 0.255 m<sup>3</sup>/s flow at 27 Pa yielding 6.9 W is low but reasonable for a wide, short duct. HVAC systems often see 50-100 Pa for longer runs, but this 12.6 m loop with minimal bends aligns with the calculation.
- **Fan power:** 10 W is modest, reflecting the low pressure drop and high flow efficiency of the wide channel. Real-world fans might draw slightly more (15-20 W) due to motor losses or suboptimal sizing.

If you'd like to refine fan count, add specific fittings, or adjust efficiency, let me know!



Now back to the air and water comparison. The heat is produced by a heat pump. Even when the Watt for heating is nearly identical, the Watt electrical would be different. In both cases will be a heat pump with R290 as refrigerant. The heat pump takes the heat out from 15° C warm water. What COP and Watt electricity to the heat pump at the 3 temperatures can be expected at the air and water scenario?

Let's compare the air and water systems by incorporating a heat pump using R290 (propane) as the refrigerant, which extracts heat from a 15°C water source and delivers it to either the air or water circulating in the 17 mm gap. The heat pump's Coefficient of Performance (COP) and electrical power input (Watts) will depend on the temperature difference between the source (15°C) and the sink (the air or water temperature in the gap). Since the heating demands are nearly identical between air and water (254.98 W, ~486.8 W, ~834.5 W at 0°C, -20°C, -50°C), the key difference lies in the sink temperature and its impact on COP.

### Step 1: Define the Scenarios

#### Air System

- **Heat output (Q<sub>total</sub>):** 254.98 W (0°C), 486.74 W (-20°C), 834.39 W (-50°C).
- **Sink temperatures (Warm side):** 22.82°C, 23.56°C, 24.67°C (air exiting the heat exchanger).
- **Fan power:** ~10 W (additional electrical input).
- **Source temperature:** 15°C (water).

#### Water System

- **Heat output (Q<sub>total</sub>):** 254.98 W (0°C), 486.82 W (-20°C), 834.55 W (-50°C).
- **Sink temperatures (Warm side):** 22.42°C, 22.80°C, 23.38°C (water exiting the heat exchanger).
- **Pump power:** TBD (we'll estimate later).
- **Source temperature:** 15°C (water).

#### Heat Pump Details

- **Refrigerant:** R290 (propane), efficient and suitable for low-temperature applications.
- **Source:** 15°C water (assume a stable geothermal or waste heat source).
- **Sink:** Warm-side temperature of the air or water loop (delivery temperature).

### Step 2: Estimate COP for R290 Heat Pump

The COP of a heat pump is based on the Carnot efficiency, adjusted for real-world losses:

- **Ideal Carnot COP = T<sub>sink</sub> / (T<sub>sink</sub> - T<sub>source</sub>)** (in Kelvin).
- **Real COP ≈ η × Carnot COP**, where η (efficiency factor) for modern heat pumps is typically 0.4-0.5 due to compressor, heat exchanger, and cycle inefficiencies.
- Use η = 0.45 (mid-range for R290 systems).

Convert temperatures to Kelvin:

- Source = 15°C = 288 K.
- Sink (Air): 22.82°C = 295.97 K, 23.56°C = 296.71 K, 24.67°C = 297.82 K.
- Sink (Water): 22.42°C = 295.57 K, 22.80°C = 295.95 K, 23.38°C = 296.53 K.

#### Air System COP

- **At 0°C (T<sub>sink</sub> = 295.97 K):**
  - Carnot COP = 295.97 / (295.97 - 288) = 295.97 / 7.97 ≈ **37.13**.
  - Real COP = 0.45 × 37.13 ≈ **16.71**.
- **At -20°C (T<sub>sink</sub> = 296.71 K):**
  - Carnot COP = 296.71 / (296.71 - 288) = 296.71 / 8.71 ≈ **34.06**.
  - Real COP = 0.45 × 34.06 ≈ **15.33**.
- **At -50°C (T<sub>sink</sub> = 297.82 K):**



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- Carnot COP =  $297.82 / (297.82 - 288) = 297.82 / 9.82 \approx 30.33$ .
- Real COP =  $0.45 \times 30.33 \approx 13.65$ .

#### Water System COP

- **At 0°C (T<sub>sink</sub> = 295.57 K):**
  - Carnot COP =  $295.57 / (295.57 - 288) = 295.57 / 7.57 \approx 39.04$ .
  - Real COP =  $0.45 \times 39.04 \approx 17.57$ .
- **At -20°C (T<sub>sink</sub> = 295.95 K):**
  - Carnot COP =  $295.95 / (295.95 - 288) = 295.95 / 7.95 \approx 37.23$ .
  - Real COP =  $0.45 \times 37.23 \approx 16.75$ .
- **At -50°C (T<sub>sink</sub> = 296.53 K):**
  - Carnot COP =  $296.53 / (296.53 - 288) = 296.53 / 8.53 \approx 34.76$ .
  - Real COP =  $0.45 \times 34.76 \approx 15.64$ .

The water system has a higher COP because its sink temperatures are lower, reducing the temperature lift (T<sub>sink</sub> - T<sub>source</sub>).

### Step 3: Electrical Power Input to Heat Pump

- $W_{elec} = Q_{total} / COP$ .
- Q<sub>total</sub> is the heat delivered to the loop.

#### Air System

- **At 0°C:**  $W_{elec} = 254.98 / 16.71 \approx 15.26$  W.
- **At -20°C:**  $W_{elec} = 486.74 / 15.33 \approx 31.75$  W.
- **At -50°C:**  $W_{elec} = 834.39 / 13.65 \approx 61.13$  W.

#### Water System

- **At 0°C:**  $W_{elec} = 254.98 / 17.57 \approx 14.51$  W.
- **At -20°C:**  $W_{elec} = 486.82 / 16.75 \approx 29.06$  W.
- **At -50°C:**  $W_{elec} = 834.55 / 15.64 \approx 53.36$  W.

### Step 4: Add Circulation Power

#### Air System (Fans)

- Fan power = **10 W** (from previous calculation).
- **Total electrical input:**
  - At 0°C:  $15.26 + 10 = 25.26$  W.
  - At -20°C:  $31.75 + 10 = 41.75$  W.
  - At -50°C:  $61.13 + 10 = 71.13$  W.

#### Water System (Pump)

Estimate pump power:

- **Volume flow:**  $0.0017 \text{ m}^3/\text{s} = 1.7 \text{ L/s}$ .
- **Pressure drop:** Assume similar friction to air (adjusted for water).
  - $Re = (\rho \times v \times D_h) / \mu$ ,  $\rho = 1000 \text{ kg/m}^3$ ,  $v = 0.01 \text{ m/s}$ ,  $D_h = 0.034 \text{ m}$ ,  $\mu = 0.001 \text{ kg/m}\cdot\text{s}$ .
  - $Re = (1000 \times 0.01 \times 0.034) / 0.001 = 340$  (laminar).
  - $f$  (laminar) =  $64 / Re = 64 / 340 \approx 0.188$ .
  - $\Delta P = f \times (L / D_h) \times (\rho \times v^2 / 2) = 0.188 \times (12.6 / 0.034) \times (1000 \times 0.01^2 / 2)$ .
  - $\Delta P = 0.188 \times 370.59 \times (0.05) = 3.48$  Pa.
- **Power** = Volume flow  $\times \Delta P = 0.0017 \times 3.48 \approx 0.0059$  W (negligible, but unrealistic).
- Water systems typically have higher  $\Delta P$  due to fittings/pipes. Assume **50 Pa** (conservative for 12.6 m loop with heat exchanger):
  - Power =  $0.0017 \times 50 = 0.085$  W.
- Practical pump: Small circulator pumps (e.g., for radiators) use **5-15 W**. Assume **10 W** (comparable to fans).
- **Total electrical input:**
  - At 0°C:  $14.51 + 10 = 24.51$  W.
  - At -20°C:  $29.06 + 10 = 39.06$  W.
  - At -50°C:  $53.36 + 10 = 63.36$  W.

### Final Answers

#### Air System



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- **COP:** 16.71 (0°C), 15.33 (-20°C), 13.65 (-50°C).
- **Watts Electricity (Heat Pump + Fans):**
  - At 0°C: **25.26 W.**
  - At -20°C: **41.75 W.**
  - At -50°C: **71.13 W.**

#### Water System

- **COP:** 17.57 (0°C), 16.75 (-20°C), 15.64 (-50°C).
- **Watts Electricity (Heat Pump + Pump):**
  - At 0°C: **24.51 W.**
  - At -20°C: **39.06 W.**
  - At -50°C: **63.36 W.**

### Comparison

- **COP:** Water system has higher COP (1-2 points) due to lower sink temperatures (22.42–23.38°C vs. 22.82–24.67°C), reducing compressor work.
- **Electrical Input:** Water saves ~0.7–7.8 W (3–11%) thanks to higher COP, despite identical pump/fan power (10 W assumed). Savings increase at lower outdoor temperatures.
- **Practicality:** Water's lower  $\Delta T$  (0.036–0.117°C vs. 0.83–2.73°C) offers better thermal stability, but requires watertight sealing and pump maintenance.

The water system is more electrically efficient, especially at -50°C (63.36 W vs. 71.13 W), due to its tighter temperature control and higher COP. Want to tweak pump power or heat pump efficiency further?



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We have now the air and the water scenario.  
 Lets add the water-pipe scenario.  
 Walls filled with water inside are not usual.  
 The usual way are water pipes.  
 So I call it the water-pipe scenario.  
 Here is a common pipe, let's use the 16 mm variant.  
 Where we had in the air scenario the 17 mm gap,  
 there is now each meter of wall width about 63 m of this alu plastic pipe in curves, that the average distance from pipe to pipe is 100 mm.  
 So the 9 m width wall and ceiling has 9 such pipe systems.  
 The water travels with 1.5 m/sec in the pipes.  
 The place between in the 17 mm is filled with gypsum.



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 So the 9 m width wall and ceiling has 9 such pipe systems.  
 The water travels with 1.5 m/sec in the pipes.  
 The place between in the 17 mm is filled with gypsum.

Let's complete the analysis for the "water-pipe scenario" using the PELIA Metallverbundrohr (16 x 2 mm variant) and compare it with the air and water scenarios. The heat pump with R290 refrigerant extracts heat from 15°C water to maintain 22°C indoors at outdoor temperatures of 0°C, -20°C, and -50°C. I'll calculate the heat exchanger output, water temperatures in the pipes, COP, and electrical input, including pump power, then compare all three systems.

### Step 1: Define the Water-Pipe Scenario

- **Structure:** 2.5 m wall + 3.8 m ceiling = 63 m<sup>2</sup> (10 m width).

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- **Window:**  $2.5 \text{ m}^2$ ,  $U = 0.8 \text{ W/m}^2\text{-K}$ .
- **Ventilation:**  $25 \text{ m}^3/\text{h}$ , 85% heat recovery.
- **Pipe system:**
  - PELIA  $16 \times 2 \text{ mm}$ : Outer diameter = 16 mm, inner diameter = 12 mm, Al thickness = 0.2 mm (assuming standard from doc).
  - 63 m pipe per 1 m width, 100 mm spacing, total =  $9 \times 63 = 567 \text{ m}$ .
  - Embedded in 17 mm gypsum ( $k = 0.25 \text{ W/m-K}$ ).
- **Water flow:** 1.5 m/s.
- **Heat pump:** R290, source =  $15^\circ\text{C}$  (288 K), sink = water in pipes.

## Step 2: Calculate Heat Losses

### Window Heat Loss

- $0^\circ\text{C}$ : **44 W**.
- $-20^\circ\text{C}$ : **84 W**.
- $-50^\circ\text{C}$ : **144 W**.

### Ventilation Heat Loss

- $1.2509 \times \Delta T$ :
  - $0^\circ\text{C}$ : **27.52 W**.
  - $-20^\circ\text{C}$ : **52.54 W**.
  - $-50^\circ\text{C}$ : **90.06 W**.

### Total Room Heat Loss ( $Q_{\text{room}}$ )

- $0^\circ\text{C}$ :  $44 + 27.52 = 71.52 \text{ W}$ .
- $-20^\circ\text{C}$ :  $84 + 52.54 = 136.54 \text{ W}$ .
- $-50^\circ\text{C}$ :  $144 + 90.06 = 234.06 \text{ W}$ .

## Step 3: Heat Transfer and Pipe Water Temperature

### U-values Adjusted

- **Inner side (pipes to indoor air):**
  - Gypsum layer: 17 mm,  $R = 0.017 / 0.25 = 0.068 \text{ m}^2\text{-K/W}$ .
  - Surface resistance:  $R_{\text{si}} = 0.13 \text{ m}^2\text{-K/W}$ .
  - Pipe wall (PE-X/Al/PE):  $R = 0.0046 \text{ m}^2\text{-K/W}$  (from doc).
  - Total  $R_{\text{in}} = 0.068 + 0.13 + 0.0046 = 0.2026 \text{ m}^2\text{-K/W}$ ,  $U_{\text{in}} = 1 / 0.2026 \approx 4.94 \text{ W/m}^2\text{-K}$ .
  - Effective  $U_{\text{in}}$  depends on pipe spacing; adjust later with heat transfer model.
- **Outer side:**  $U_{\text{out}} = 0.13 \text{ W/m}^2\text{-K}$  (glasswool, steel, PU), adjust with gypsum:
  - $R_{\text{out}} = 1 / 0.13 + 0.068 = 7.692 + 0.068 = 7.76 \text{ m}^2\text{-K/W}$ ,  $U_{\text{out}} = 0.129 \text{ W/m}^2\text{-K}$ .

### Simplified Heat Transfer

- $Q_{\text{in}} = U_{\text{in}} \times \text{Area} \times (T_{\text{water}} - T_{\text{surface}}) = Q_{\text{room}}$ .
- $Q_{\text{out}} = U_{\text{out}} \times \text{Area} \times (T_{\text{water}} - T_{\text{out}})$ .
- **Surface to indoor:**  $Q_{\text{in}} = 7.69 \times 63 \times (T_{\text{surface}} - 22)$ ,  $h = 1/0.13$ .
- **At  $0^\circ\text{C}$ :**
  - $484.47 \times (T_{\text{surface}} - 22) = 71.52$ ,  $T_{\text{surface}} = 22.15^\circ\text{C}$ .
  - $4.94 \times 63 \times (T_{\text{water}} - 22.15) = 71.52$ ,  $311.22 \times (T_{\text{water}} - 22.15) = 71.52$ .
  - $T_{\text{water}} - 22.15 = 0.230$ ,  $T_{\text{water}} = 22.38^\circ\text{C}$ .
  - $Q_{\text{out}} = 0.129 \times 63 \times (22.38 - 0) = 8.127 \times 22.38 = 181.88 \text{ W}$ .
  - $Q_{\text{total}} = 71.52 + 181.88 = 253.40 \text{ W}$ .
- **At  $-20^\circ\text{C}$ :**
  - $T_{\text{surface}} = 22 + 136.54 / 484.47 = 22.28^\circ\text{C}$ .
  - $311.22 \times (T_{\text{water}} - 22.28) = 136.54$ ,  $T_{\text{water}} = 22.28 + 0.439 = 22.72^\circ\text{C}$ .
  - $Q_{\text{out}} = 0.129 \times 63 \times (22.72 - (-20)) = 8.127 \times 42.72 = 347.19 \text{ W}$ .
  - $Q_{\text{total}} = 136.54 + 347.19 = 483.73 \text{ W}$ .
- **At  $-50^\circ\text{C}$ :**
  - $T_{\text{surface}} = 22 + 234.06 / 484.47 = 22.48^\circ\text{C}$ .
  - $311.22 \times (T_{\text{water}} - 22.48) = 234.06$ ,  $T_{\text{water}} = 22.48 + 0.752 = 23.23^\circ\text{C}$ .
  - $Q_{\text{out}} = 0.129 \times 63 \times (23.23 - (-50)) = 8.127 \times 73.23 = 594.99 \text{ W}$ .
  - $Q_{\text{total}} = 234.06 + 594.99 = 829.05 \text{ W}$ .



## Step 4: Water Temperatures in Pipes





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- **Pipe area:**  $\pi \times (0.006)^2 = 1.1309 \times 10^{-4} \text{ m}^2$ .
- **Volume flow:**  $1.5 \times 1.1309 \times 10^{-4} \times 9 = 0.001527 \text{ m}^3/\text{s}$  (1.527 L/s).
- **Mass flow:**  $1000 \times 0.001527 = 1.527 \text{ kg/s}$ .
- **$\Delta T_{\text{water}} = Q_{\text{total}} / (\dot{m} \times 4186)$ ,  $\dot{m} \times c = 6389.62$ .**
- $0^\circ\text{C}$ :  $253.40 / 6389.62 = \mathbf{0.0397^\circ\text{C}}$ .
- $-20^\circ\text{C}$ :  $483.73 / 6389.62 = \mathbf{0.0757^\circ\text{C}}$ .
- $-50^\circ\text{C}$ :  $829.05 / 6389.62 = \mathbf{0.1298^\circ\text{C}}$ .
- **Warm side, Cold side:**
  - $0^\circ\text{C}$ : Warm =  $22.38 + 0.0199 = \mathbf{22.40^\circ\text{C}}$ , Cold =  $\mathbf{22.36^\circ\text{C}}$ .
  - $-20^\circ\text{C}$ : Warm =  $22.72 + 0.0379 = \mathbf{22.76^\circ\text{C}}$ , Cold =  $\mathbf{22.68^\circ\text{C}}$ .
  - $-50^\circ\text{C}$ : Warm =  $23.23 + 0.0649 = \mathbf{23.29^\circ\text{C}}$ , Cold =  $\mathbf{23.17^\circ\text{C}}$ .

### Step 5: COP and Heat Pump Power

- **Sink = Warm side**, Source = 288 K,  $\eta = 0.45$ .

#### COP

- **$0^\circ\text{C}$  (295.55 K):** Carnot =  $295.55 / 7.55 = 39.15$ , COP =  $0.45 \times 39.15 = \mathbf{17.62}$ .
- **$-20^\circ\text{C}$  (295.91 K):** Carnot =  $295.91 / 7.91 \approx 37.41$ , COP =  $0.45 \times 37.41 = \mathbf{16.83}$ .
- **$-50^\circ\text{C}$  (296.44 K):** Carnot =  $296.44 / 8.44 \approx 35.12$ , COP =  $0.45 \times 35.12 = \mathbf{15.80}$ .

#### Heat Pump Electrical Input

- **$W_{\text{elec}} = Q_{\text{total}} / \text{COP}$ .**
- $0^\circ\text{C}$ :  $253.40 / 17.62 \approx \mathbf{14.38 \text{ W}}$ .
- $-20^\circ\text{C}$ :  $483.73 / 16.83 \approx \mathbf{28.74 \text{ W}}$ .
- $-50^\circ\text{C}$ :  $829.05 / 15.80 \approx \mathbf{52.47 \text{ W}}$ .

### Step 6: Pump Power

- **Pressure drop:** Darcy-Weisbach,  $f = 0.316 / \text{Re}^{0.25}$ .
- $\text{Re} = (1000 \times 1.5 \times 0.012) / 0.001 = 18,000$  (turbulent).
- $f = 0.316 / 18,000^{0.25} \approx 0.0275$ .
- $\Delta P = f \times (L / D) \times (\rho \times v^2 / 2) = 0.0275 \times (567 / 0.012) \times (1000 \times 1.5^2 / 2)$ .
- $\Delta P = 0.0275 \times 47,250 \times 1125 = \mathbf{1,461 \text{ Pa}}$ .
- **Power** =  $0.001527 \times 1461 \approx \mathbf{2.23 \text{ W}}$ .
- Practical pump (fittings, bends): Assume **10 W** (typical small circulator).

#### Total Electrical Input

- $0^\circ\text{C}$ :  $14.38 + 10 = \mathbf{24.38 \text{ W}}$ .
- $-20^\circ\text{C}$ :  $28.74 + 10 = \mathbf{38.74 \text{ W}}$ .
- $-50^\circ\text{C}$ :  $52.47 + 10 = \mathbf{62.47 \text{ W}}$ .

### Comparison: Air vs. Water vs. Water-Pipe

#### Heat Output (W)

- **Air:** 254.98, 486.74, 834.39.
- **Water:** 254.98, 486.82, 834.55.
- **Water-Pipe:** 253.40, 483.73, 829.05.
- *Similar, slight variance due to U-value adjustments.*

#### COP

- **Air:** 16.71, 15.33, 13.65.
- **Water:** 17.57, 16.75, 15.64.
- **Water-Pipe:** 17.62, 16.83, 15.80.
- *Water and Water-Pipe higher due to lower sink temps (22.4–23.3°C vs. 22.8–24.7°C).*

#### Electrical Input (W, incl. circulation)

- **Air:** 25.26, 41.75, 71.13 (10 W fans).
- **Water:** 24.51, 39.06, 63.36 (10 W pump).
- **Water-Pipe:** 24.38, 38.74, 62.47 (10 W pump).
- *Water-Pipe most efficient, followed by Water, then Air.*



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### Circulation Power

- **Air:** 10 W (fans, low  $\Delta P = 27$  Pa).
- **Water:** 10 W (assumed,  $\Delta P \approx 50$  Pa).
- **Water-Pipe:** 10 W (assumed,  $\Delta P \approx 1461$  Pa theoretical, adjusted for practicality).

### Practical Notes

- **Air:** Simple, higher  $\Delta T$  (0.83–2.73°C), less efficient COP.
- **Water:** Uniform  $\Delta T$  (0.036–0.117°C), sealing complexity.
- **Water-Pipe:** Standard practice,  $\Delta T$  (0.04–0.13°C), higher pump  $\Delta P$  but efficient heat transfer via gypsum.

The **water-pipe scenario** edges out slightly in electrical efficiency due to its lower sink temperature and comparable circulation power, making it a practical and efficient choice for this wall heating system. Want to refine pump specifics or adjust assumptions?

Did You consider that the inner wall has no uniform surface temperature because some parts are direct over a pipe and some part are 50 mm distant from the next pipe?

You're absolutely right—I didn't explicitly account for the non-uniform surface temperature in the water-pipe scenario due to the 100 mm pipe spacing, where some areas are directly over the 16 mm pipes and others are up to 50 mm away (half the spacing). This variation affects heat transfer from the pipes to the indoor air, as well as the average surface temperature and overall system performance. Let's refine the calculation by modeling the heat distribution through the gypsum layer, considering the pipe spacing, and adjust the results accordingly.

## Step 1: Reassess the Water-Pipe Scenario with Non-Uniform Temperature

- **Setup:** 63 m<sup>2</sup> (2.5 m wall + 3.8 m ceiling, 10 m width), 567 m of 16 × 2 mm PELIA pipe (9 × 63 m), 100 mm spacing, embedded in 17 mm gypsum ( $k = 0.25$  W/m-K), water at 1.5 m/s.
- **Heat losses:**  $Q_{\text{room}} = 71.52$  W (0°C), 136.54 W (-20°C), 234.06 W (-50°C).
- **Goal:** Maintain 22°C indoors, adjust  $T_{\text{water}}$  and  $Q_{\text{total}}$  for non-uniform heat flux.

### Pipe Spacing and Heat Distribution

- **Pipe coverage:** 10 pipes/m width (100 mm spacing), outer diameter = 16 mm.
- **Gypsum layer:** 17 mm thick, pipes centered (8.5 mm to each surface).
- **Distance variation:**
  - Directly over pipe: ~8.5 mm to inner surface.
  - Midpoint (50 mm from pipe center): Heat spreads laterally through gypsum.

Heat flows radially from the pipes through the gypsum to the inner surface, then to the room. The temperature drops from the pipe to the surface, with a gradient due to spacing. We'll use a simplified 1D heat transfer model with an effective U-value, then validate with a temperature profile.

## Step 2: Effective Heat Transfer Model

### Inner Side (Pipes to Indoor Air)

- **Pipe-to-surface resistance:** Gypsum over pipe (8.5 mm),  $R_{\text{gypsum}} = 0.0085 / 0.25 = 0.034$  m<sup>2</sup>-K/W.
- **Pipe wall:**  $R_{\text{pipe}} = 0.0046$  m<sup>2</sup>-K/W (from PELIA doc).
- **Surface resistance:**  $R_{\text{si}} = 0.13$  m<sup>2</sup>-K/W.
- **Lateral spread:** For 100 mm spacing, heat conducts laterally. Use a shape factor for cylindrical pipes in a plane:
  - Shape factor ( $S$ )  $\approx (2\pi L) / \ln(4h/d)$ ,  $h = 8.5$  mm,  $d = 16$  mm,  $L = 1$  m (per meter width).
  - $S \approx (2\pi) / \ln(4 \times 0.0085 / 0.016) = 6.28 / \ln(2.125) \approx 8.28$  W/m-K (approximate).
  - Effective  $R_{\text{gypsum}} = 1 / (k \times S) \times \text{spacing} = 1 / (0.25 \times 8.28) \times 0.1 = 0.048$  m<sup>2</sup>-K/W (per pipe segment).
- **Total R<sub>in</sub> (per pipe area):**  $0.0046 + 0.048 + 0.13 \approx 0.1826$  m<sup>2</sup>-K/W,  $U_{\text{in}} = 1 / 0.1826 \approx 5.48$  W/m<sup>2</sup>-K (local).
- **Effective U<sub>in</sub>:** Pipes cover ~16 mm / 100 mm = 16% of the surface directly, but heat spreads. Average  $U_{\text{in}}$  drops due to spacing:
  - Adjust with fin analogy or empirical floor heating data:  $U_{\text{eff}} \approx 4\text{--}5$  W/m<sup>2</sup>-K. Use **4.5 W/m<sup>2</sup>-K** (conservative, reflecting spacing).

### Outer Side

- $U_{\text{out}} = 0.129$  W/m<sup>2</sup>-K (adjusted with gypsum, as before).

### Solve with Non-Uniformity

- $Q_{\text{in}} = U_{\text{in,eff}} \times \text{Area} \times (T_{\text{avg}} - T_{\text{surface,avg}}) = Q_{\text{room}}$ .
- $Q_{\text{out}} = U_{\text{out}} \times \text{Area} \times (T_{\text{avg}} - T_{\text{out}})$ .



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•  $T_{avg}$  = average pipe water temperature,  $T_{surface\_avg}$  = average inner surface temperature.

• **At 0°C:**

- $4.5 \times 63 \times (T_{avg} - T_{surface\_avg}) = 71.52$ ,  $283.5 \times (T_{avg} - T_{surface\_avg}) = 71.52$ .
- $T_{avg} - T_{surface\_avg} = 0.252$ , assume  $T_{surface\_avg} = 22.15^\circ\text{C}$  (weighted average).
- $T_{avg} = 22.15 + 0.252 = \mathbf{22.40^\circ\text{C}}$ .
- $Q_{out} = 0.129 \times 63 \times (22.40 - 0) = 8.127 \times 22.40 = \mathbf{182.05\text{ W}}$ .
- $Q_{total} = 71.52 + 182.05 = \mathbf{253.57\text{ W}}$ .

• **At -20°C:**

- $T_{surface\_avg} = 22.28^\circ\text{C}$ ,  $283.5 \times (T_{avg} - 22.28) = 136.54$ .
- $T_{avg} - 22.28 = 0.482$ ,  $T_{avg} = \mathbf{22.76^\circ\text{C}}$ .
- $Q_{out} = 8.127 \times (22.76 - (-20)) = 8.127 \times 42.76 = \mathbf{347.51\text{ W}}$ .
- $Q_{total} = 136.54 + 347.51 = \mathbf{484.05\text{ W}}$ .

• **At -50°C:**

- $T_{surface\_avg} = 22.48^\circ\text{C}$ ,  $283.5 \times (T_{avg} - 22.48) = 234.06$ .
- $T_{avg} - 22.48 = 0.826$ ,  $T_{avg} = \mathbf{23.31^\circ\text{C}}$ .
- $Q_{out} = 8.127 \times (23.31 - (-50)) = 8.127 \times 73.31 = \mathbf{595.60\text{ W}}$ .
- $Q_{total} = 234.06 + 595.60 = \mathbf{829.66\text{ W}}$ .

#### Surface Temperature Variation

- **Over pipe:**  $T_{surface} \approx T_{water} - Q_{in} / (h \times A_{pipe})$ ,  $h = 7.69\text{ W/m}^2\cdot\text{K}$ ,  $A_{pipe} = 567\text{ m} \times \pi \times 0.016 = 28.5\text{ m}^2$ .
  - 0°C:  $T_{surface} \approx 22.40 - 71.52 / (7.69 \times 63) \approx 22.25^\circ\text{C}$ .
  - Midpoint (50 mm): Lateral conduction,  $\Delta T \approx (Q_{in} / \text{Area}) \times (\text{distance} / k) \approx (71.52 / 63) \times (0.05 / 0.25) \approx 0.23^\circ\text{C}$  lower,  $\sim \mathbf{22.02^\circ\text{C}}$ .
- Range:  $\sim 22.02\text{--}22.25^\circ\text{C}$ , average =  $22.15^\circ\text{C}$ , sufficient for comfort.

#### Step 4: Water Temperatures

- **Mass flow:**  $1.527\text{ kg/s}$ ,  $\dot{m} \times c = 6389.62$ .
- **$\Delta T_{water}$ :**
  - 0°C:  $253.57 / 6389.62 = \mathbf{0.0397^\circ\text{C}}$ .
  - -20°C:  $484.05 / 6389.62 = \mathbf{0.0758^\circ\text{C}}$ .
  - -50°C:  $829.66 / 6389.62 = \mathbf{0.1299^\circ\text{C}}$ .
- **Warm side, Cold side:**
  - 0°C: Warm =  $22.40 + 0.0199 = \mathbf{22.42^\circ\text{C}}$ , Cold =  $\mathbf{22.38^\circ\text{C}}$ .
  - -20°C: Warm =  $22.76 + 0.0379 = \mathbf{22.80^\circ\text{C}}$ , Cold =  $\mathbf{22.72^\circ\text{C}}$ .
  - -50°C: Warm =  $23.31 + 0.0650 = \mathbf{23.37^\circ\text{C}}$ , Cold =  $\mathbf{23.24^\circ\text{C}}$ .

#### Step 5: COP and Electrical Input

- Sink = Warm side,  $\eta = 0.45$ .

#### COP

- 0°C (295.57 K): 39.16, COP = **17.62**.
- -20°C (295.95 K): 37.34, COP = **16.80**.
- -50°C (296.52 K): 34.91, COP = **15.71**.

#### Heat Pump Power

- 0°C:  $253.57 / 17.62 \approx \mathbf{14.39\text{ W}}$ .
- -20°C:  $484.05 / 16.80 \approx \mathbf{28.81\text{ W}}$ .
- -50°C:  $829.66 / 15.71 \approx \mathbf{52.81\text{ W}}$ .

#### Pump Power

- $\Delta P \approx 1461\text{ Pa}$  (turbulent), Power =  $0.001527 \times 1461 \approx 2.23\text{ W}$ , assume **10 W** (practical).
- **Total:**
  - 0°C:  $14.39 + 10 = \mathbf{24.39\text{ W}}$ .
  - -20°C:  $28.81 + 10 = \mathbf{38.81\text{ W}}$ .
  - -50°C:  $52.81 + 10 = \mathbf{62.81\text{ W}}$ .

#### Revised Comparison

##### Heat Output (W)

- **Air:** 254.98, 486.74, 834.39.
- **Water:** 254.98, 486.82, 834.55.



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- **Water-Pipe:** 253.57, 484.05, 829.66.

#### COP

- **Air:** 16.71, 15.33, 13.65.
- **Water:** 17.57, 16.75, 15.64.
- **Water-Pipe:** 17.62, 16.80, 15.71.

#### Electrical Input (W)

- **Air:** 25.26, 41.75, 71.13.
- **Water:** 24.51, 39.06, 63.36.
- **Water-Pipe:** 24.39, 38.81, 62.81.

#### Surface Temperature

- **Air:** Uniform (22.15–22.48°C).
- **Water:** Uniform (22.15–22.48°C).
- **Water-Pipe:** ~22.0–22.3°C (0°C), ~22.1–22.4°C (-20°C), ~22.3–22.6°C (-50°C), average aligns with prior values.

### Impact of Non-Uniformity

- **Adjustment:**  $U_{in}$  reduced to 4.5 W/m<sup>2</sup>·K (from 4.94) reflects lower effective heat transfer due to spacing, increasing  $T_{water}$  slightly (e.g., 22.40°C vs. 22.38°C at 0°C).
- **Performance:** Results shift minimally (0.01–0.3 W difference in  $W_{elec}$ ), as  $Q_{room}$  drives the system, and gypsum evens out gradients ( $\Delta T_{surface} \approx 0.2$ – $0.3$ °C).
- **Comfort:** Variation is small, acceptable for radiant heating (typical floor heating allows 2–3°C).

The water-pipe scenario remains competitive, with non-uniformity having a minor impact due to gypsum's conductivity. Want to explore tighter spacing or a 2D heat flow model?



2025-03-15 floor for floor only.pdf  
File

Floor, wall and ceiling heated is not usual.  
So we add now the floor only scenario.  
There is at the bottom 3.8 m floor. See attached PDF.  
Inside the floor are the same 16 mm pipes like in the water pipe scenario.  
Each 2 m are about 76 m pipes in the floor with 100 mm distance in average from the next pipe.  
So over the 10 m width are 5 groups, each 2 m with about 76 m pipes.  
Below the floor is also 22° C.  
The 17 mm gap from the air scenario is now filled with mineral wool like the next layer towards outside.  
The floor heating has to compensate the thermal losses from wall, ceiling, window and air exchange.  
So additional to the usual data, what is the surface temperature of the floor?

Let's add the "floor-only scenario" to our comparison, where the heating is provided solely by 16 mm PELIA Metallverbundrohr pipes embedded in a 3.8 m × 10 m floor, compensating for the thermal losses from the wall, ceiling, window, and ventilation. The 17 mm gap in the wall and ceiling (previously air or water-filled) is now filled with mineral wool, matching the outer insulation layer's properties. We'll calculate the heat pump output, COP, electrical input, and the floor surface temperature, then compare with the air, water, and water-pipe scenarios.

### Step 1: Define the Floor-Only Scenario

- **Structure:**
  - Floor: 3.8 m × 10 m = 38 m<sup>2</sup>.
  - Wall: 2.5 m × 10 m = 25 m<sup>2</sup>.
  - Ceiling: 3.8 m × 10 m = 38 m<sup>2</sup>.
  - Total exterior (wall + ceiling): 63 m<sup>2</sup>.
  - Window: 2.5 m<sup>2</sup>, U = 0.8 W/m<sup>2</sup>·K (in wall).
- **Ventilation:** 25 m<sup>3</sup>/h, 85% heat recovery.
- **Floor heating:**
  - PELIA 16 × 2 mm pipes (inner diameter 12 mm).
  - 76 m pipe per 2 m width, 100 mm spacing, 5 groups over 10 m = 5 × 76 m = **380 m total**.
  - Embedded in 110 mm lightweight concrete (from Ubakus: 100 mm concrete + 10 mm oak).



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- **Wall/Ceiling:**
  - 17 mm gap now mineral wool (assume glasswool,  $k = 0.035$  W/m·K, same as outer layer).
  - Original  $U_{out} = 0.13$  W/m<sup>2</sup>·K (glasswool, steel, PU foam, steel).
  - Inner layer: 12.5 mm gypsum board ( $U_{in} = 4.55$  W/m<sup>2</sup>·K from prior).
- **Conditions:** Indoor = 22°C, Outdoor = 0°C, -20°C, -50°C, below floor = 22°C (e.g., basement or adjacent room).
- **Heat pump:** R290, source = 15°C water.

## Step 2: Adjust Wall/Ceiling U-value

- **Original wall:**
  - $U = 0.12$  W/m<sup>2</sup>·K (unheated),  $U_{out} = 0.13$  W/m<sup>2</sup>·K (air gap to outside).
  - 17 mm air gap replaced with mineral wool:  $R = 0.017 / 0.035 = 0.486$  m<sup>2</sup>·K/W.
- **New U-value:**
  - $R_{out} = 1 / 0.13 = 7.692$  m<sup>2</sup>·K/W, add  $R_{mw} = 0.486$ ,  $R_{total} = 8.178$  m<sup>2</sup>·K/W.
  - $U_{out\_new} = 1 / 8.178 = 0.122$  W/m<sup>2</sup>·K.
  - Whole wall:  $R_{in} = 1 / 4.55 = 0.22$  m<sup>2</sup>·K/W,  $R_{total} = 0.22 + 8.178 = 8.398$  m<sup>2</sup>·K/W,  $U = 1 / 8.398 = 0.119$  W/m<sup>2</sup>·K (close to original 0.12 W/m<sup>2</sup>·K, validating consistency).

## Step 3: Calculate Heat Losses

### Window

- 0°C:  $0.8 \times 2.5 \times (22 - 0) = 44$  W.
- -20°C:  $0.8 \times 2.5 \times (22 - (-20)) = 84$  W.
- -50°C:  $0.8 \times 2.5 \times (22 - (-50)) = 144$  W.

### Ventilation

- $1.2509 \times \Delta T$ :
  - 0°C: **27.52 W.**
  - -20°C: **52.54 W.**
  - -50°C: **90.06 W.**

### Wall + Ceiling (Unheated)

- Area = 63 m<sup>2</sup>,  $U = 0.119$  W/m<sup>2</sup>·K.
- 0°C:  $0.119 \times 63 \times (22 - 0) = 7.497 \times 22 = 164.93$  W.
- -20°C:  $7.497 \times (22 - (-20)) = 7.497 \times 42 = 314.87$  W.
- -50°C:  $7.497 \times (22 - (-50)) = 7.497 \times 72 = 539.78$  W.

### Floor

- Below floor = 22°C, no heat loss ( $T_{inside} = T_{below}$ ).

### Total Heat Loss (Q\_room)

- 0°C:  $44 + 27.52 + 164.93 = 236.45$  W.
- -20°C:  $84 + 52.54 + 314.87 = 451.41$  W.
- -50°C:  $144 + 90.06 + 539.78 = 773.84$  W.

## Step 4: Floor Heating and Surface Temperature

- **Floor structure** (Ubakus):
  - 10 mm oak ( $k = 0.18$  W/m·K,  $R = 0.056$  m<sup>2</sup>·K/W).
  - 100 mm lightweight concrete ( $k = 1.3$  W/m·K,  $R = 0.077$  m<sup>2</sup>·K/W).
  - $R_{si} = 0.1$  m<sup>2</sup>·K/W,  $R_{se} = 0.04$  m<sup>2</sup>·K/W (adjusted for below).
  - Total  $R = 0.1 + 0.056 + 0.077 + 0.04 = 0.273$  m<sup>2</sup>·K/W,  $U = 3.67$  W/m<sup>2</sup>·K (unheated).
- **Pipes:** 76 m per 2 m width, 100 mm spacing, 38 m<sup>2</sup> floor = 380 m total.

### Heat Transfer

- **$Q_{floor} = U_{floor\_eff} \times Area \times (T_{surface\_avg} - 22) = Q_{room}$ .**
- **Pipe to surface:** Adjust for spacing (similar to water-pipe scenario).
- **$U_{floor\_eff}$ :** Ubakus gives heated  $U_{eff} = 19.53$  W/m<sup>2</sup>·K, but that's with 24°C water and 22°C outside. Recalculate:
  - $R_{concrete}$  (8.5 mm to surface) =  $0.0085 / 1.3 = 0.0065$  m<sup>2</sup>·K/W.
  - $R_{oak} = 0.056$ ,  $R_{si} = 0.1$ ,  $R_{pipe} = 0.0046$ .
  - Effective  $R = 0.0065 + 0.056 + 0.1 + 0.0046 = 0.1671$  m<sup>2</sup>·K/W,  $U_{local} = 5.98$  W/m<sup>2</sup>·K.
  - Spacing adjustment:  $\sim 4.5$  W/m<sup>2</sup>·K (consistent with prior).
- **$T_{surface\_avg}$ :**



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- 0°C:  $4.5 \times 38 \times (T_{\text{surface\_avg}} - 22) = 236.45$ ,  $T_{\text{surface\_avg}} - 22 = 1.384$ ,  $T_{\text{surface\_avg}} = \mathbf{23.38^\circ\text{C}}$ .
- -20°C:  $171 \times (T_{\text{surface\_avg}} - 22) = 451.41$ ,  $T_{\text{surface\_avg}} = 22 + 2.64 = \mathbf{24.64^\circ\text{C}}$ .
- -50°C:  $171 \times (T_{\text{surface\_avg}} - 22) = 773.84$ ,  $T_{\text{surface\_avg}} = 22 + 4.53 = \mathbf{26.53^\circ\text{C}}$ .

#### Pipe Water Temperature ( $T_{\text{water}}$ )

- $Q_{\text{floor}} = h \times A_{\text{pipe}} \times (T_{\text{water}} - T_{\text{surface}})$ ,  $h \approx 20 \text{ W/m}^2\text{-K}$  (pipe to concrete convection).
- $A_{\text{pipe}} = 380 \text{ m} \times \pi \times 0.016 = 19.11 \text{ m}^2$ .
- 0°C:  $20 \times 19.11 \times (T_{\text{water}} - 23.38) = 236.45$ ,  $T_{\text{water}} - 23.38 = 0.619$ ,  $T_{\text{water}} = \mathbf{24.00^\circ\text{C}}$ .
- -20°C:  $382.2 \times (T_{\text{water}} - 24.64) = 451.41$ ,  $T_{\text{water}} = 24.64 + 1.18 = \mathbf{25.82^\circ\text{C}}$ .
- -50°C:  $382.2 \times (T_{\text{water}} - 26.53) = 773.84$ ,  $T_{\text{water}} = 26.53 + 2.03 = \mathbf{28.56^\circ\text{C}}$ .

#### $\Delta T_{\text{water}}$

- $\dot{m} = 1.527 \text{ kg/s}$  (380 m, 1.5 m/s),  $\dot{m} \times c = 6389.62$ .
- 0°C:  $236.45 / 6389.62 = \mathbf{0.037^\circ\text{C}}$ .
- -20°C:  $451.41 / 6389.62 = \mathbf{0.071^\circ\text{C}}$ .
- -50°C:  $773.84 / 6389.62 = \mathbf{0.121^\circ\text{C}}$ .
- Warm: 24.02°C, 25.86°C, 28.60°C; Cold: 23.98°C, 25.79°C, 28.48°C.

### Step 5: COP and Electrical Input

- Sink = Warm side,  $\eta = 0.45$ .

#### COP

- 0°C (297.17 K):  $297.17 / 9.17 = 32.41$ ,  $\text{COP} = 0.45 \times 32.41 = \mathbf{14.58}$ .
- -20°C (299.01 K):  $299.01 / 11.01 = 27.16$ ,  $\text{COP} = 0.45 \times 27.16 = \mathbf{12.22}$ .
- -50°C (301.75 K):  $301.75 / 13.75 = 21.95$ ,  $\text{COP} = 0.45 \times 21.95 = \mathbf{9.88}$ .

#### Heat Pump Power

- 0°C:  $236.45 / 14.58 \approx \mathbf{16.22 \text{ W}}$ .
- -20°C:  $451.41 / 12.22 \approx \mathbf{36.94 \text{ W}}$ .
- -50°C:  $773.84 / 9.88 \approx \mathbf{78.32 \text{ W}}$ .

#### Pump Power

- $\Delta P = 0.0275 \times (380 / 0.012) \times (1000 \times 1.5^2 / 2) \approx 979 \text{ Pa}$ .
- Power =  $0.001527 \times 979 \approx 1.5 \text{ W}$ , assume **10 W**.
- Total:
  - 0°C:  $16.22 + 10 = \mathbf{26.22 \text{ W}}$ .
  - -20°C:  $36.94 + 10 = \mathbf{46.94 \text{ W}}$ .
  - -50°C:  $78.32 + 10 = \mathbf{88.32 \text{ W}}$ .

### Step 6: Floor Surface Temperature

- Range: 100 mm spacing, similar to water-pipe:
  - 0°C:  $\sim 23.3\text{--}23.5^\circ\text{C}$ , avg =  $\mathbf{23.38^\circ\text{C}}$ .
  - -20°C:  $\sim 24.5\text{--}24.8^\circ\text{C}$ , avg =  $\mathbf{24.64^\circ\text{C}}$ .
  - -50°C:  $\sim 26.4\text{--}26.7^\circ\text{C}$ , avg =  $\mathbf{26.53^\circ\text{C}}$ .

### Comparison

#### Heat Output (W)

- Air: 254.98, 486.74, 834.39.
- Water: 254.98, 486.82, 834.55.
- Water-Pipe: 253.57, 484.05, 829.66.
- Floor-Only: 236.45, 451.41, 773.84 (lower due to no floor loss).

#### COP

- Air: 16.71, 15.33, 13.65.
- Water: 17.57, 16.75, 15.64.
- Water-Pipe: 17.62, 16.80, 15.71.
- Floor-Only: 14.58, 12.22, 9.88 (lower due to higher  $T_{\text{water}}$ ).

#### Electrical Input (W)

- Air: 25.26, 41.75, 71.13.
- Water: 24.51, 39.06, 63.36.



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- Water-Pipe: 24.39, 38.81, 62.81.
- Floor-Only: 26.22, 46.94, 88.32 (highest due to lower COP).

#### Surface Temperature (°C)

- Air: 22.15, 22.28, 22.48 (wall/ceiling).
- Water: 22.15, 22.28, 22.48 (wall/ceiling).
- Water-Pipe: 22.0–22.3, 22.1–22.4, 22.3–22.6 (wall/ceiling).
- Floor-Only: 23.3–23.5, 24.5–24.8, 26.4–26.7 (floor).

#### Analysis

- **Floor-Only:** Higher  $T_{\text{water}}$  (24–28.6°C) reduces COP, increasing electrical input. Floor surface (23.38–26.53°C) is comfortable but warmer than wall systems.
- **Efficiency:** Water and Water-Pipe outperform due to lower sink temperatures and higher COP.
- **Practicality:** Floor-only simplifies wall/ceiling design but requires more heat pump power.

Need further refinement or a different floor structure?



Now test cooling. Inside temperature remains 22° C. Test it with 35° C and 45° C outside.

Let's test the cooling performance for the air, water, water-pipe, and floor-only scenarios, maintaining an indoor temperature of 22°C with outdoor temperatures of 35°C and 45°C. The heat pump (R290) now acts as a chiller, extracting heat from the indoor medium (air, water, or pipes) and rejecting it to a 15°C water source (e.g., geothermal or chilled water loop). We'll calculate the cooling load, COP, electrical input, and surface temperatures for each scenario.

#### Step 1: Common Parameters

- **Indoor:** 22°C.
- **Outdoor:** 35°C, 45°C.
- **Wall + Ceiling:** 63 m<sup>2</sup> (25 m<sup>2</sup> wall, 38 m<sup>2</sup> ceiling).
- **Window:** 2.5 m<sup>2</sup>,  $U = 0.8 \text{ W/m}^2\text{-K}$ .
- **Ventilation:** 25 m<sup>3</sup>/h, 85% heat recovery (reversed for cooling).
- **Heat pump:** R290, sink = 15°C (288 K), source = medium's cold side.
- **Floor:** 38 m<sup>2</sup> (floor-only scenario).

#### Cooling Load Components

1. **Window:**  $Q_{\text{win}} = U \times \text{Area} \times (T_{\text{out}} - T_{\text{in}})$ .
2. **Ventilation:**  $Q_{\text{vent}} = \rho \times c \times \text{Volume flow} \times \Delta T \times (1 - \text{efficiency})$ , reversed direction.
3. **Wall + Ceiling:** Varies by scenario (unheated U-value or effective U with cooling).

#### Step 2: Cooling Load Calculations

##### Window

- 35°C:  $0.8 \times 2.5 \times (35 - 22) = 2 \times 13 = \mathbf{26 \text{ W}}$ .
- 45°C:  $2 \times (45 - 22) = 2 \times 23 = \mathbf{46 \text{ W}}$ .

##### Ventilation (Heat Gain)

- Volume =  $0.006944 \text{ m}^3/\text{s}$ ,  $\rho \times c \times V \times (1 - 0.85) = 1.2509$ .
- 35°C:  $1.2509 \times (35 - 22) = 1.2509 \times 13 = \mathbf{16.26 \text{ W}}$ .
- 45°C:  $1.2509 \times (45 - 22) = 1.2509 \times 23 = \mathbf{28.77 \text{ W}}$ .

##### Wall + Ceiling (Base Load, Unheated)

- **Air/Water/Water-Pipe:**  $U = 0.12 \text{ W/m}^2\text{-K}$  (original unheated).
  - 35°C:  $0.12 \times 63 \times (35 - 22) = 7.56 \times 13 = \mathbf{98.28 \text{ W}}$ .
  - 45°C:  $7.56 \times 23 = \mathbf{173.88 \text{ W}}$ .
- **Floor-Only:**  $U = 0.119 \text{ W/m}^2\text{-K}$  (mineral wool gap).
  - 35°C:  $0.119 \times 63 \times 13 = 7.497 \times 13 = \mathbf{97.46 \text{ W}}$ .
  - 45°C:  $7.497 \times 23 = \mathbf{172.43 \text{ W}}$ .



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### Step 3: Scenario-Specific Cooling Loads

#### Air Scenario

- **17 mm air gap**, 1.5 m/s,  $U_{in} = 4.55 \text{ W/m}^2\text{-K}$ ,  $U_{out} = 0.13 \text{ W/m}^2\text{-K}$ .
- $Q_{cool} = Q_{in}$  (heat removed from room).
- Base load: 98.28 W (35°C), 173.88 W (45°C).
- **$T_{gap} < 22^\circ\text{C}$** , increases  $Q_{out}$ :
  - 35°C:  $Q_{in} = 4.55 \times 63 \times (22 - T_{gap})$ ,  $Q_{out} = 0.13 \times 63 \times (35 - T_{gap})$ .
  - $Q_{cool} = Q_{win} + Q_{vent} + Q_{out} = 26 + 16.26 + 8.19 \times (35 - T_{gap})$ .
  - Assume  $T_{gap} = 21^\circ\text{C}$  (trial):
    - $Q_{in} = 4.55 \times 63 \times 1 = 286.65 \text{ W}$ .
    - $Q_{out} = 8.19 \times (35 - 21) = 8.19 \times 14 = 114.66 \text{ W}$ .
    - $Q_{cool} = 26 + 16.26 + 114.66 = \mathbf{156.92 \text{ W}}$ .
    - $Q_{in} = 286.65 \text{ W} > 156.92 \text{ W}$ , adjust  $T_{gap}$  higher.
  - Solve:  $286.65 \times (22 - T_{gap}) = 42.26 + 8.19 \times (35 - T_{gap})$ .
    - $286.65 \times 22 - 286.65 T_{gap} = 42.26 + 286.65 - 8.19 T_{gap}$ .
    - $6306.3 - 286.65 T_{gap} = 328.91 - 8.19 T_{gap}$ .
    - $5977.39 = 278.46 T_{gap}$ ,  $T_{gap} = \mathbf{21.46^\circ\text{C}}$ .
    - $Q_{out} = 8.19 \times (35 - 21.46) = 110.78 \text{ W}$ ,  $Q_{cool} = 42.26 + 110.78 = \mathbf{153.04 \text{ W}}$ .
  - 45°C:  $286.65 \times (22 - T_{gap}) = 74.77 + 8.19 \times (45 - T_{gap})$ .
    - $6306.3 - 286.65 T_{gap} = 443.58 - 8.19 T_{gap}$ .
    - $5862.72 = 278.46 T_{gap}$ ,  $T_{gap} = \mathbf{21.05^\circ\text{C}}$ .
    - $Q_{out} = 8.19 \times (45 - 21.05) = 196.18 \text{ W}$ ,  $Q_{cool} = 74.77 + 196.18 = \mathbf{270.95 \text{ W}}$ .

#### Water Scenario

- **17 mm water gap**, 0.01 m/s,  $U_{in} = 4.45 \text{ W/m}^2\text{-K}$ ,  $U_{out} = 0.129 \text{ W/m}^2\text{-K}$ .
- 35°C:  $280.35 \times (22 - T_{water}) = 42.26 + 8.127 \times (35 - T_{water})$ .
  - $6167.7 - 280.35 T_{water} = 326.69 - 8.127 T_{water}$ .
  - $5841.01 = 272.223 T_{water}$ ,  $T_{water} = \mathbf{21.46^\circ\text{C}}$ .
  - $Q_{cool} = 42.26 + 8.127 \times (35 - 21.46) = 42.26 + 110.15 = \mathbf{152.41 \text{ W}}$ .
- 45°C:  $6167.7 - 280.35 T_{water} = 441.36 - 8.127 T_{water}$ .
  - $5726.34 = 272.223 T_{water}$ ,  $T_{water} = \mathbf{21.04^\circ\text{C}}$ .
  - $Q_{cool} = 74.77 + 8.127 \times (45 - 21.04) = 74.77 + 194.71 = \mathbf{269.48 \text{ W}}$ .

#### Water-Pipe Scenario

- **Pipes in gypsum**,  $U_{in} = 4.5 \text{ W/m}^2\text{-K}$ ,  $U_{out} = 0.129 \text{ W/m}^2\text{-K}$ .
- 35°C:  $283.5 \times (22 - T_{water}) = 42.26 + 8.127 \times (35 - T_{water})$ .
  - $6237 - 283.5 T_{water} = 326.69 - 8.127 T_{water}$ .
  - $5910.31 = 275.373 T_{water}$ ,  $T_{water} = \mathbf{21.46^\circ\text{C}}$ .
  - $Q_{cool} = \mathbf{152.41 \text{ W}}$ .
- 45°C:  $6237 - 283.5 T_{water} = 441.36 - 8.127 T_{water}$ .
  - $5795.64 = 275.373 T_{water}$ ,  $T_{water} = \mathbf{21.04^\circ\text{C}}$ .
  - $Q_{cool} = \mathbf{269.48 \text{ W}}$ .

#### Floor-Only Scenario

- **Floor pipes**, Wall/Ceiling  $U = 0.119 \text{ W/m}^2\text{-K}$ ,  $Q_{cool} = Q_{win} + Q_{vent} + Q_{wall}$ .
- 35°C:  $26 + 16.26 + 97.46 = \mathbf{139.72 \text{ W}}$ .
- 45°C:  $46 + 28.77 + 172.43 = \mathbf{247.20 \text{ W}}$ .

### Step 4: Temperatures and COP

#### Air

- **$T_{cold}$** : 21.46°C (35°C), 21.05°C (45°C).
- **$\Delta T_{air}$** :  $0.255 \text{ m}^3/\text{s}$ ,  $\dot{m} = 0.306 \text{ kg/s}$ ,  $c = 1000 \text{ J/kg}\cdot\text{K}$ .
  - 35°C:  $153.04 / (0.306 \times 1000) = 0.50^\circ\text{C}$ ,  $T_{warm} = 21.96^\circ\text{C}$ .
  - 45°C:  $270.95 / 306 = 0.89^\circ\text{C}$ ,  $T_{warm} = 21.94^\circ\text{C}$ .
- **COP**: Source =  $T_{cold}$ , Sink = 15°C (288 K).
  - 35°C (294.61 K):  $294.61 / (294.61 - 288) = 44.67$ , COP =  $0.45 \times 44.67 = \mathbf{20.10}$ .
  - 45°C (294.20 K):  $294.20 / 6.20 = 47.45$ , COP =  $0.45 \times 47.45 = \mathbf{21.35}$ .

#### Water

- **$\Delta T_{water}$** :  $\dot{m} = 1.7 \text{ kg/s}$ ,  $c = 4186 \text{ J/kg}\cdot\text{K}$ .
  - 35°C:  $152.41 / (1.7 \times 4186) = 0.021^\circ\text{C}$ ,  $T_{cold} = 21.45^\circ\text{C}$ ,  $T_{warm} = 21.47^\circ\text{C}$ .
  - 45°C:  $269.48 / 7116.2 = 0.038^\circ\text{C}$ ,  $T_{cold} = 21.02^\circ\text{C}$ ,  $T_{warm} = 21.06^\circ\text{C}$ .
- **COP**:
  - 35°C (294.60 K):  $44.70$ , COP =  $\mathbf{20.11}$ .
  - 45°C (294.17 K):  $47.53$ , COP =  $\mathbf{21.39}$ .



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#### Water-Pipe

- **$\Delta T_{\text{water}}$ :**  $\dot{m} = 1.527 \text{ kg/s}$ ,  $c = 4186 \text{ J/kg}\cdot\text{K}$ .
  - 35°C:  $152.41 / 6389.62 = 0.024^\circ\text{C}$ ,  $T_{\text{cold}} = 21.45^\circ\text{C}$ ,  $T_{\text{warm}} = 21.47^\circ\text{C}$ .
  - 45°C:  $269.48 / 6389.62 = 0.042^\circ\text{C}$ ,  $T_{\text{cold}} = 21.02^\circ\text{C}$ ,  $T_{\text{warm}} = 21.06^\circ\text{C}$ .
- **COP:** Same as Water (negligible difference).

#### Floor-Only

- **$\Delta T_{\text{water}}$ :**  $\dot{m} = 1.527 \text{ kg/s}$ .
  - 35°C:  $139.72 / 6389.62 = 0.022^\circ\text{C}$ ,  $T_{\text{warm}} = 22.01^\circ\text{C}$ ,  $T_{\text{cold}} = 21.99^\circ\text{C}$ .
  - 45°C:  $247.20 / 6389.62 = 0.039^\circ\text{C}$ ,  $T_{\text{warm}} = 22.02^\circ\text{C}$ ,  $T_{\text{cold}} = 21.98^\circ\text{C}$ .
- **COP:**
  - 35°C (295.14 K): 42.77, COP = **19.25**.
  - 45°C (295.13 K): 42.79, COP = **19.26**.

### Step 5: Electrical Input

- **$W_{\text{elec}} = Q_{\text{cool}} / \text{COP} + \text{Circulation}$ .**
- Circulation: Air = 10 W (fans), Water/Water-Pipe/Floor = 10 W (pump).

#### Air

- 35°C:  $153.04 / 20.10 + 10 = 7.61 + 10 = \mathbf{17.61 \text{ W}}$ .
- 45°C:  $270.95 / 21.35 + 10 = 12.69 + 10 = \mathbf{22.69 \text{ W}}$ .

#### Water

- 35°C:  $152.41 / 20.11 + 10 = 7.58 + 10 = \mathbf{17.58 \text{ W}}$ .
- 45°C:  $269.48 / 21.39 + 10 = 12.60 + 10 = \mathbf{22.60 \text{ W}}$ .

#### Water-Pipe

- 35°C: **17.58 W**.
- 45°C: **22.60 W**.

#### Floor-Only

- 35°C:  $139.72 / 19.25 + 10 = 7.26 + 10 = \mathbf{17.26 \text{ W}}$ .
- 45°C:  $247.20 / 19.26 + 10 = 12.83 + 10 = \mathbf{22.83 \text{ W}}$ .

### Step 6: Surface Temperatures

- **Air:**  $T_{\text{surface}} = 22 - Q_{\text{in}} / (4.55 \times 63)$ .
  - 35°C:  $22 - 153.04 / 286.65 = 22 - 0.53 = \mathbf{21.47^\circ\text{C}}$ .
  - 45°C:  $22 - 270.95 / 286.65 = 22 - 0.95 = \mathbf{21.05^\circ\text{C}}$ .
- **Water:**  $T_{\text{surface}} = 22 - Q_{\text{in}} / (4.45 \times 63)$ .
  - 35°C:  $22 - 152.41 / 280.35 = \mathbf{21.46^\circ\text{C}}$ .
  - 45°C:  $22 - 269.48 / 280.35 = \mathbf{21.04^\circ\text{C}}$ .
- **Water-Pipe:**  $T_{\text{surface}} = 22 - Q_{\text{in}} / (4.5 \times 63)$ .
  - 35°C:  $22 - 152.41 / 283.5 = \mathbf{21.46^\circ\text{C}}$ .
  - 45°C:  $22 - 269.48 / 283.5 = \mathbf{21.05^\circ\text{C}}$ .
- **Floor-Only:**  $T_{\text{surface}} = 22 - Q_{\text{cool}} / (4.5 \times 38)$ .
  - 35°C:  $22 - 139.72 / 171 = 22 - 0.82 = \mathbf{21.18^\circ\text{C}}$ .
  - 45°C:  $22 - 247.20 / 171 = 22 - 1.45 = \mathbf{20.55^\circ\text{C}}$ .

### Comparison

#### Cooling Load (W)

- Air: 153.04, 270.95.
- Water: 152.41, 269.48.
- Water-Pipe: 152.41, 269.48.
- Floor-Only: 139.72, 247.20 (lowest, no wall/ceiling cooling loss).

#### COP

- Air: 20.10, 21.35.
- Water: 20.11, 21.39.
- Water-Pipe: 20.11, 21.39.
- Floor-Only: 19.25, 19.26 (lower due to higher  $T_{\text{water}}$ ).

#### Electrical Input (W)



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- Air: 17.61, 22.69.
- Water: 17.58, 22.60.
- Water-Pipe: 17.58, 22.60.
- Floor-Only: 17.26, 22.83 (best at 35°C, slightly worse at 45°C).

#### Surface Temperature (°C)

- Air: 21.47, 21.05 (wall/ceiling).
- Water: 21.46, 21.04 (wall/ceiling).
- Water-Pipe: 21.46, 21.05 (wall/ceiling).
- Floor-Only: 21.18, 20.55 (floor).

#### Analysis

- **Floor-Only:** Lowest cooling load (no additional wall/ceiling loss), best at 35°C, but higher T<sub>water</sub> reduces COP at 45°C.
- **Water/Water-Pipe:** Nearly identical, efficient cooling, uniform surface temps.
- **Air:** Slightly higher load, comparable efficiency.
- **Surface Temps:** All below 22°C, effective cooling, floor-only slightly cooler due to concentrated load.

The floor-only scenario excels in mild cooling (35°C) but loses efficiency at 45°C due to higher pipe temperatures. Want to adjust cooling parameters or add solar gains?



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