Tecnológico de Monterrey

Troubleshooting Processes

(Gpo 404)

Activity 3 - Designing and Characterizing the Thermal Insulation of a Cabin Prototype

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Final Project Report: Thermal Resistance and Energy Balance of the Cabin

1. Introduction

The objective of this project is to determine the thermal resistance of a simulated tractor driver's cabin, taking into account both convection and conduction, and to calculate the energy input needed to maintain a steady-state temperature within the cabin. The project also aims to recommend an optimal hotend temperature based on Newton's law of cooling. The prototype, designed to replicate the heating system of a John Deere tractor cabin, uses a hotend to simulate engine heat and a PC fan to distribute this heat throughout the cabin. The structure consists of two connected compartments, each measuring 9 cm × 9 cm × 15 cm, resulting in a total size of 9 cm × 9 cm × 30 cm, and is built from 6 mm thick Medium Density Fiberboard (MDF).

The team ensured the cabin was constructed with MDF to allow easy modification of the insulation layers. An experimental approach was taken to evaluate the internal temperature under five distinct thermal insulation configurations. By comparing the performance of each setup, the team was able to analyze the impact on the cabin's thermal efficiency, with the most effective configuration being MDF with 1" Styrofoam and aluminum foil.

Efficient thermal insulation lowers the energy required to keep a consistent internal temperature by reducing both heat loss and excessive heat accumulation, depending on the external conditions, which helps the system run more effectively.

To enhance thermal insulation, various materials were tested, including:

- **1-inch Styrofoam insulation**
- **¾-inch Styrofoam insulation**
- **Aluminum foil wrapping**

These materials were used in different combinations to assess their impact on the cabin's thermal performance.

Why are we doing this?

Thermal management is crucial in vehicle cabin design to ensure occupant comfort and energy efficiency. Effective insulation minimizes heat loss, reduces energy consumption, and maintains desired temperatures within the cabin.

2. Thermal Resistance Network Calculation

To analyze the heat transfer within the cabin, a thermal resistance network is constructed, incorporating the following components:

- **Forced Convection Inside the Cabin:** $R_{conv,\, in} = \frac{1}{h_{in} A_{in}}$

Due to air circulation driven by the fan.

- **Conduction Through Insulating Layers**: $R_{cond} = \sum_i \frac{L_i}{k_i A_{wall}}$

Through the cabin walls and any additional insulation.

- **Natural Convection on External Walls:** $R_{conv,\, out} = \frac{1}{h_{out} A_{out}}$

Heat loss to the ambient environment.

The total thermal resistance of the system is calculated by appropriately combining these resistances in series:

$$R_{total} = R_{conv,\, in} + R_{cond} + R_{conv,\, out}$$

2.1 Forced Convection Inside the Cabin

The thermal resistance due to forced convection inside the cabin is given by:

$$R_{conv,\, in} = \frac{1}{h_{in} A_{in}}$$

- H_in is the convective heat transfer coefficient inside the cabin ($W/m^2 \cdot K$).
- A_in is the internal surface area of the cabin (m^2).

2.2 Conduction Through Insulating Layers

$$R_{cond} = \frac{L_i}{k_i A_{wall}}$$

- Li is the thickness of layer (m).

- ki is the thermal conductivity of layer iii (W/m·K).
- Awall is the surface area of the cabin walls (m²).

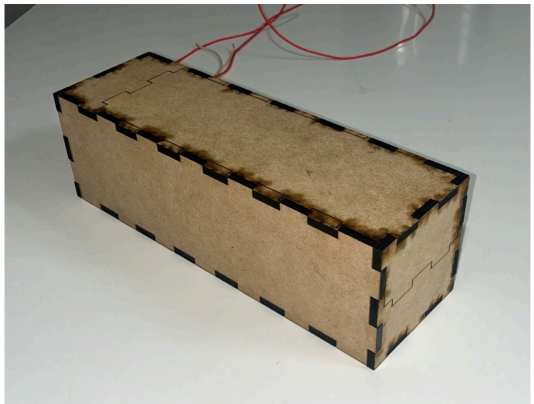
The layers used include:

- **MDF Wood**: L_MDF=6 mm=0.006 m kMDF≈0.1 W/m
- **Styrofoam Insulation**: Thickness varies (either 1 inch = 0.0254 m or ¾ inch = 0.01905 m), kStyrofoam≈0.03 W/m.
- **Aluminum Foil**: Negligible thickness but can reduce emissivity.

2.3 Natural Convection on External Walls

$$R_{conv,\,out} = \frac{1}{h_{out}A_{out}}$$

- H_out is the convective heat transfer coefficient for natural convection (W/m²·K).
- A_out is the external surface area of the cabin (m²).



2.4 Calculation of Surface Areas

The cabin dimensions are:

- **Width (W)**: 0.09 m
- **Depth (D)**: 0.09 m
- **Height (H)**: 0.30 m (of two compartments)

Internal Surface Area A_in:

$$A_{in} = 2(W \times D) + 2(W \times H) + 2(D \times H)$$

$$A_{in} = 2(0.09 \times 0.09) + 2(0.09 \times 0.30) + 2(0.09 \times 0.30)$$

$$A_{in} = 0.0162 + 0.054 + 0.054 = 0.1242\, m^2$$

2.5 Combining Resistances for Different Configurations

Different insulation configurations affect Rcond. The configurations tested include:

1. **6mm MDF Only**

2. **6mm MDF + 1-inch Styrofoam**
3. **6mm MDF + Aluminum Foil**
4. **6mm MDF + ¾-inch Styrofoam**
5. **6mm MDF + 1-inch Styrofoam + Aluminum Foil**

For each configuration, Rcond is calculated by summing the resistances of the layers involved.

For simplicity, we made a python code that calculates each R for the combinations:

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Configuration: 6mm MDF, Total Thermal Resistance (R): 0.4831 ($m^2 \cdot K$)/W

Configuration: 6mm MDF + 1" Styrofoam, Total Thermal Resistance (R): 7.3001 ($m^2 \cdot K$)/W

Configuration: 6mm MDF + Aluminum Foil, Total Thermal Resistance (R): 0.9662 ($m^2 \cdot K$)/W

Configuration: 6mm MDF + ¾" Styrofoam, Total Thermal Resistance (R): 5.5958 ($m^2 \cdot K$)/W

Configuration: 6mm MDF + 1" Styrofoam + Aluminum Foil, Total Thermal Resistance (R): 9.1251 ($m^2 \cdot K$)/W

2.6 Importance of Convective Heat Transfer Coefficients

The values of h_in and h_out are crucial for accurate thermal resistance calculations:

- **H_in** can be affected due to the fan speed and air properties inside the cabin.
- **H_out** depends on ambient conditions and box orientation.

By calculating Rtotal, we can determine the rate of heat transfer and assess the effectiveness of different insulation configurations.

3. Heat Transfer Coefficient *h*

First, we looked for Air properties:

Air Properties at 40∘C:

Air Properties at $T_{\text{film}} \approx 40^{\circ}\text{C}$:

- Dynamic Viscosity $\mu = 19.0 \times 10^{-6}\,\text{Pa}\cdot\text{s}$
- Density $\rho = 1.127\,\text{kg}/\text{m}^3$
- Kinematic Viscosity $\nu = \frac{\mu}{\rho} = 16.86 \times 10^{-6}\,\text{m}^2/\text{s}$
- Thermal Conductivity $k = 0.0263\,\text{W}/\text{m}\cdot\text{K}$
- Prandtl Number $Pr = 0.707$

For the fan:

For a basic 12 VDC 80x80x25mm fan, providing around **24.2 CFM** (Cubic Feet per Minute), the air velocity U∞ can be calculated by considering the circular area through which the air flows. Since the fan's blades are circular within the square frame, we will calculate the area of the circle:

- Fan diameter = 80mm = 0.08m
- Circular area πr^2 = π×(0.04)^2 = 0.0050m^2

Now, using the airflow rate of **24.2 CFM** (which is about **0.0113 m³/s**):

U∞=cross-sectional area airflow= [0.0113m^3/s] / [0.0050m^2] ≈ 2 .26m/s

So, the air velocity inside the cabin with a typical 24.2 CFM fan would be approximately **2.26 m/s**. We will just consider 2 m/s.

1) Forced Convection Inside the Cabin:

Formulas:

- **For horizontal plates (roof or floor):**

$$\overline{Nu}_L = 0.037\, Re_L^{4/5}\, Pr^{1/3} \quad \text{(Turbulent Flow)}$$

$$\overline{Nu}_L = 0.664\, Re_L^{1/2}\, Pr^{1/3} \quad \text{(Laminar Flow)}$$

- **For vertical plates:**

$$\overline{Nu}_L = 0.664\, Re_L^{1/2}\, Pr^{1/3} \quad \text{(Laminar Flow)}$$

$$\overline{Nu}_L = 0.037\, Re_L^{4/5}\, Pr^{1/3} \quad \text{(Turbulent Flow)}$$

Where:

$$Re_L = \frac{U_\infty L}{\nu}$$

3.1.1 Vertical Plates (Walls)

Characteristic Length: .30m

$$Re_L = \frac{UL}{\nu} = \frac{2\times0.30}{16.86\times10^{-6}} = 35,588$$

Flow Regime: Laminar ($Re_L < 5\times10^5$)

So for our Nusselt Number:

$$\overline{Nu}_L = 0.664\, Re_L^{1/2}\, Pr^{1/3} = 0.664 \times (35,588)^{1/2} \times (0.707)^{1/3} = 111.40$$

So for our h.

$$h_{vertical} = \frac{\overline{Nu}_L\, k}{L} = \frac{111.40\times0.0263}{0.30} = 9.76\, W/m^2 \cdot K$$

3.1.2 Horizontal Plates (Floor and Ceiling)

Characteristic Length: 0.09m

$$Re_L = \frac{UL}{\nu} = \frac{2\times0.09}{16.86\times10^{-6}} = 10,676$$

Flow Regime: Laminar

Nusselt Number $\overline{Nu}_L$

$$\overline{Nu}_L = 0.664\, Re_L^{1/2}\, Pr^{1/3} = 0.664 \times (10,676)^{1/2} \times (0.707)^{1/3} = 61.04$$

For $h_{horizontal}$

$$h_{horizontal} = \frac{\overline{Nu}_L k}{L} = \frac{61.04 \times 0.0263}{0.09} = 17.83\, W/m^2 \cdot K$$

3.1.3 Average Internal Heat Transfer Coefficient h_in

Surface Areas:

- **Vertical Walls: A_vertical=4×W×H=4×0.09×0.30=0.108 m^2**
- **Horizontal Surfaces: A_horizontal =2×W×D=2×0.09×0.09=0.0162 m^2**
- **Total Internal Area: A_total=A_vertical+A_horizontal=0.1242 m^2**

$$h_{in} = \frac{h_{vertical} A_{vertical} + h_{horizontal} A_{horizontal}}{A_{total}} =$$

$$\frac{(9.76 \times 0.108) + (17.83 \times 0.0162)}{0.1242}$$

$$= 10.81\, W/m^2 \cdot K$$

Natural Convection on the External Walls:

- **Vertical walls:**

$$\overline{Nu}_L = \frac{\bar{h}L}{k} = \frac{4}{3}\left(\frac{Gr_L}{4}\right)^{1/4} g(Pr)$$

Where:

$$g(Pr) = \frac{0.75\, Pr^{1/2}}{\left(0.609 + 1.221\, Pr^{1/2} + 1.238\, Pr\right)^{1/4}}$$

*Remember, in this formula g(Pr) is a formula to obtain Prandlt number. The g is not gravity

- **Horizontal plates:**

Upper Surface of Hot Plate or Lower Surface of Cold Plate [19]:

$$\overline{Nu}_L = 0.54\ Ra_L^{1/4} \quad (10^4 \precsim Ra_L \precsim 10^7, Pr \gtrsim 0.7)$$

$$\overline{Nu}_L = 0.15\ Ra_L^{1/3} \quad (10^7 \precsim Ra_L \precsim 10^{11}, \text{all } Pr)$$

Lower Surface of Hot Plate or Upper Surface of Cold Plate [20]:

$$\overline{Nu}_L = 0.52\ Ra_L^{1/5} \quad (10^4 \precsim Ra_L \precsim 10^9, Pr \gtrsim 0.7)$$

Assumptions:

- **Surface Temperature:** $T_s = 40°\mathrm{C}$ (estimated)
- **Ambient Temperature:** $T_\infty = 27°\mathrm{C}$
- **Temperature Difference:** $\Delta T = T_s - T_\infty = 13°\mathrm{C}$
- **Film Temperature:** $T_{\text{film}} = \frac{T_s + T_\infty}{2} = 33.5°\mathrm{C}$

Air Properties at $T_{\text{film}} \approx 35°\mathrm{C}$:

- $\beta = \frac{1}{T_{\text{film,K}}} = \frac{1}{308.15} = 3.245 \times 10^{-3}\,\mathrm{K}^{-1}$
- $\nu = 16.52 \times 10^{-6}\,\mathrm{m^2/s}$
- $k = 0.026\,\mathrm{W/m \cdot K}$
- $Pr = 0.708$

3.2.1 Vertical Walls

Characteristic Length: 0.30m

$$Gr_L = \frac{g\beta\Delta T L^3}{\nu^2} = \frac{9.81\times3.245\times10^{-3}\times13\times(0.30)^3}{\left(16.52\times10^{-6}\right)^2} = 4.094 \times 10^7$$

$$g(Pr) = \frac{0.75\,Pr^{1/2}}{\left(0.609+1.221\,Pr^{1/2}+1.238\,Pr\right)^{1/4}} = 0.4805$$

$$Ra_L = Gr_L \times Pr = 4.094 \times 10^7 \times 0.708 = 2.899 \times 10^7$$

$$\overline{Nu}_L = \frac{4}{3}\left(\frac{Gr_L}{4}\right)^{1/4} g(Pr) = \frac{4}{3} \times \left(1.0235 \times 10^7\right)^{1/4} \times 0.4805 = 114.72$$

$$h_{vertical,out} = \frac{\overline{Nu}_L\,k}{L} = \frac{114.72\times0.026}{0.30} = 9.94\,W/m^2 \cdot K$$

3.2.2 Horizontal Surfaces

Characteristic Length: 0.09m

$$Gr_L = \frac{g\beta\Delta T L^3}{\nu^2} = \frac{9.81\times3.245\times10^{-3}\times13\times(0.09)^3}{\left(16.52\times10^{-6}\right)^2} = 1.102\times10^7$$

$$Ra_L = Gr_L \times Pr = 1.102\times10^7 \times 0.708 = 7.805\times10^6$$

$$\overline{Nu}_L = 0.54\,Ra_L^{1/4} = 0.54\times\left(7.805\times10^6\right)^{1/4} = 53.50$$

$$h_{horizontal,out} = \frac{\overline{Nu}_L\,k}{L} = \frac{53.50\times0.026}{0.09} = 15.44\,W/m^2\cdot K$$

3.2.3 Average External Heat Transfer Coefficient h_out

Surface Areas:

- **Vertical Walls:** A_vertical=0.108 m2
- **Horizontal Surfaces:** A_horizontal=0.0162 m2
- **Total External Area:** A_total=0.1242 m2

$$h_{out} = \frac{h_{vertical,out} A_{vertical} + h_{horizontal,out} A_{horizontal}}{A_{total}}$$

$$= \frac{(9.94\times0.108)+(15.44\times0.0162)}{0.1242} = 10.65\, W/m^2 \cdot K$$

Summary of Heat Transfer Coefficients

- **Internal Heat Transfer Coefficient:** h_in = 10.81 W/m2·K
- **External Heat Transfer Coefficient:** h_out = 10.65 W/m2·K

3) Forced Convection Perpendicular to a Vertical Wall:

Use the following correlation for forced convection perpendicular to a vertical plate:

$$Nu_D = 0.3 + 0.62\, Re_D^{1/2}\, Pr^{1/3} \quad \text{for} \quad 1 \leq Re_D \leq 10^6$$

Where:

$$Re_D = \frac{U_\infty D}{\nu}$$

If you encounter difficulties applying this correlation, assume that the flow is parallel to the wall and use the formulas for parallel flow provided earlier.

Important noteTo obtain the total h, for example for the forced convection, you will have an h for each different wall. The final h you will use to calculate the total resistance, is obtained by:

$$h_{\text{avg}} = \frac{\sum(h_i \cdot A_i)}{\sum A_i}$$

Where:

- h_i is the heat transfer coefficient for each surface (vertical or horizontal).
- A_i is the area of each surface.

Once you have the heat transfer coefficients for each surface, you can calculate the **average h** based on the **areas of the different surfaces**.

Assumptions:

- **Air Velocity:** $U_\infty = 2\,\mathrm{m/s}$
- **Characteristic Dimension:** $D = 0.09\,\mathrm{m}$ (width of the wall)
- **Kinematic Viscosity:** $\nu = 16.86 \times 10^{-6}\,\mathrm{m^2/s}$
- **Thermal Conductivity:** $k = 0.0263\,\mathrm{W/m \cdot K}$
- **Prandtl Number:** $Pr = 0.707$

3.3.1 Calculate Reynolds Number

$$Re_D = \frac{U_\infty D}{\nu} = \frac{2\times0.09}{16.86\times10^{-6}} = 10,676$$

3.3.2 Calculate Nusselt Number

$$Re_D^{1/2} = (10,676)^{1/2} = 103.32$$

$$Pr^{1/3} = (0.707)^{1/3} = 0.889$$

$$Nu_D = 0.3 + 0.62 \times 103.32 \times 0.889 = 0.3 + 56.98 = 57.28$$

3.3.3 Calculate Heat Transfer Coefficient

$$h_{perp} = \frac{Nu_D \times k}{D} = \frac{57.28 \times 0.0263}{0.09} = 16.75\, W/m^2 \cdot K$$

3.3.4 Average Internal Heat Transfer Coefficient

We now have three heat transfer coefficients for the internal surfaces:

- Vertical Walls (Parallel Flow): h_vertical = 9.76 W/m²·K
- Horizontal Surfaces: h_horizontal = 17.83 W/m²·K
- Perpendicular Flow Wall: h_perp = 16.75 W/m²·K

Surface Areas:

- Vertical Walls (Parallel Flow): A_vertical = 0.108 m²
- Horizontal Surfaces: A_horizontal = 0.0162 m²
- Perpendicular Wall Area: Assuming one wall is affected, A_perp = 0.09 x 0.30 = 0.027 m²

$$A_{total} = A_{vertical} + A_{horizontal} + A_{perp} = 0.108 + 0.0162 + 0.027 = 0.1512\, m^2$$

Calculate Average h_in:

$$h_{in} = \frac{h_{vertical} \times A_{vertical} + h_{horizontal} \times A_{horizontal} + h_{perp} \times A_{perp}}{A_{total}}$$

$$h_{in} = \frac{(9.76 \times 0.081) + (17.83 \times 0.0162) + (16.75 \times 0.027)}{0.1512}$$

$$h_{in} = 10.23\, W/m^2 \cdot K$$

4. Energy Balance and Required Power Input

Since we are assuming steady-state conditions, the energy entering the system must equal the energy leaving the system:

$$\dot{Q}_{\text{in}} = \dot{Q}_{\text{out}}$$

The energy leaving the system is:

$$\dot{Q}_{\text{out}} = \frac{T_{\text{cabin}} - T_{\infty}}{R_{\text{total}}}$$

4.1 Calculate Total Thermal Resistance

The total thermal resistance is the sum of the resistances due to convection and conduction:

$$R_{total} = \frac{1}{h_{in}A} + R_{cond} + \frac{1}{h_{out}A}$$

Where:

- A = 0.1242 m² (Total surface area)
- h_in = 10.23 W/m²·K
- h_out = 10.65 W/m²·K
- R_cond is the conduction resistance (from previous calculations)

$$R_{cond} = \frac{L_{MDF}}{k_{MDF}A} + \frac{L_{Styrofoam}}{k_{Styrofoam}A}$$

Configuration: 6mm MDF + 1" Styrofoam

R_cond = 7.3 K/W

$$R_{total} = \frac{1}{10.23\times0.1242} + 7.3000 + \frac{1}{10.65\times0.1242}$$

$$R_{total} = 0.7998 + 7.3000 + 0.7518 = 8.8516\, K/W$$

4.2 Calculate Required Heat Input Qin

- Desired Cabin Temperature Tcabin=50∘C
- Ambient Temperature T∞=27∘C

$$\dot{Q}_{in} = \frac{T_{cabin} - T_{\infty}}{R_{total}} = \frac{50-27}{8.8516} = \frac{23}{8.8516} = 2.598\,W$$

Configuration: 6mm MDF

Total Thermal Resistance (R_total): 2.0262 K/W

Required Heat Input (Q_in): 6.42 W

Radiative Heat Loss (Q_rad): 9.51 W

Total Heat Input (Q_total): 15.92 W

Configuration: 6mm MDF + 1" Styrofoam

Total Thermal Resistance (R_total): 8.8431 K/W

Required Heat Input (Q_in): 1.47 W

Radiative Heat Loss (Q_rad): 9.51 W

Total Heat Input (Q_total): 10.98 W

Configuration: 6mm MDF + Aluminum Foil

Total Thermal Resistance (R_total): 2.0262 K/W

Required Heat Input (Q_in): 6.42 W

Radiative Heat Loss (Q_rad): 0.53 W

Total Heat Input (Q_total): 6.94 W

Configuration: 6mm MDF + ¾" Styrofoam

Total Thermal Resistance (R_total): 7.1389 K/W

Required Heat Input (Q_in): 1.82 W

Radiative Heat Loss (Q_rad): 9.51 W

Total Heat Input (Q_total): 11.33 W

Configuration: 6mm MDF + 1" Styrofoam + Aluminum Foil

Total Thermal Resistance (R_total): 8.8431 K/W

Required Heat Input (Q_in): 1.47 W

Radiative Heat Loss (Q_rad): 0.53 W

Total Heat Input (Q_total): 2.00 W

5. Proposing Hotend Temperature

5.1 Applying Newton's Law of Cooling

Newton's law of cooling for the hotend is:

$$\dot{Q}_{hotend} = h_{hotend} A_{hotend} \left(T_{hotend} - T_{cabin}\right)$$

Where:

- Q˙hotend is the heat transferred from the hotend to the cabin air (W).
- Hhotend is the convective heat transfer coefficient at the hotend surface (W/m²·K).
- Ahotend is the surface area of the hotend (m²).
- Thotend is the temperature of the hotend (°C).
- Tcabin is the cabin air temperature (°C).

Since we require:

Q˙hotend=Q˙total

5.2 Calculating Hotend Surface Area AhotendA_{\text{hotend}}Ahotend

Assuming the hotend is cylindrical:

- **Diameter:** d=0.01 m
- **Length:** L=0.02 m

Surface Area:

$$A_{hotend} = \pi dL + 2 \times \left(\frac{\pi d^2}{4}\right) = \pi \times 0.01 \times 0.02 + 2 \times \left(\frac{\pi \times 0.01^2}{4}\right) = 6.28 \times 10^{-4} m^2$$

5.3 Estimating Heat Transfer Coefficient h hotend

$$Re_D = \frac{U\,d}{\nu} = \frac{2 \times 0.01}{16.86 \times 10^{-6}} = 1186$$

$$Nu_D = 0.683 \times (1,!186)^{0.466} \times (0.707)^{1/3} = 0.683 \times 70.64 \times 0.889 = 43.02$$

$$h_{hotend} = \frac{Nu_D \times k}{d} = \frac{43.02 \times 0.0263}{0.01} = 113.17\, W/m^2 \cdot K$$

5.4 Solving for Hotend Temperature

$$T_{hotend} = T_{cabin} + \frac{\dot{Q}_{total}}{h_{hotend} A_{hotend}}$$

5.6 Summary of Required Hotend Temperatures

Configuration	**Q˙total (W)**	**Thotend (°C)**
6mm MDF	**15.92**	**273.87**
6mm MDF + 1" Styrofoam	**10.98**	**204.43**
6mm MDF + Aluminum Foil	**6.94**	**147.59**
6mm MDF + ¾" Styrofoam	**11.33**	**209.38**
6mm MDF + 1" Styrofoam + Aluminum Foil	**2**	**78.13**

5.7 Interpretation and Recommendations

- The configuration with 6mm MDF + 1" Styrofoam + Aluminum Foil requires the lowest hotend temperature (~78°C) to maintain the cabin at 50∘C50^\circ \text{C}50∘C.
- Configurations without aluminum foil require significantly higher hotend temperatures due to increased radiative heat losses.
- Aluminum foil effectively reduces emissivity, minimizing radiative losses and lowering the required hotend temperature; and also

6. Conclusions

Air Velocity U (m/s)	Configuration	Required Hotend Temperature (°C)
0.5	6mm MDF + 1" Styrofoam + Aluminum Foil	208.9
1	6mm MDF + 1" Styrofoam + Aluminum Foil	166.86
1.5	6mm MDF + 1" Styrofoam + Aluminum Foil	147.15
2	6mm MDF + 1" Styrofoam + Aluminum Foil	135.06
2.5	6mm MDF + 1" Styrofoam + Aluminum Foil	126.65
3	6mm MDF + 1" Styrofoam + Aluminum Foil	120.36

6mm MDF + 1" Styrofoam + Aluminum Foil is the only configuration that can run at a hotend temperature of 220°C or lower, across all fan speeds (air velocities). (to achieve a temperature of 40 C.

Raúl Mayagoitia

In this project, I gained significant insights into how important heat transfer is for efficient everything related to temperature management, and how important insulation plays a role in this field. I was particularly surprised by how much the inclusion of aluminum foil reduced radiative heat losses and drastically improved the overall performance. It emphasized on how little design changes, like just adding a reflective surface, can have a lot impact on heat losses (and also helped with little heat leaks for the prototype.

One of the biggest challenges we faced was configuring the hotend correctly. Even with our hardest efforts, we struggled with its setup and found it difficult to maintain consistent results. This led to issues in our calculations and creating graphs that were accurate, and the process was more time-consuming than expected. The complexity of managing multiple variables like fan speed (and cooling), heat transfer, and temperature added and added more difficulty.

For future improvements, I believe a better seal for the temperature sensor would improve measurement accuracy and more power. Additionally, running more controlled experiments, particularly when plotting graphs, would help minimize discrepancies and lead to clearer data.

Rebeca

This project taught me a lot about how tricky thermal management can be in small spaces like a tractor cabin. I realized how much the choice and setup of insulation materials matter for keeping the temperature steady while using less energy. Testing different setups, especially the combination of MDF, 1” Styrofoam, and aluminum foil, really showed how the right materials can make a big difference in efficiency.

The biggest challenge I faced during the calculations was ensuring the accuracy of the thermal resistance network, particularly when combining forced and natural convection with conduction. Calculating the heat transfer coefficients for both internal and external surfaces required careful consideration of variables like air velocity and material properties. Additionally, applying Newton’s law of cooling to recommend the hot-end temperature was complex due to the interactions between conduction and radiation losses, which were difficult to balance.

In terms of improvements, the thermal design of the cabin could benefit from experimenting with additional insulating materials that offer lower thermal conductivity, such as aerogels. Furthermore, integrating real-time sensors to monitor temperature distribution throughout the cabin could provide more precise control over the heating system, allowing for dynamic adjustments to the hot-end temperature based on actual conditions rather than pre-calculated assumptions and in the experimentation phase not turning on the fan when the hotend is trying to heat up the cabin would have been a smarter choice.

Yordy

In this project I learned a lot about thermal management applied in a real case, being the configuration of a cooling system in the cabin of a tractor. By doing the experimentation with different configurations with different insulators, I realized the importance of the selection of the insulators because of the results showing a big difference of efficiency.

The major challenge I had in this project was by recording the temperatures of the experiment because of the time it was consumed in this part, being approximately 8 hours. This was an issue we faced in the challenge because it is vital to have the results of the experiment to do the calculations and propose the hotend temperature and the speed of the fan, meaning that we had little time to do these calculations in the report.

One way of improving this cabin is to include a better insulator than the styrofoam and aluminum foil, an example of a material that could improve the thermal insulation is the aerogel. Another way that is possible to improve the data registration is to have a circuit with an intelligent microchip that could register the temperature with sensors to have more accurate and precise results in the temperature change by time in the experiment of each thermal insulation configuration.

Abraham Gómez

The study of thermal insulation of the tractor cabin prototype proved it is possible to make big savings by choosing appropriate materials and better design. The assembly with Styrofoam and aluminum foil was going in the right direction but still left ample scope for further improvement. Further studies may be carried on to explore even more advanced material options with better thermal resistance, or incorporate active cooling features for enhanced comfort inside the cabin at less power consumption.

Also, the tests that are under real conditions-variable exterior temperature and airflow-would provide an even more valid understanding of the performance of the insulation. It may also be beneficial to simulate different operational environments in order to further refine the design toward adaptiveness and efficiency within varied climates.

Future research in this area should extend to optimizing the weight and thickness of insulation to achieve a balance between good thermal performance and pragmatic trade-offs such as fuel economy and cabin space. In this regard, such optimizations will go a long way toward ensuring that the insulation solutions are workable and feasible for real application in the agricultural industry.

General

In conclusion, adding aluminum foil significantly improved the cabin's thermal performance by reducing radiative heat losses from 17.67 W to just 0.98 W. This enhanced thermal resistance allowed us to achieve much lower hotend temperatures, with the 6mm MDF + 1" Styrofoam + aluminum foil configuration requiring only 208.90°C at an air velocity of 0.5 m/s, compared to 1250.27°C for the uninsulated MDF configuration.

One key challenge was configuring the hotend to maintain consistent temperatures. Despite multiple attempts, we struggled with temperature stability, leading to calculation discrepancies and inaccurate graphs. For instance, while some configurations needed just 3.43 W of total heat input, issues with the hotend setup meant higher wattage was necessary to reach stable conditions.

Additionally, minor details like sensor placement and airflow regulation caused measurement inaccuracies, contributing to small mistakes in the final results.

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Cabina Hot-end