Trinity College Dublin
Coláiste na Tríonóide, Baile Átha Cliath
The University of Dublin

RadiAir

RadiAir: Saving Space and Winning Back Time in Modern Urban Living

MEP55B03 Project 1: Design and Development of Novel Heat Exchangers

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Executive Summary

RadiAir intersects multi-billion-dollar market shares of central heating radiation, low-energy indoor clothes drying solutions, and multifunction space-saving furniture. The solution is designed to be manufactured at an inexpensive unit cost and a forecast sale price of €120 - €180 with a market size of 270,000 households in Ireland alone, but with global markets in North America, Asia, and continental Europe.

The design features simple geometries which leverage natural convection heat transfer flow regimes to induce variable performance figures without the need for complex controls. The solution offers the user the ability to operate at a traditional 1,000W ambient heating mode, or a 2,000W drying mode with little effort and no need for additional equipment. The system requires no maintenance and can be retrofitted to existing central heating systems with the opportunity of secondary markets through accessory design and sales.

Declaration

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1. Proposed Technical Approach

1.1. Problem Statement and Target Market

Countries where central heating is prevalent such as Europe, Northern America, and Asia, the climate is the leading influence creating a demand for central heating radiators. In addition, the small living areas of densely populated cities, such as those in the EU and many Asian countries, are tasked with finding compact and energy efficient means of undergoing basic daily tasks, a key one being washing and drying clothes, with the traditional washer-dryer combo often consuming a lot of power and still requiring final drying post-fact. According to a study conducted in the University of Southampton [1], the amount of time spent by individuals on laundry (including drying and hanging of clothes) has remained relatively consistent, typically taking between 2-5% of available hours in the day, while consuming considerable amounts of energy (where tumble dryers are among the largest energy consumers in the household [2]).

This has led to the creation of novel technologies such as portable heated/aired clothes dryers. The concept proposed in this document aims to provide an integrated solution to central heating and a compact, unobstructive, and aesthetic solution to clothes drying.

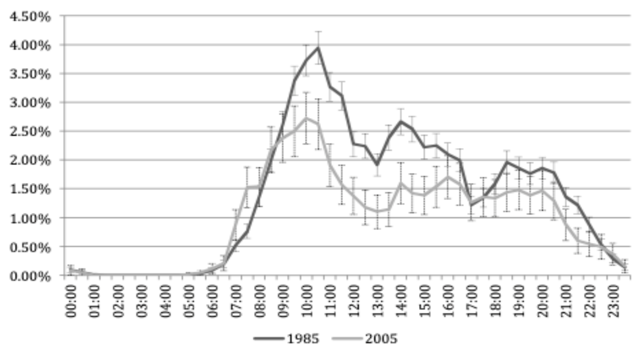


Figure 1: Time Spent During the Day on Laundry [1]

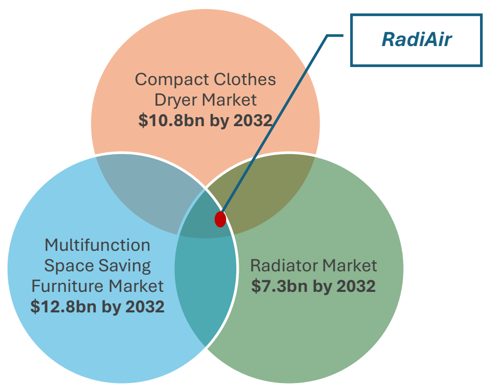


Figure 2: RadiAir Market Fit [3], [4], [5]

The *RadiAir* solution, displayed in Figure 3, intersects growing markets for compact clothes dryers, water-based radiators, and multifunctional furniture perfectly, while offering an alternative to purchasing multiple large appliances that take up large amounts of space, are generally unsightly, and consume electricity than necessary, while wasting user's time in assembly and disassembly or extended drying periods.

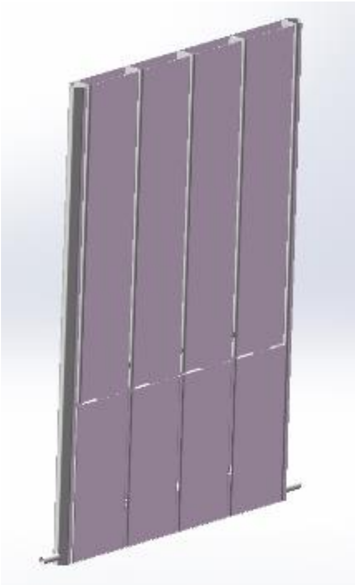
Figure 3 : RadiAir Solution In-house Renderings

This solution leverages the simple phenomenon of natural convection heat transfer (as is utilized by traditional radiator design), however, by incorporating an adjustable geometry, varying heat transfer capacities can be induced *without* requiring additional energy consumption or separate structures. RadiAir has also been specifically designed to minimize the need for additional time spent on tasks such as assembly and disassembly of separate drying equipment and hanging of clothes post-drying.

The solution can be retrofitted to existing central heating systems and an excellent choice for new residential and commercial developments (such as apartments, student accommodation, homes, assisted living facilities, gyms, hotels, lodges, laundromats, hospitals and clinics, cruise ships, fire stations, and any market with a focus on energy efficiency and sustainability).

1.2. Technology and Design Overview

The design's total height is 1800 mm, and its width is 1200 mm. When folded, its thickness is 70mm when mounted but reaches 700 mm (including the drying rods and top panel) when extended. The design consists of four main parts: the fin radiator, the panel, the drying rods, and the outer frame. In the folded position, a cover panel is placed outside the radiator, preventing fin damage, ensuring user safety, and maintaining a clean aesthetic. When extended, the panel slides along two parallel grooves to reach the top. The groove angles are specially designed so that the panel rotates from a vertical to a horizontal position as it is manually moved. Once the panel reaches the top, three rods support it to prevent it from sliding down. These rods also provide multiple slots at different heights, allowing users to hang clothes and trousers.

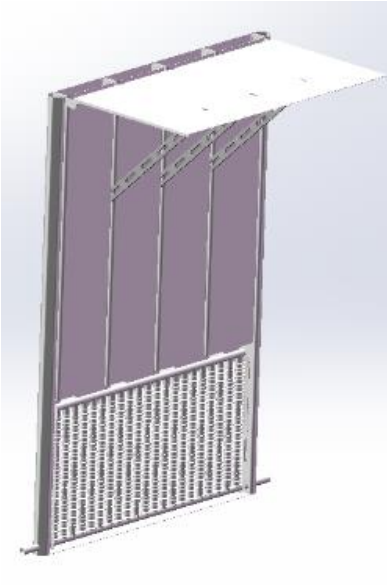


Orientation: 0°

Radiator Only

Water Flow: 3lpm

Power Output ~1000W



Orientation: 0°

Radiator and Drying

Water Flow: 3lpm

Power Output ~1000W

Orientation: 45°

Radiator and Drying

Water Flow: 3lpm

Power Output ~2000W

The fin radiator can rotate between horizontal and vertical positions and is connected to the outer frame through sliding slots. Inside the sliding track are 10 sets of indents, which enable the radiator to be fixed at 10 different angles when drying clothes. However, they do not have a heating function to prevent overheating and damage to clothes. Instead, they rely on heat convection from the radiator to warm and dry the clothes.

To ensure proper airflow, a row of ventilation slots is designed at the corresponding position on the frame. This design allows the radiator to maintain ambient heating performance even when in a vertical position. The design features extruded multiport tubing acting as the radiator's fins, which have been optimally spaced to ensure maximum power delivery (more than 2,000W) when orientated in the "drying" configuration but still capable of satisfying the 1,000W ambient heating requirements of traditional radiators when stowed away. As the volume and temperature variation of the incoming water is consistent in either configuration, the total energy usage will be equivalent to a standard convection radiator in either of the configurations.

1.3. Performance

RadiAir was designed to satisfy 1,000W output for ambient heating requirements and closer to 2,000W for drying configuration (refer Appendix A and Appendix C for details regarding this specification and simplifying methods). The following were the assumptions taken while designing the radiator.

Table 1: Analytical Design Assumptions

Conditions	***Values***	***Unit***
Power (Q)	1,000 – 2,000	W
Inlet Temp (T_{in})	75	°C
OutletTemp (T_{out})	65	°C
Ambient Temp (T_{inf})	20	°C
Delta Pressure	20	kPa
Average Temp (T_{avg})	70	°C
Parameters (Air)	***Values***	***Unit***
Heat Carrying Capacity (C_p)	1007.294	J/kg·K
Density (ρ)	1.09354	kg/m^3
Volumetric coefficient of thermal expansion (β)	0.00312	K^{-1}
Dynamic Viscosity (μ)	0.00002	Pa·s
Thermal Conductivity (k)	0.02807	W/m
A^{ccn} due to Gravity (g)	9.81	m/s^2

1.3.1. Analytical Estimates

Radiator fins were designed and optimized by referring to work by Bar-Cohen *et.al.*[6]. By studying various dimensions for fins, it was finalized to use fin dimensions of 50 mm X 600 mm X 2.4 mm. The geometric parameter **P** was estimated to calculate optimum spacing between fins and along with modified channel number (**Ra'**) with Nusselt number (**Nu**) was calculated to get heat transfer coefficient (**h**).

$$\boldsymbol{P} = \frac{C_p \rho^2 g \beta \Delta T}{\mu k L} \text{ and } \textbf{Optimum spacing } (\boldsymbol{b_{opt}}) = \frac{2.714}{\boldsymbol{P}^{1/4}} \quad (1)$$

Where L is the length of the fin, ΔT is the temperature difference which was used to calculate optimum spacing between 2 consecutive fins. The estimated optimum spacing was 9.93mm.

As per Cohen *et.al.* [6] **Ra'** and **Nu** are calculated as follows:

$$\boldsymbol{Ra'} = \frac{C_p \rho^2 g \beta \Delta T}{\mu k L} * b_{opt} \tag{2}$$

$$\boldsymbol{Nu} = \frac{576}{Ra'^2} + \left(\frac{2.873}{\sqrt{Ra'}}\right)^{-0.5} \tag{3}$$

As per Robinson [7]

$$\boldsymbol{h} = \frac{Nu * k}{b_{opt}} \tag{4}$$

Obtained values for **Ra'** and **Nu** were 54.37 and 1.309, which are closer to 54.3 and 1.31, respectively which were achieved verifying basic calculations. With this estimated number of fins were 96 and estimated power was 1,242.58W. Similar calculations were done for "drying" orientation and with same dimensions for fins with optimum spacing of 5.82mm estimated power was 2,116.73 W. To make sure we satisfy both scenarios it was finalized to keep same dimensions for fins but **change spacing to 9mm** and **reduce number of fins** to **86**. For the fluid channel, Aluminum Multiport Tubes were used as fins which were available in same dimensions and Square Aluminium Tube with Central Rib of 50mm X 50mm X 1000mm.

1.3.2. Computer-Aided Verification

Basic simulations were performed on SolidWorks Flow Simulation Software to estimate the performance[1]. Two basic tests were performed to plot characteristics curves for temperature vs power generated and flow rate vs pressure drop and following were the results. Average error between analytical and CFD result for temperature vs power was around 5%.

[1] Note that a "horizontal" geometry was also tested but the available power from this arrangement exceeded 2,500W which would have increased the temperature drop of the flowing water and therefore is not considered further. However, smaller variants of the design could make use of the horizontal orientation to promote higher convective heat transfer.

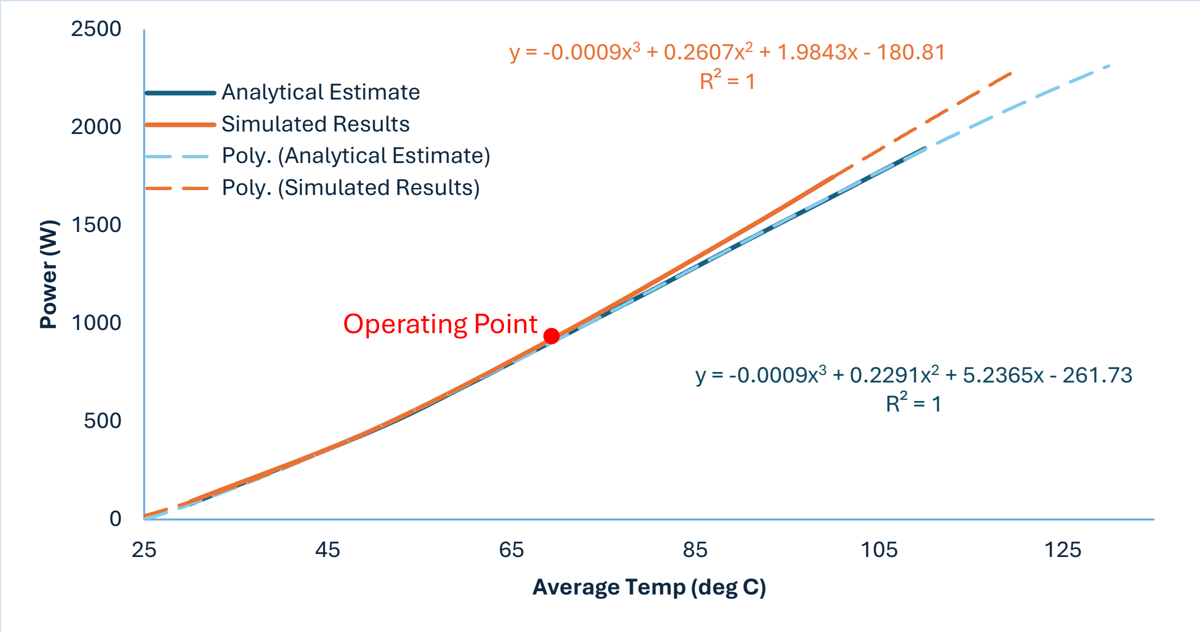


Figure 4: Temperature vs Power Output. Analytical vs Simulations for 0deg Inclination. Mesh Cells ~520,000.

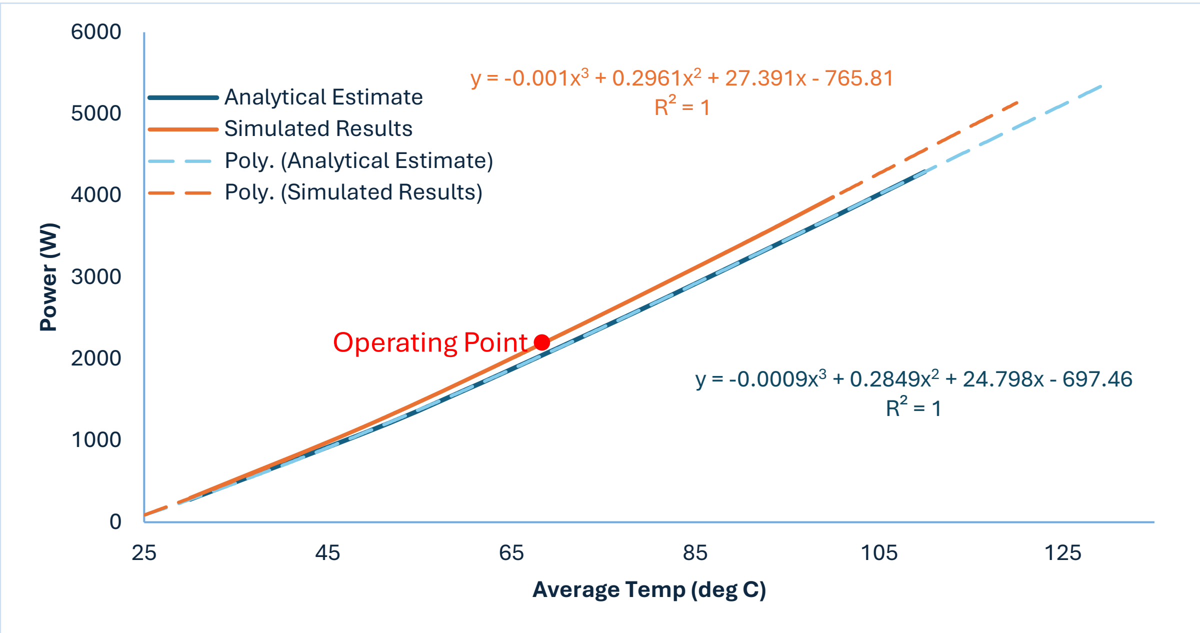


Figure 5: Temperature vs Power Output. Analytical vs Simulations for 45deg Inclination. Mesh Cells ~620,000.

To assess repeatability of the simulated results, a mesh independence study was conducted to verify the accuracy of the simulation by placing the radiator at 45° inclination.

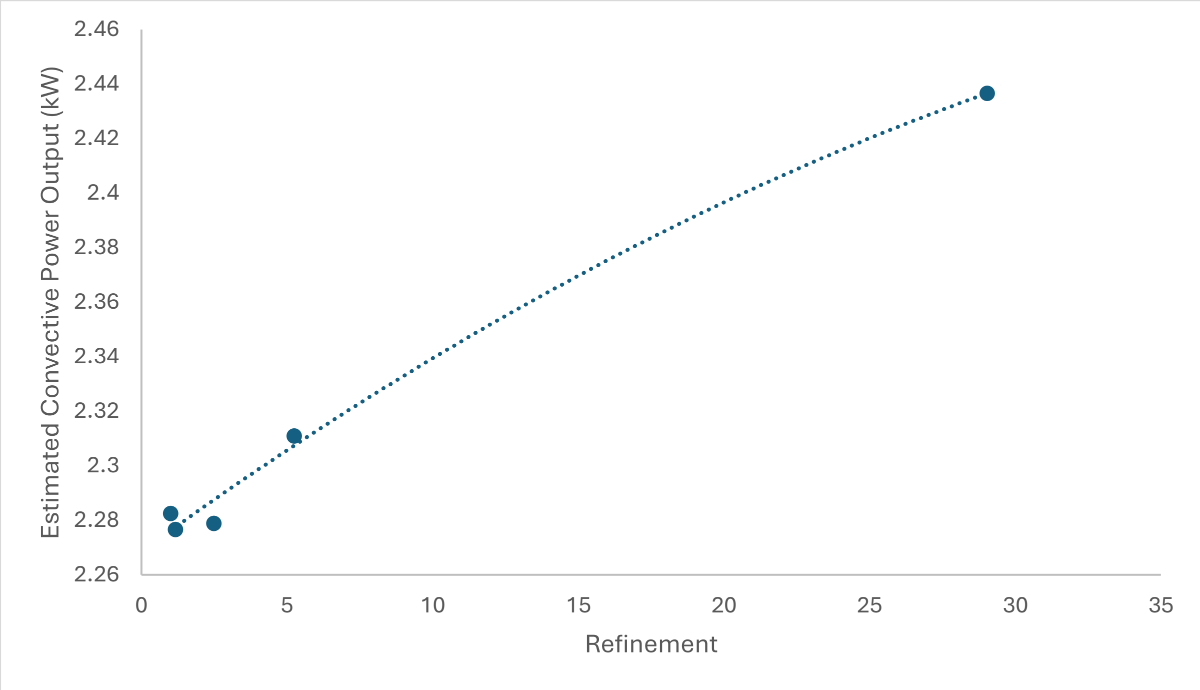


Figure 6: Results from a Mesh Refinement Study

It was seen that for a number of cells up to approximately 630,000 cells, results remain fairly consistent. However, as the mesh is further refined, the estimated power increases (albeit asymptotically). This may be because of increasing numerical error, and refining further will just increase computational time. Total Mesh count was finalized to 639,642 cells. This was deemed acceptable for this phase because 1) this is likely an underestimate of the performance and 2) this still does not include heat transfer via radiation. Note too that the hydraulic loss simulations were also undertaken and results included in Appendix D but are well below the 20kPa threshold.

1.3.3. Variable Operation

As demonstrated, the radiator design can perform at two different set-points without the need for complicated controls or variations in the flow rate or temperature differences in the supply stream. The design outperforms initial estimates and meets targets without considering radiation yet therefore suggesting only further improvements. It is noted that Robinson [8] predicts losses to environment from a system like this, which have not been included here. However, as the design includes a frame assembly (Appendix E) these losses can be minimized with proper coating and materials selections without incurring additional costs.

2. Manufacturing Design

2.1. Materials and Manufacturing

The materials chosen for use is aligned with those typically associated with mass-production and inexpensive radiator design which includes a combination of mild steel and aluminium alloy. The target is to provide a design which offers good heat transfer capabilities as a priority but does so using materials and manufacturing methods which are inexpensive and durable.

According to data from the Changjiang Nonferrous Metals Market and other global suppliers [9], [10], the 2025 bulk order price for Q235 carbon steel (orders over 1 ton) is approximately €0.4 per kg. For 1.5mm thick sheets, this equates to €4.8 per square meter (excluding shipping costs). Multiport aluminium tubing costs are $2.5 - 5 per kg [10], [11] but this does not include the impact of business-relationship negotiations which we would hope to leverage in future. We used a baseline price of $3.7 per kg (approximately €3.5 per kg) for calculations. Each tubing segment weighs 0.037 kg, resulting in a cost of €0.13 per piece. Using the same methodology, the aluminium square tube (with a central rib) costs €2.2 per 1,000mm.

Through manufacturer consultations on Chinese trading platforms, we determined that laser welding of aluminium profiles costs €0.6 per weld point. This value will be used for BOM table calculations. Finally, we selected LCL (Less than Container Load) sea freight due to its cost-effectiveness for partial-container shipments. With an estimated rate of €45 per cubic meter and a unit volume of 0.2m^3 the per-unit freight cost is calculated as 0.2 × 45 = €9.

2.2. Bill of Materials

A preliminary (and conservative) bill of materials was prepared per single unit as in Table 2. The initially estimated production cost is therefore €102.55. A sample of the relevant manufacturing drawings are contained within Appendix E.

Table 2: *RadiAir* Bill of Materials

Serial #	Part Name	Material	Qty	Unit Price(€)	Notes
1	Main Frame	0.15mm Q235 carbon steel	1	19.7	4.10m^2
2	Plate Cover	0.15mm Q235 carbon steel	1	3.2	0.67m^2
3	Connecting Rod	0.15mm Q235 carbon steel	3	0.3	0.023m^2
4	Connecting Rod	0.15mm Q235 carbon steel	2	0.9	0.007m^2
5	M10×20 Hex Bolt	Carbon steel	19	0.02	-

Serial #	Part Name	Material	Qty	Unit Price(€)	Notes
6	M10 lock Nut	Carbon steel	19	0.01	-
7	M10×60 Expansion Bolt	Galvanized steel	8	0.2	-
8	Swivel Faucet Adapter	Brass	2	1.5	-
9	Aluminum Multiport Tube	6063 Aluminum	86	0.13	600mm
10	Aluminum Square Tube with Central Rib	6063 Aluminum	1	2.2	1000mm
11	Shipping	\	\	9	-
12	Cutting & Bending	\	\	15	-
13	Welding	\	\	34.4	-
	Total price			**102.55**	

2.3. Commercials and Pricing

The system is designed for simple mass production using common and available materials and components. As discussed, we anticipate a total landed unit cost of <€103 (inclusive of materials, manufacturing, packaging and shipping). Based on the available alternatives (i.e., the cost of purchasing both a central-heating radiator and clothes drying system [12], [13]) and to retail between €120 and €180 (which will undercut the purchase of stand-alone radiator and dryer purchases). This price does not include the "green premium" developers may choose to impart on the product in place of energy intensive conventional dryers [1].

Another configuration dedicated to the drying of clothes (for example, during summer time when heating of the room may not be ideal) is also available by an optional cover to be purchased for the unit as shown in Figure 7 alongside. This will be marketed separately as an accessory to the basic unit should customers require. There are also options for additional hanger accessories and customisations such as colour-ways for promotion of a secondary market.

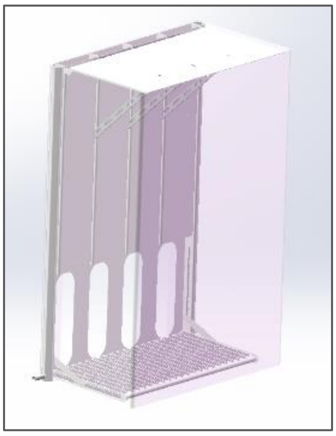


Figure 7: *RadiAir* with Drying Cover Included

3. Technology Outlook

So long as cool, wet, urban and rural climates exist, the demand for central heating and a means for reliably washing and drying one's clothing will remain relevant. *RadiAir* provides an elegant alternative to bulky, expensive, and time-wasting clothes drying equipment and therefore intersects a +$10bn market for radiators, multifunction furniture, and clothes drying equipment. Customers will feature individual consumers but also the hospitality industry, residential developers, and industrial customers.

Investment into the product will allow for final prototyping and testing; obtaining relevant compliance to product standards such as BS EN 44-1:2014; and further negotiating with suppliers and factories for a reduced cost per unit produced.

The design as presented herein is the minimum potential performance for such a device, with additional research into the impact of alternative surface finishes and colours promoting radiation heat transfer and therefore allowing for more compact alternatives will serve to further broaden the market and offer the opportunities for *RadiAir* to be incorporated into countless homes and businesses.

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APPENDECIES

A. DESIGN BRIEF AND MINIMUM FUNCTIONAL SPECIFICATION

Basic Concept Outline and Design Objectives

The group aimed to provide a proposal for a novel heat exchanger, suitable for central heating systems. The designs must match or exceed standard designs but incorporate an aesthetic or dual-purpose function by making use of new designs, technology, materials or approaches to suite new or disrupt existing markets.

Minimum Functional Specification

The proposed design must:

- Have a maximum thermal output of 1,000W for central heating purposes.
- Consider an inlet water temperature of 75°C and ambient room temperature of 20°C.
- Have total pressure loss in the water circuit not exceeding 20kPa.
- Be rated for a 600kPa operating pressure. BS EN 44-1:2014 requires a design pressure of 1.69MOP therefore design for 1,014kPa [14].

Design Restrictions

The proposed design may not consider methods of "unproven" technologies not in commercial use. Nor may the design utilise any form of mechanically forced convection. The designs should be suitable for domestic use (which extends to a restriction on the use of toxic or flammable fluids/materials). The design should avoid the requirement for additional maintenance beyond what is expected for conventional designs. Finally, the design should be easily manufactured and scalable for the intended market.

B. CONCEPTS AND DOWN SELECTION

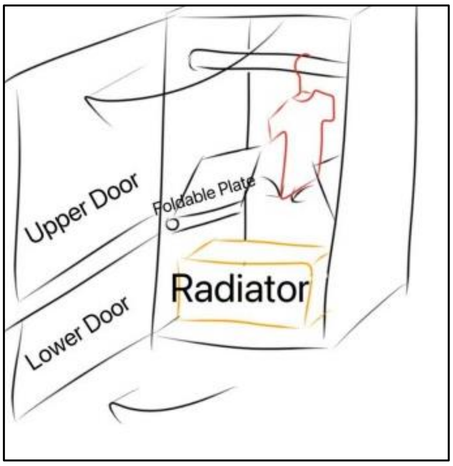
Focusing on the design brief and minimum functional specification, a concept study and down-selection process was undertaken. The group devised three concepts which would meet the brief and which the group believed provided a marketable solution.

Design Concept 1 (DC001):

A radiator-integrated wall divider/ large-scale window blind. The concept focuses on incorporating the central heating radiation system as a feature of a room or building. By leveraging the idea that heated fins (when optimally spaced) can induce strong natural convective currents and promote heat transfer, one could design a feature wall/ window blind which offered the necessary heating but also providing a function. The market here would be commercial infrastructure such as offices, convention centres, and hotels, with smaller designs being relevant for developers in apartment building construction, for example. The idea included the ability to provide a dual-sided marketing canvas on either side of the heated fins as another income stream for the purchaser, or simply to customise the design.

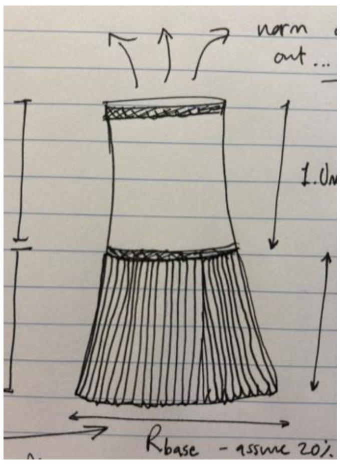
Design Concept 2 (DC002):

A clothes dryer/ radiator design. The original concept included an integrated radiator/closet which could be manipulated to allow the unit to function either as a normal radiator, or a clothes dryer, or a combination of both. The design was marketable and viable in highly populated cities originally. The original concept focused on manipulating the *housing* to facilitated directional heat transfer either into the room or onto the clothes and therefore did not consider alternative power outputs from the 1,000W specified. Manufacturing would be simple and use largely existing technologies and methods to achieve the desired function.



Design Concept 3 (DC003):

A floor lamp/ radiator design. The concept included a large (approx. 1.7m) floor lamp with integrated radiator which served to remove the unsightly conventional radiators from a room but also incorporate the stacking effect through properly shaping the lamp and therefore improving the convective currents caused by the radiator to improve efficiencies. The design would be simple to manufacture but would appeal to a smaller market as it was more of a design/ statement item than it is practical.



Each member of the team then scored the concepts, and an average score was used to determine which concept should move to detail design. The concepts were scored out of four (4) based on the following weighted questions:

1. Is the design capable of producing 1,000W (weighted 15%).
2. Is the design likely capable of a maximum pressure drop of 20kPa (weighted 15%)
3. Is there a market for the design (weighted 15%)
4. Is the concept easy to manufacture using known techniques (weighted 15%)
5. Is there sufficient time to present a concept within the allocated time (weighted 20%)
6. Is there form/function novelty (weighted 20%)

The results of the scoring process are summarised in Table 3 below. Here, a low score is represented by a one (1) and a high score by a four (4).

Table 3: Summary of Concept Scoring

Category	Weight	DC001	DC002	DC003
1000W Capable	15%	4.0	4.0	4.0
Max 20kPa dP	15%	3.0	3.0	3.0
Market	15%	2.6	3.8	2.6
Manufacturability	15%	3.0	3.8	3.0
Sufficient time to design	20%	4.0	3.8	3.8
Form/function novelty	20%	3.4	3.8	3.8
Total Score		3.37	**3.71**	3.41

Design concept 2 (clothes dryer/ radiator) was chosen as the concept for detail design. The team now commenced with preliminary calculations and estimations to understand what the necessary restraints or opportunities were present with the design.

C. PRELIMINARY DESIGN CONSIDERATIONS

Thermal Network:

Similar to work presented by Robinson [8] which modelled the heat transfer mechanisms from a wall-mounted radiator, a thermal network model may be represented by Figure 8. The problem boundary conditions for the solution are highlighted in red. However, it is noted that specific considerations into the materials and manufacturing have been considered in the final design to minimize the losses (Q_{loss}) of the system.

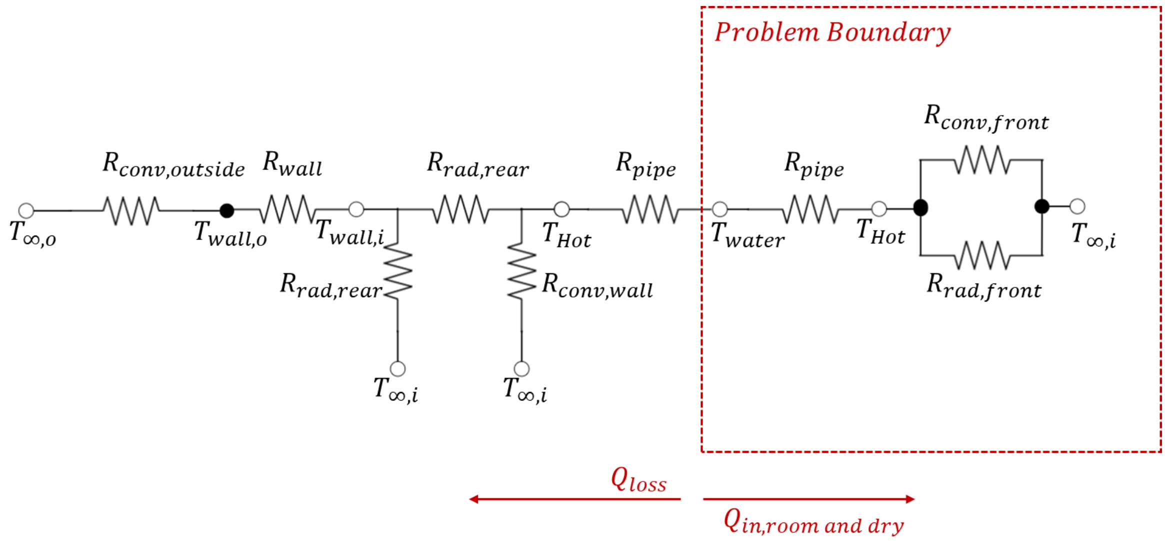


Figure 8: Thermal Network Diagram

The thermal network model is a useful staring point in determining the necessary design parameters for the system. In essence, the thermal resistances within the network should be minimised to ensure that the maximum heat available from the incoming fluid (i.e., warm water into the radiator) is transferred to the surroundings (be it the room or the clothing).

Design Simplification:

It is first noted that the resistivity of the water is not considered in the thermal network diagram. As the water travels through the annulus, the centre point of the channel will have the largest temperature with a gradual reduction towards the annulus' inner edge. This thermal resistivity is estimated as $R_{water} = (h_{water} \cdot 2\pi r_{in} L)^{-1}$ which, for a laminar flow regime would result in a

Nusselt number of 4.36 [15] and a heat transfer coefficient of $h_{water} = 5.75 \cdot r_{in}^{-1}$. Therefore, $R_{water} \sim 0.028 \cdot L^{-1}$. This is sufficiently small to be disregarded in this preliminary design.

Following on from this assumption, the contributions of R_{pipe}, $R_{conv,front}$, and $R_{rad,front}$ are estimated. Assuming a tubular design (understanding that the pipe carrying the water may be of a different profile in the final design) the overall thermal resistivity may be presented as follows:

$$R_{pipe} = \ln\left(r_{out}/r_{in}\right) \cdot \left(2\pi k_{pipe} L\right)^{-1} \quad (5)$$

$$R_{conv,front} = \left(h_{air} \cdot 2\pi r_{out} L\right)^{-1} \quad (6)$$

$$R_{rad,front} = \left(\varepsilon\sigma\left(T_{Hot}^2 + T_{\infty,i}^2\right)\left(T_{Hot} + T_{\infty,i}\right) \cdot 2\pi r_{out} L\right)^{-1} \quad (7)$$

$$Let\ R_{equiv} = \left(R_{conv,front}\right)^{-1} + \left(R_{rad,front}\right)^{-1} \quad (8)$$

Assuming now that the unit comprises of a unit length ($L = 1m$) and $k_{pipe} = 237W/m^2 \cdot K$. Further, that the surface temperature $T_{Hot} = 70°C$ and $T_{\infty,i} = 20°C$. Finally, for a pipe diameter of $25.4mm$ and wall thickness of $1.6mm$ these thermal resistivities may be initially estimated.

$R_{pipe} \sim 9 \times 10^{-5} K/W$

$R_{conv,front} = 1.25 \times 10^{-2} \cdot (h_{air})^{-1}$

$R_{rad,front} = 1.7 \times 10^{-3} \varepsilon^{-1}$

$\therefore 1.7 \times 10^{-1} \lesssim R_{rad,front} \lesssim 1.9 \times 10^{-3}$ for $0.01 < \varepsilon < 0.9$

Unless specifically treated (such as anodization) the radiation from an aluminium tube will be considered a small component for now with $\varepsilon \sim 0.1$.

$\therefore R_{equiv} \approx (1.25 \times 10^{-2} \cdot (h_{air})^{-1})^{-1} + (1.7 \times 10^{-2})^{-1}$

$R_{equiv} \approx (80h_{air} + 58.6) K/W$

Now it can be seen that $R_{equiv} \gg R_{water} \gg R_{pipe}$. Furthermore, for $h_{air} > 2W/m^2 \cdot K$ the resistance due to convection will exceed almost three times that of radiation. For the purposes of initial estimates, therefore, only $R_{conv,front}$ will be considered as the highest contributor to thermal transmissibility. This assumption will be verified in the detail design stage.

Heat transfer coefficient, areas, and pressure drops:

To further refine the problem boundary, the team developed approximations for what the required surface area would need to be for a range of heat transfer coefficients likely to be seen in applications using natural convection. The total available heat was defined based on the parameters for incoming water flow. The incoming water temperature was specified in the brief as $75°C$ with an industry-standard temperature delta and mass flow of $10°C$ and $2-3\ litres\ per\ minute\ (lpm)$, respectively. The available heat transfer is defined by Equation (9). Which is determined to be approximately $2{,}090W$. This is far higher than the basic $1{,}000W$ and therefore indicated room for improvement.

$$\dot{Q}_{available} = \dot{m}Cp\Delta T \quad (9)$$
$$\dot{Q}_{available} = \left(\frac{(3l/min)\cdot(997.43kg/m^3)}{(1000l/m^3)(60s/min)}\right)(4190J/kgK)(10K)$$
$$\dot{Q}_{available} \approx 2{,}090W$$

As a result, an opportunity was taken to expand the design brief. Described simply in Section 1.2, the brief was expanded from the original minimum functional specification per Appendix A to now include <u>two</u> distinct heating scenarios.

1. A conventional radiator function capable of 1,000W net power output.
2. The drying function which the team elected to be an additional 300-1,000W [13] power output which would allow for continuous ambient heating as well as the drying function.

To do this, the team focused on geometries which would allow for a variable $R_{conv,front}$ value (refer Figure 8) while maintaining the same overall surface area (recalling that the aim was to also minimize any additional components in the system).

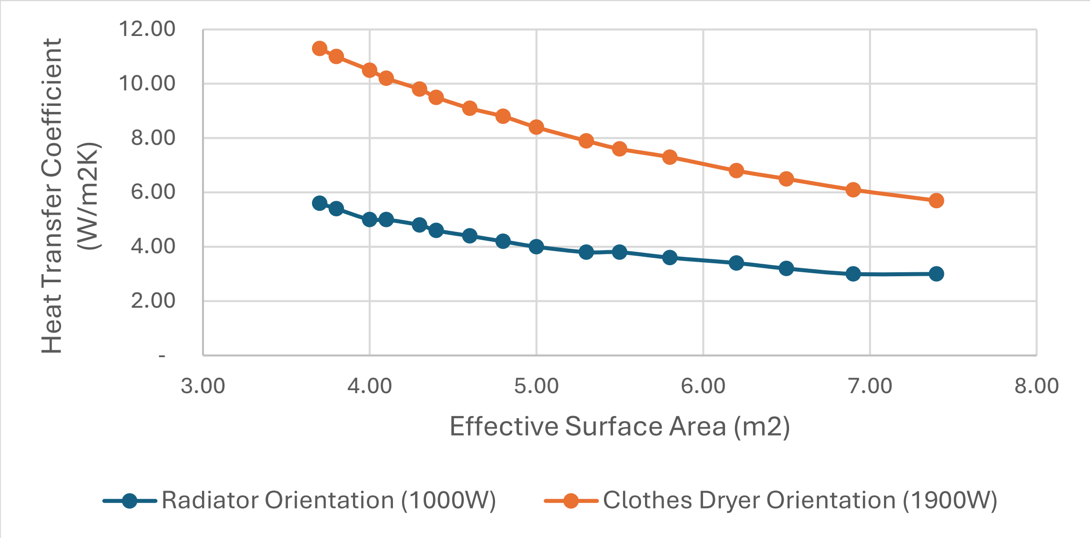


Figure 9: Area vs Heat Transfer Coefficient for Different Configurations

Figure 9 shows the impact of required surface area and heat transfer coefficients required for the design. It is clear from this that an effective doubling of the heat transfer coefficient would be required for the same surface area. The team estimated that a surface area of $6 - 7m^2$ is ideal as the difference in required heat transfer coefficient is minimized at these larger areas (mainly because the 1,000W output target becomes relatively simple to achieve.

As for the pressure drop, the team performed a rudimentary estimate of the drop for an aluminium tube of average surface roughness with the maximum fluid flow (i.e., assuming all radiator flow passed through a single tube). From this, a clear consideration would remain throughout the design process: if the tube diameter decreased (to increase surface area) the mass flow would need to be equivalently decreased through the tubes. However, at this point it was determined that the maximum pressure drop would not be a limiting design factor.

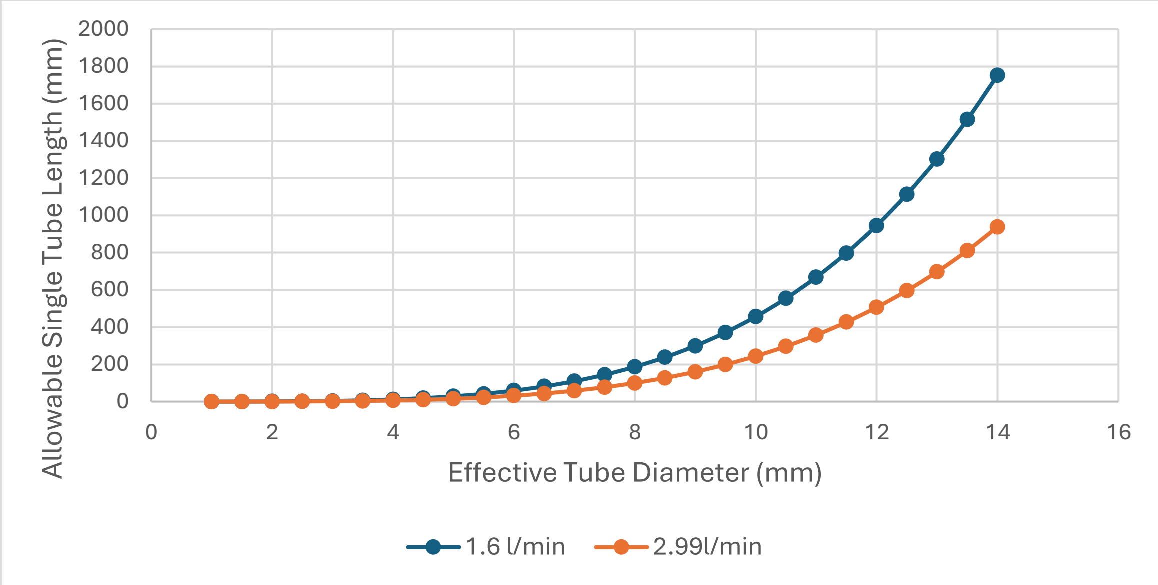
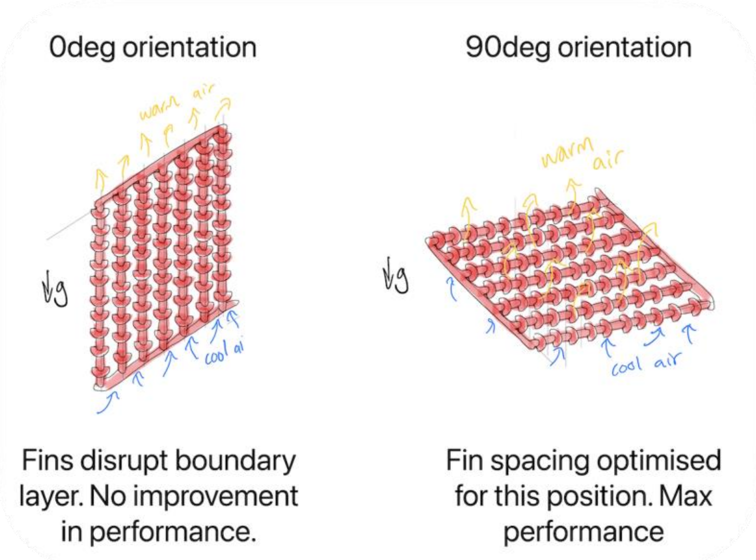


Figure 10: Allowable Tube Length vs. Diameter

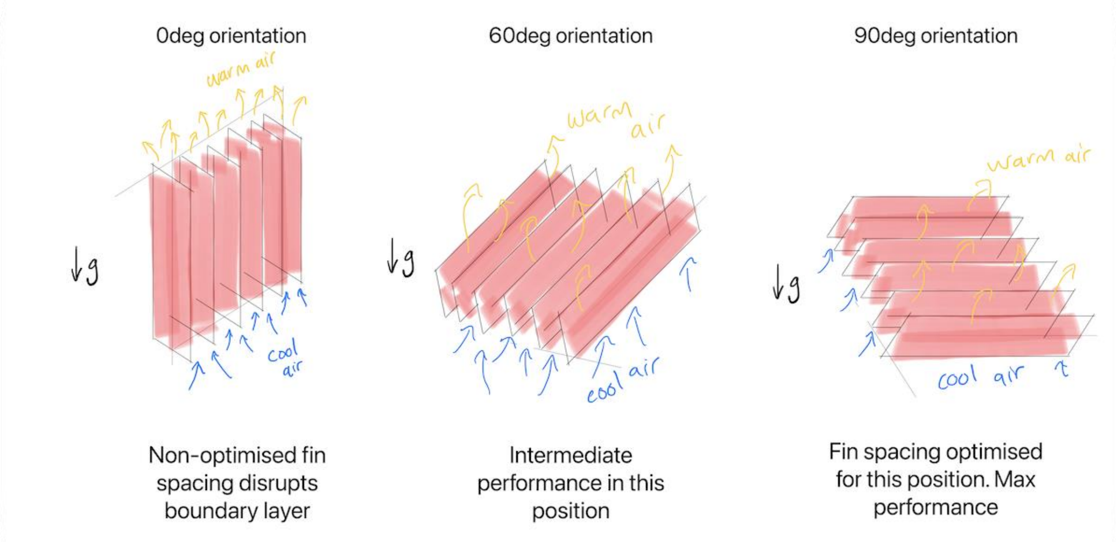
Alternative configurations:

The most effective manner (which the team identified conceptually) to manipulate the heat transfer coefficient for the same surface area would be to alter the orientation of the radiator to either promote or hamper boundary layer development across the heating surfaces.

The team decided on two preliminary design configurations for further study as shown in Figure 11. In the design process, the fin optimisation techniques would be applied to the horizontal orientation (90°) which result in non-optimal operation when placed vertically (0°).



a) tube-fin configuration



b) fin-only configuration

Figure 11: Concept Designs for Variable Heat Transfer Coefficient Radiator (Impacted by Orientation)

The initial designs were therefore based on maximising the potential wattage and was limited by the overall footprint which the radiator could encompass. The selected footprint was based on similar wall-mounted radiators which are $500-600mm$ tall and $800-1{,}000mm$ wide.

I. Tube-Fin Type Design

The first "proof of concept" involved a simple comparison between horizontal and vertical straight pipes and how the configuration angle impacts the heat transfer coefficient. The Nusselt number in these configurations are provided for natural convection by Equation (10) for a horizontal fin arrangement and Equation (11) [15], [16].

$$Nu_D = \left\{0.6 + \frac{0.387Ra_D^{\frac{1}{6}}}{\left[1+(0.559/Pr)^{\frac{9}{16}}\right]^{\frac{8}{27}}}\right\}^2 \quad (10)$$

$$Nu_L = \left\{0.825 + \frac{0.387Ra_L^{\frac{1}{6}}}{\left[1+(0.492/Pr)^{\frac{9}{16}}\right]^{\frac{8}{27}}}\right\}^2 \quad (11)$$

$$h = \frac{Nu_{D/L} \cdot k_{air}}{D/L} \tag{12}$$

For the preliminary assessments the length was fixed at $500mm$ for the horizontal case and a radius of $20mm$ for the vertical case. In each case the Rayleigh number was varied by assuming changes in the pipe diameter and the length of the pipe. From the preliminary calculations, the available heat capacity from the two configurations is 1.4:1 (horizontal:vertical) which is promising for the intended application. However, the resulting area from the simple tube arrangement is far below the targeted $6 - 7m^2$ noted previously, therefore necessitating the use of a finned geometry.

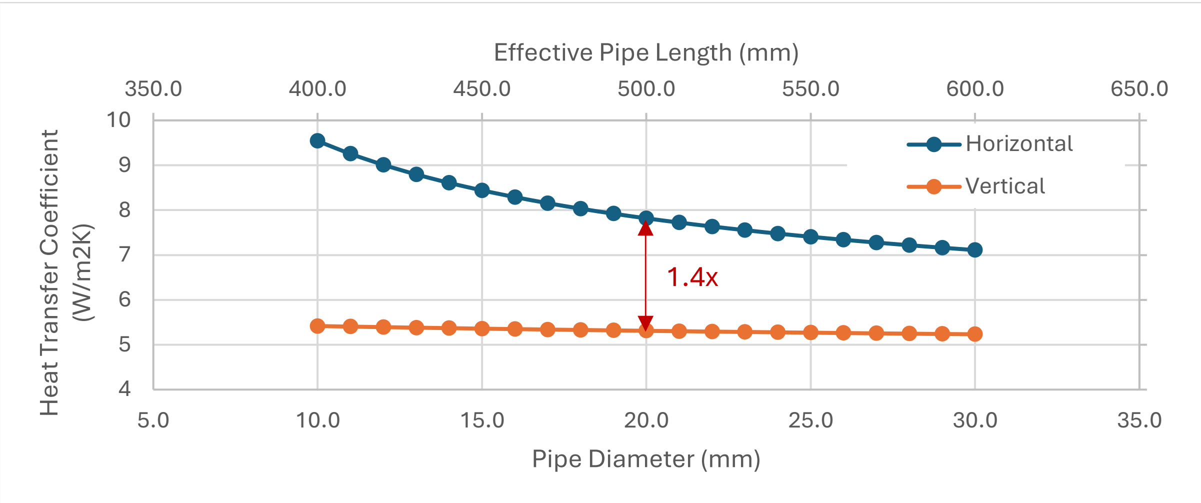


Figure 12: Heat Transfer Coefficient for a Straight Pipe Horizontally vs. Vertically

The design of finned tubes is commonly explored in various tube-bank heat exchanger designs and the experimental estimates of associated Nusselt numbers and flow characteristics are well document by [17]. This work defines the optimal ratios of fin diameter (D) to tube diameter (d) of $D/d \approx 1.87$ and the fin height (h) to spacing (s) of $h/s \approx 1.8$. These ratios were used to select the appropriate tube, fin and spacing arrangements from an available catalogue based on a radiator dimension of $1m\ x\ 0.5m$. The configurations described in Table 4 were considered.

Table 4: Fin-Tube Configurations for Assessment

Option	Tube Diameter (mm)	Fin Diameter (mm)	Fin Spacing (fin/m)
1	9,53	28	315
2	15,88	28	197
3	19,05	35	197
4	22,23	44,5	157
5	25,40	47,6	157

The total available area for these configurations was then estimated by considering the fin efficiency and a revised heat transfer coefficient. The fin efficiency was calculated using the modified Bessel function [15] and shown graphically in Figure 13.

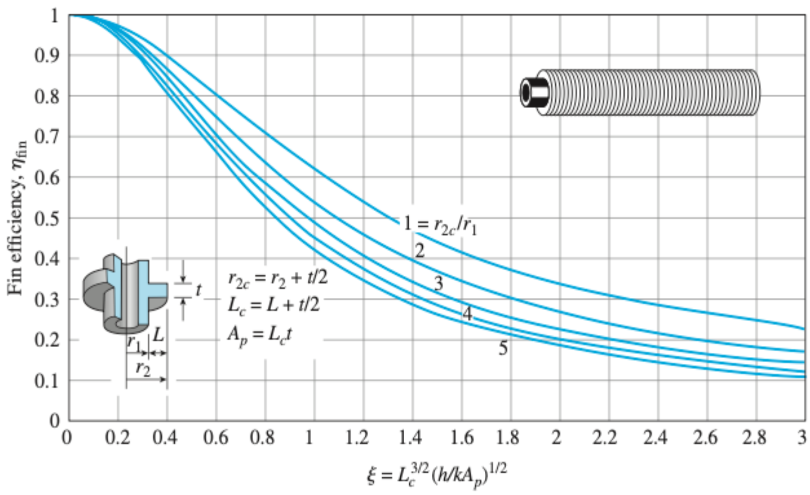


Figure 13: Efficiency of Annular Fins of Constant Thickness [15]

As many of the experimental approximations for the Nusselt number in [17] are for forced convection, it was not reliable to accurately estimate the heat transfer coefficient. Therefore, an approximate was made based the previously calculated heat transfer coefficients of Figure 12. The estimated performance is summarised in Table 5.

Table 5: Estimated Performance for Tube-Fin Radiator Design

Option	Number of Tubes	Q_horizontal (W)	Q_vertical (W)
1	35	1,778	1,020
2	35	939	686

Option	Number of Tubes	Q_horizontal (W)	Q_vertical (W)
3	28	1,038	795
4	22	1,001	798
5	21	1,010	843

A simple CFD model was set up to verify the results. For the model, Option 5 was considered and a symmetry problem defined with five of the 21-tubes modelled. The problem was set up as a isothermal surface temperature boundary condition with a surface temperature of $70°C$. This configuration yielded a total available heat transfer (in the horizontal position) of $\sim 800W$ indicating a deviation from the estimated value by $> 20\%$ and the need for significant further improvement in the geometry. The team believed that a simpler design could be deployed and explored the second configuration.

II. Fin-Only Type Design

The preliminary simulations corroborated well with the initial estimates for the fin-only design and showed a significant improvement in the potential heat transfer coefficient for different orientations. The design is simple to manufacture and aesthetically pleasing and therefore showed fewer barriers to implementation. The detailed design was therefore further developed before final modelling. For the fin-only design, the detailed design is described in Section 1.3.

D. SIMULATION SNAPSHOTS

For this phase, most of the simulations were run as decoupled momentum and energy solutions. By that, we mean that separate simulations were run to confirm the likely behaviour of the working fluid within the multiport channel design (such as flow characteristics and pressure drop). Separately, due to the simplifying assumptions that the working fluid will be adequately dispersed within the geometry and subject to low temperature deltas, the thermal modelling of the radiator was then conducted without considering the working fluid. The results are discussed in the main report (refer Section 1.3) but some of the simulation results are presented here for clarity to the reader. Additional information is available upon request[2].

Key operation configurations are given by Figure 14 and Figure 16. Figure 16 clearly shows how the geometry promotes the flow of air upwards towards the clothes which would be hanging overhead and produces effectively double the power output of the standard ambient heating arrangement (Figure 14). In both instances, the water flow rate remains constant.

[2] Note that a detailed Finite Element Analysis was not conducted on the product which would be done to verify the allowable working pressures but this will be done in the next phase of work.

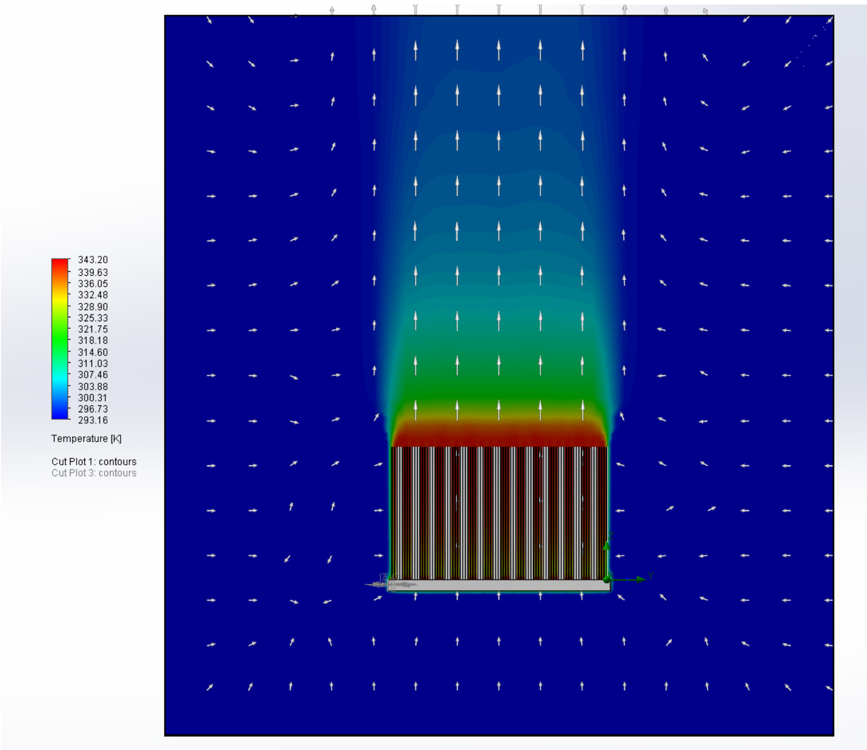


Figure 14: Temperature and Velocity Profile. Average Surface Temperature 70C. Vertical orientation (0deg). Power Output ~930W.

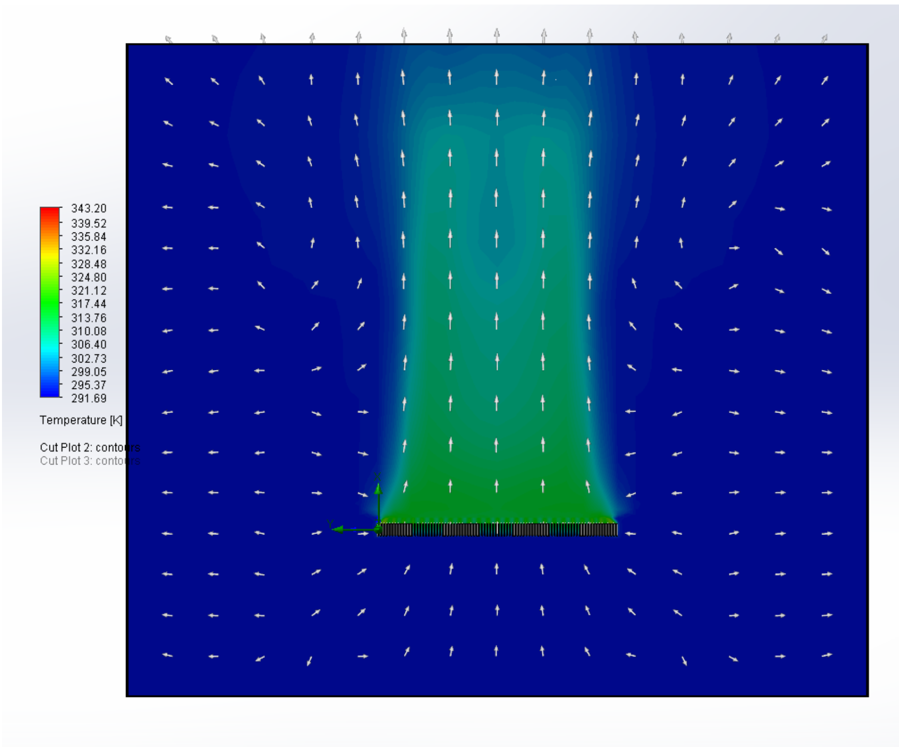


Figure 15: Temperature and Velocity Profile. Average Surface Temperature 70C. Horizontal Orientation (90deg). Power Output ~2,450W.

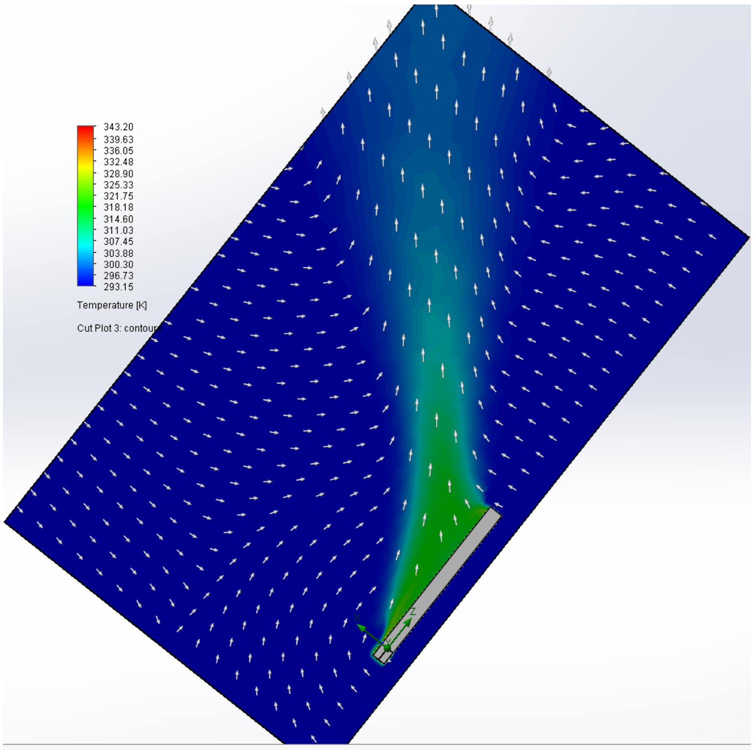


Figure 16: Temperature and Velocity Profile. Average Surface Temperature 70C. Tilted Orientation (45deg). Power Output ~2,280W.

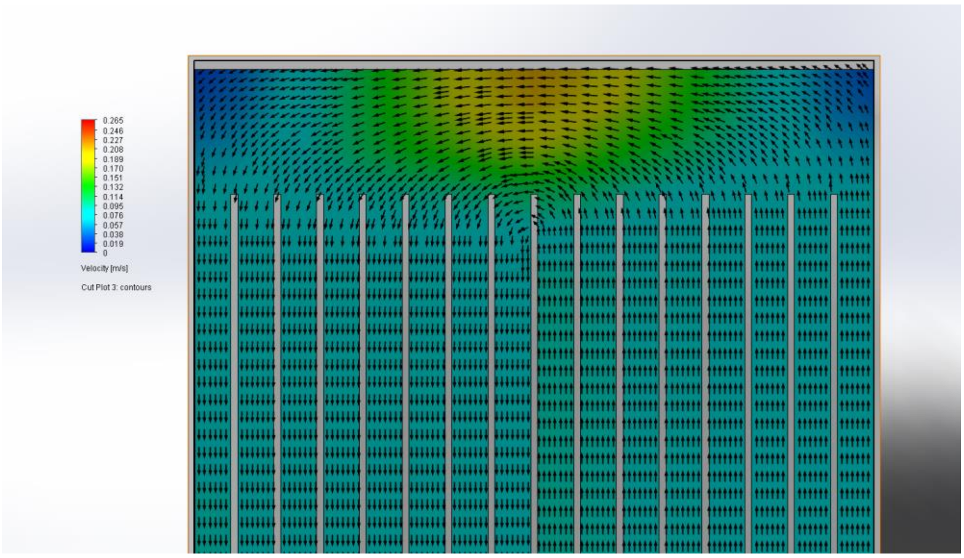


Figure 17: Velocity Profile within Aluminium Microchannels. Clear Flow Profiles with Minimal Losses.

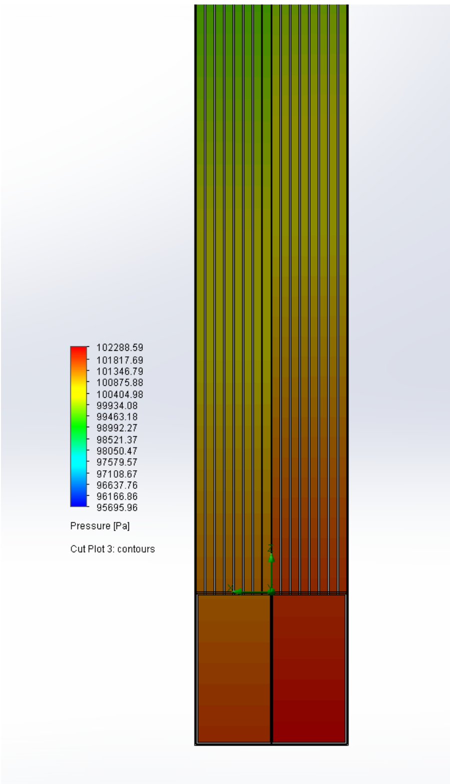


Figure 18: Pressure Contour in Manifold. Total Pressure Loss ~1kPa in Design Conditions.

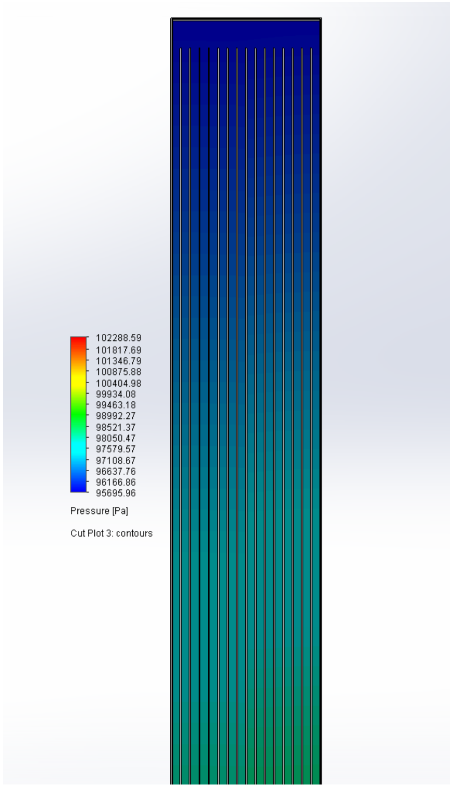


Figure 19: Pressure Contour at Top of Aluminium Channels.

The pressure drop was simulated for a range of inlet velocities at different inclination angles. In all instances, a pressure drop of 20kPa or less is easily achieved thanks to the design.

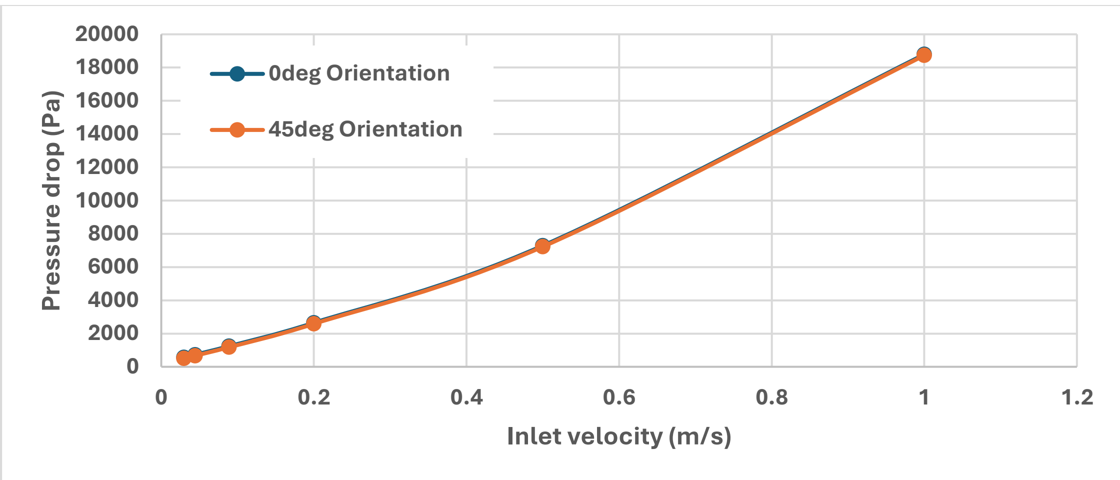


Figure 20: Pressure Drop vs. Inlet Velocity

E. MANUFACTURING DRAWINGS

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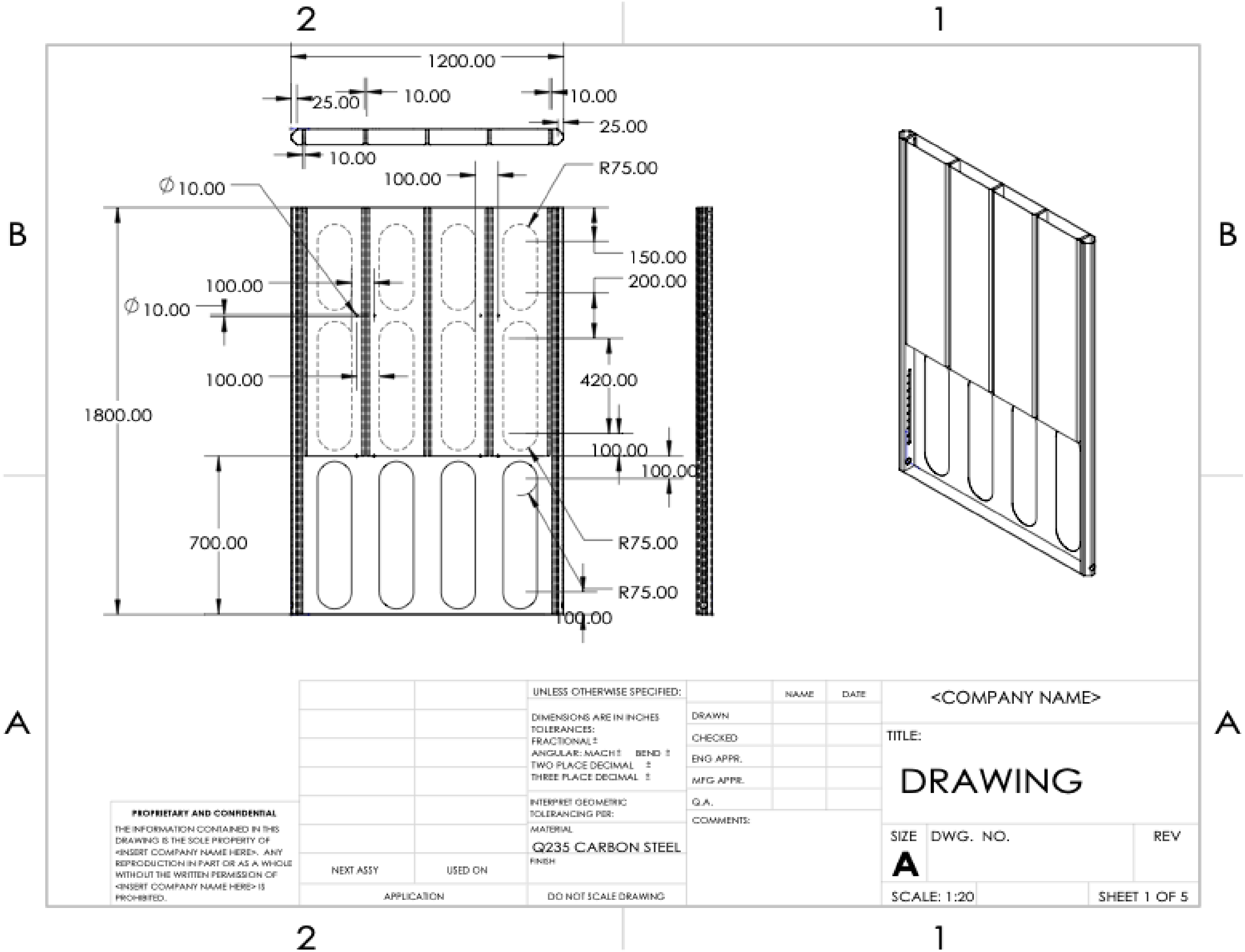

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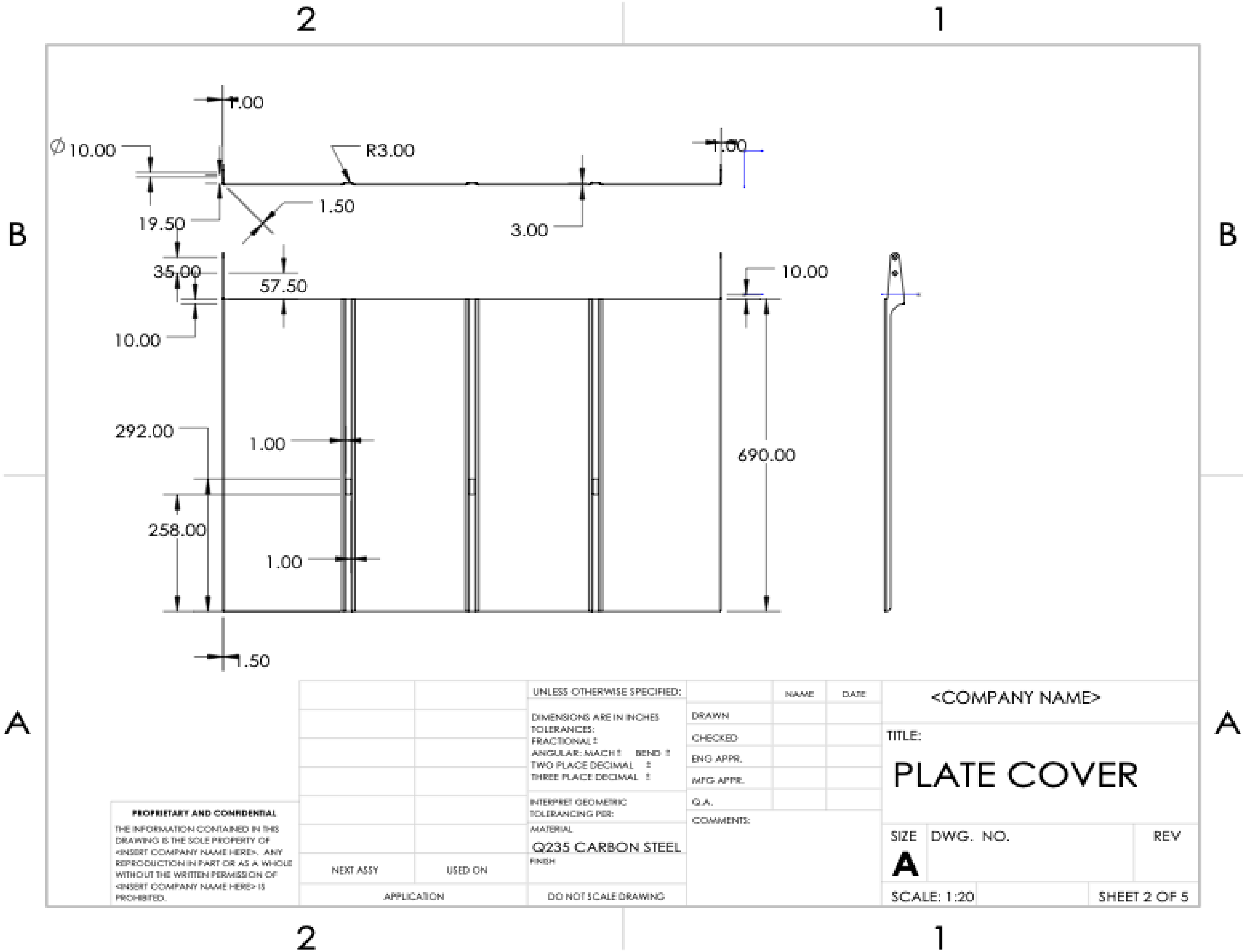

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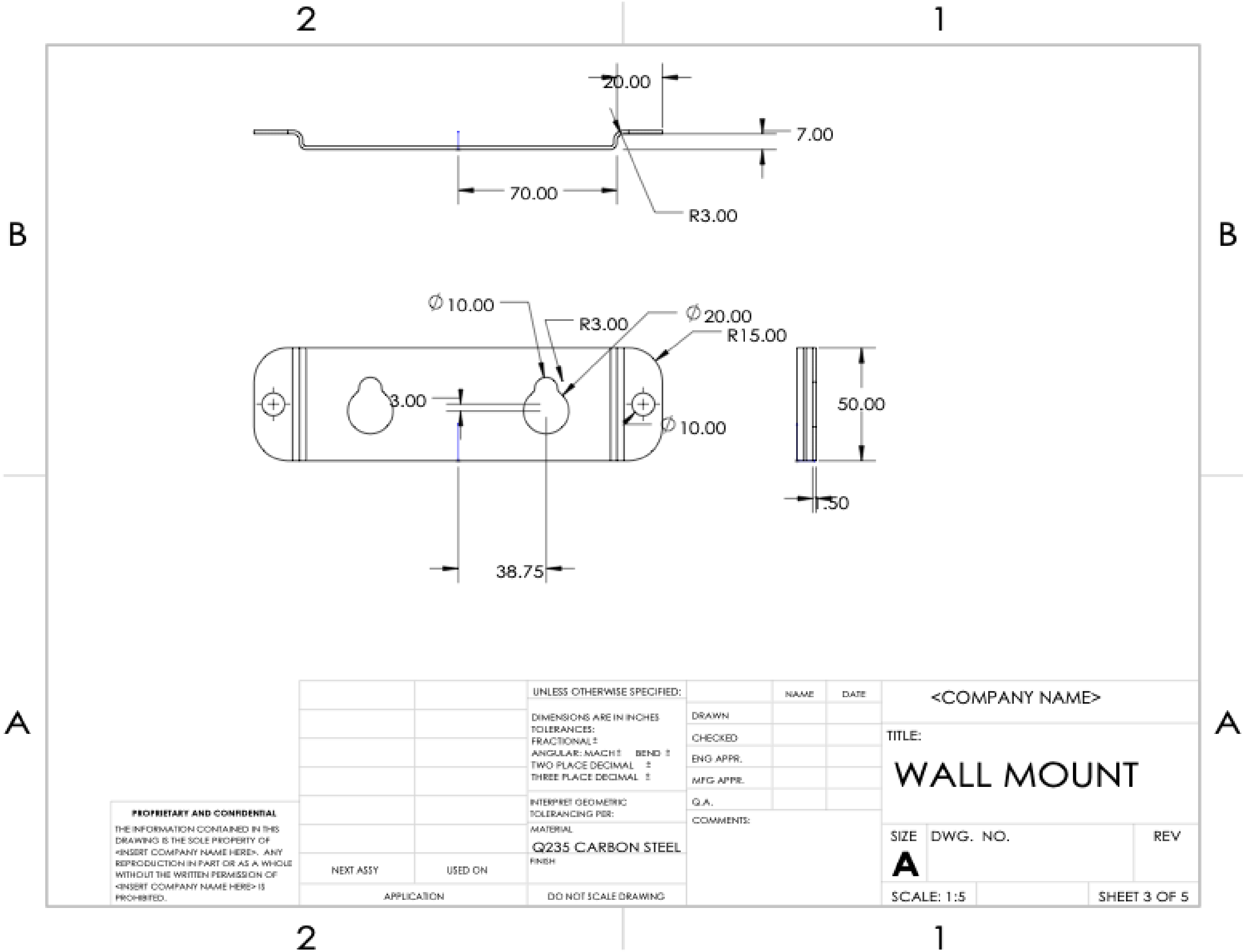

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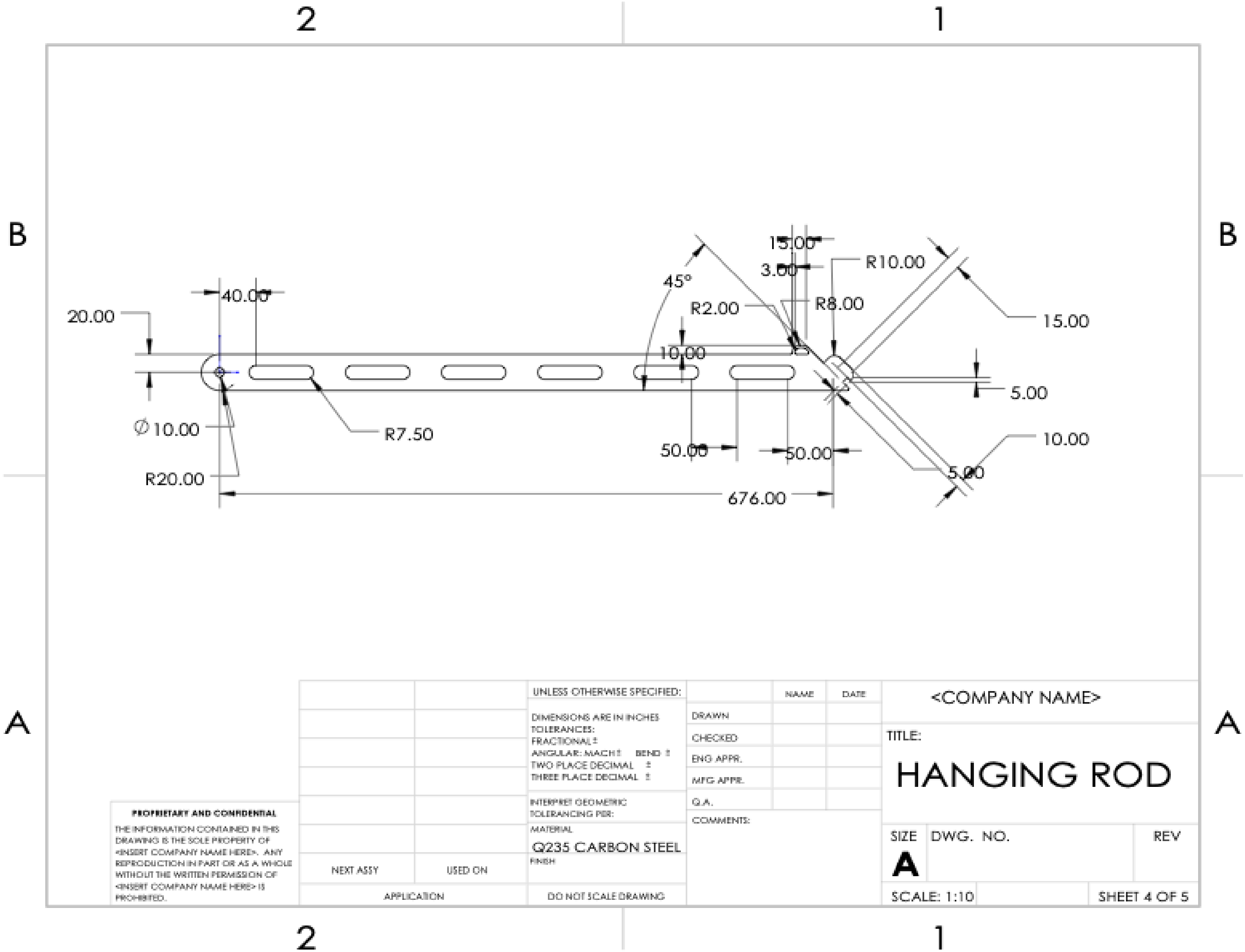

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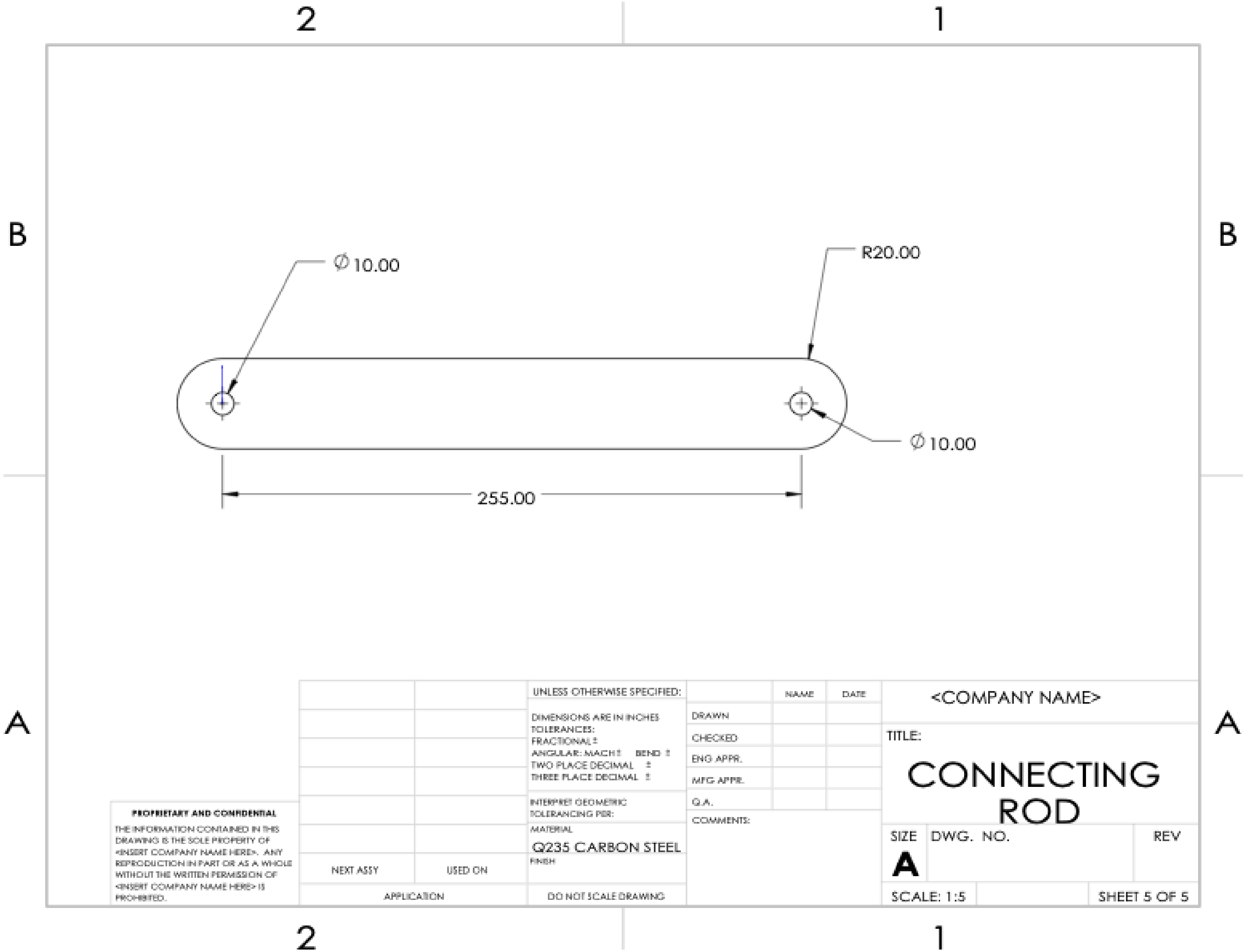

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