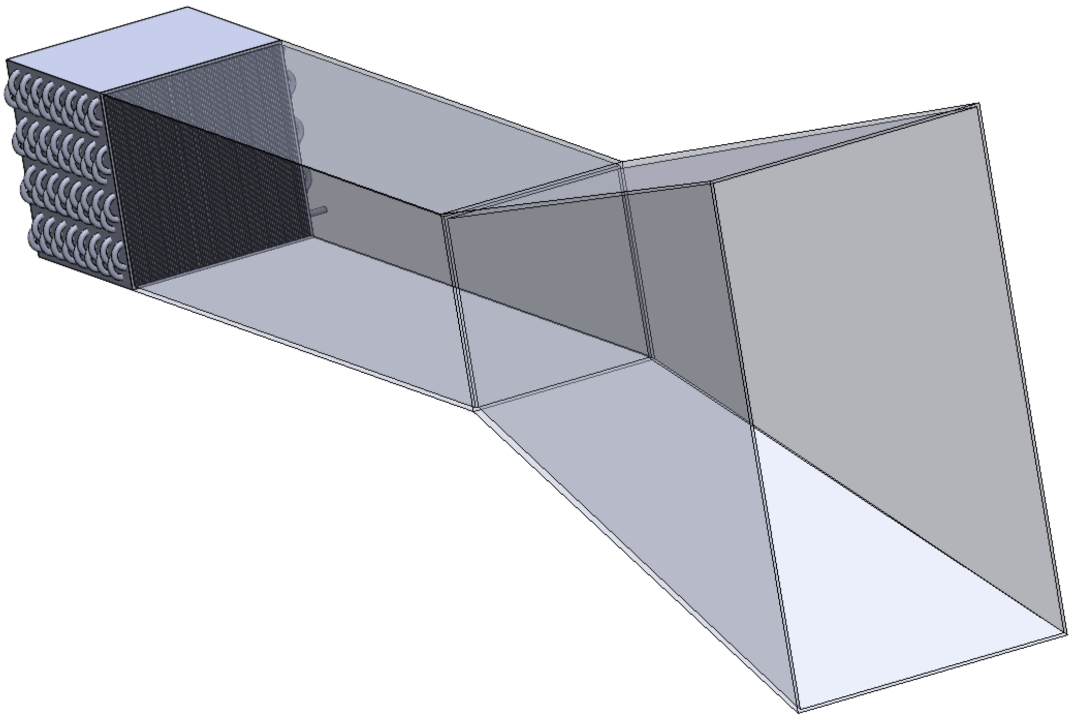
CFD Midterm Project: Automotive Heat Exchanger



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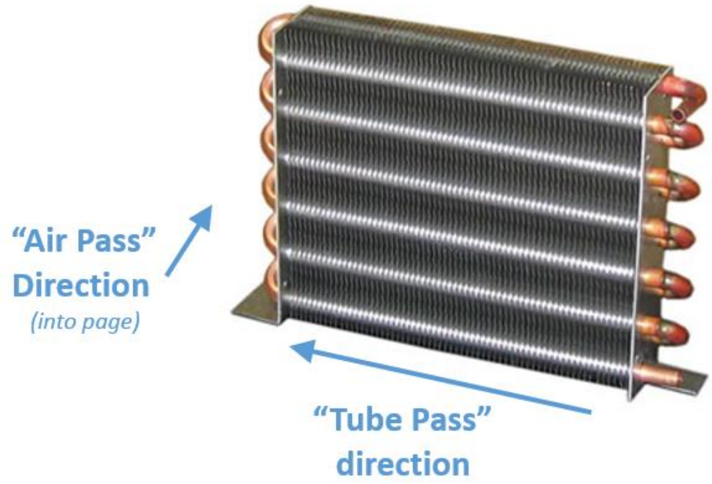
Introduction

Heat exchangers are commonly used in internal combustion engines to reduce the operating temperature of the engine by removing heat from a circulating working fluid, specifically oil in this case. The objective of this project is to design an oil cooler for an internal combustion engine. The apparatus consists of a heat exchanger which uses air to cool the oil and a duct which funnels air to the heat exchanger. The operating conditions of the heat exchanger are described in the section *Design Constraints, Assumptions, and Operating Metrics*.

Design Concept

A fin and tube heat exchanger is chosen for this oil cooler. This type of heat exchanger is common in the automotive industry and is commonly used for this application. The heat exchanger is a crossflow heat exchanger with both fluids unmixed.



CanCoil Thermal Corporation - Finned Tube Heat Exchanger example [1]

The final design parameters and weights of the heat exchanger and duct is summarized in Table 1 below:

Table 1: Final Design Parameters and Dry Weights

Parameter	Value
Heat Exchanger Width	8 in
Heat Exchanger Height	8 in
Heat Exchanger Depth	8 in
Tube Outer Diameter	0.25 in
Tube Thickness	0.035 in
Fin Thickness	0.01 in
Fin Spacing	0.04 in
Number of Fins	133
Oil Volumetric Flow Rate	5.3 gpm
Heat Exchanger Dry Weight (Copper)	24.33 lbs
Duct Dry Weight (6061 Aluminum)	5.82 lbs

The design dimensions of the heat exchanger are determined largely using hand calculations and are validated using multiple CFD simulations of increasing complexity. The first set of simulations considers a single tube, two-fin setup to validate the fin thickness, fin spacing, tube outer diameter, and tube thickness calculations. The second set of simulations considers two tubes in a staggered configuration with two fins to validate the pressure drop along a typical tube bend and the spacing between adjacent tubes. Finally, the third set of simulations considers the duct to validate the air pressure drop across the duct and the velocity at the entrance to the heat exchanger. Each of the individual CFD simulations are outlined in the table below.

CFD Modeling Roadmap

Model Description	Model Objective(s)
Two Fin Model	Validate fin thickness, fin spacing, tube outer diameter, and tube thickness
Tube Bank Model	Validate pressure drop along tube bend and spacing between adjacent tubes
Duct Model	Validate air pressure drop over heat exchanger and velocity at the entrance to the heat exchanger

Design Constraints, Assumptions, and Operating Metrics

Design Constraints:

- The heat exchanger must be able to cool oil from an inlet temperature of 350 °F to an outlet temperature of 195 °F.
- The oil has a specific gravity of 0.86, a viscosity of 34.5 cP, a thermal conductivity of 0.15 Btu/hr-ft R, and a specific heat of 0.5 Btu/lbm R.
- The heat exchanger must be effective when the car is traveling at a minimum of 25 mph.
- The heat exchanger must be effective in 108 °F ambient temperature.
- The duct inlet face must be 22 in. by 12 in.
- The duct outlet must have a maximum area of 250 in^2 with no dimension larger than 36 in.
- The heat exchanger is located 60 in. downstream of the duct inlet.
- The oil volumetric flow rate at maximum load must be chosen to be between 3.5 and 5.5 gpm.

Design Assumptions:

- All solid parts of the heat exchanger (the fins and tubes) will be constructed out of copper. Copper is a common material for heat exchangers, as its high thermal conductivity is appropriate for this application.
- Radiation is ignored in this design study. It is typical for tube surfaces and heat exchanger wall surfaces to be polished, significantly reducing the wall emissivities. Further, temperature differences between the fin and heat exchanger walls are not large enough to justify accounting for radiation in models and hand calculations, as will be demonstrated in the CFD model results.
- The duct will be constructed out of 6061 aluminum primarily for its relatively low density compared to copper with high thermal conductivity.
- Tubes in a crossflow heat exchanger can be either aligned in-line or staggered. We assume that the tubes are staggered in our heat exchanger; heat exchanger tubes aligned in a staggered configuration are typically slightly more effective than their in-line counterparts [2].

The materials used in each analysis (air, engine oil, copper, and 6061 aluminum) are summarized in Tables 2-5 below:

Table 2: Ambient Air Properties

Ambient Air	
Density	0.06989 lb/ft^3 (determined via ideal gas law at atmospheric pressure and 108 °F)
Specific Heat	0.24 BTU/lb-R [3]
Thermal Conductivity	0.0157 BTU/hr-ft-R [3]
Viscosity	$1.293*10^{-5}$ lb/s-ft [3]

Table 3: Engine Oil Properties

Engine Oil	
Density	53.664 lb/ft^3
Specific Heat	0.5 BTU/lb-R
Thermal Conductivity	0.15 BTU/hr-ft-R
Viscosity	0.02318 lb/s-ft

Table 4: Copper Properties

Copper	
Density	557 lb/ft^3 [3]
Specific Heat	0.09 BTU/lb-R [3]
Thermal Conductivity	233 BTU/hr-ft-R [3]

Table 5: Aluminum Properties

Aluminum	
Density	168 lb/ft^3 [3]
Specific Heat	0.21 BTU/lb-R [3]
Thermal Conductivity	136 BTU/hr-ft-R [3]

Operating Metrics:

The steady state performance operating metrics are shown in Table 6. These are summarized results from the CFD simulations and hand calculations.

Table 6: Heat Exchanger Operating Metrics

Oil Flow Rate		5.3 gpm
Oil Temperature	Inlet	350 °F
	Outlet	195 °F
Air Temperature	Inlet	108 °F
	Outlet	151.7 °F
Oil Pressure Drop		5787 psi
Air Pressure Drop (Duct)		0.162 psi
Air Pressure Drop (Heat Exchanger)		3.2 psi

Heat Exchanger Design: Design Hand Calculations

This heat exchanger design problem can be solved using hand calculations, which are employed in this report. CFD is used to validate the results and assumptions derived from these hand calculations, ensuring the accuracy and reliability of the design. This approach is particularly beneficial when using submodels to approximate overall scale behavior. Additionally, once a detailed workflow is established between the design inputs and the desired outputs (minimizing the weight and size of the heat exchanger), the problem can be directly solved using a multidimensional optimization tool like Excel's Solver. This is the approach taken here.

The oil entrance and exit temperatures, the air entrance temperature, and the oil and air mass flow rates are given. The air exit temperature can be calculated thermodynamically (assuming steady state behavior and perfect heat transfer between the oil and air):

$$T_{air,exit} = \frac{\rho_{oil}\dot{V}c_{p,oil}(T_{oil,entrance} - T_{oil,exit})}{\rho_{air}VA_{duct,inlet}c_{p,air}} + T_{air,entrance}$$

Plugging in:

$$T_{air,exit} = \frac{(0.86)(62.4\frac{lb}{ft^3})(5.3gpm)(0.134\frac{ft^3}{gal})(0.5\frac{BTU}{lb-R})(350^\circ F - 195^\circ F)}{(0.06989\frac{lb}{ft^3})(25mph)(88\frac{fpm}{mph})(22in)(12in)(\frac{1}{144}\frac{ft^2}{in^2})(0.24\frac{BTU}{lb-R})} + 108^\circ F = 151.7^\circ F$$

This balance also demonstrates that the amount of heat released by the oil (or absorbed by the air) is 177,221 BTU/hr. In terms of an overall heat transfer coefficient, the heat transfer can also be expressed as:

$$\dot{Q} = UAF\Delta T_{LMTD},$$

where F is the correction factor to account for differences between a counterflow heat exchanger and a crossflow heat exchanger. F can be determined graphically using relationships between inlet and outlet temperatures:

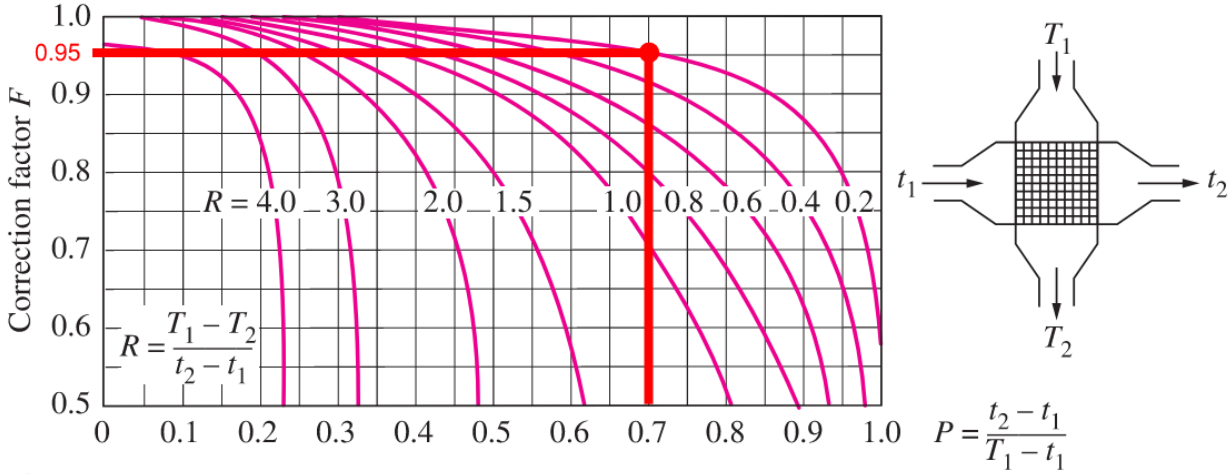


Figure 1: Correction factor graphical calculation [4]

Per Figure 1, F = 0.95. Using this, the UA product of this heat exchanger is:

$$UA = \frac{177,221\,\frac{BTU}{hr}}{(0.95)\left(\frac{(T_{oil,exit}-T_{air,entrance})-(T_{oil,entrance}-T_{air,exit})}{\ln\left(\frac{T_{oil,exit}-T_{air,entrance}}{T_{oil,entrance}-T_{air,exit}}\right)}\right)} = 1381\frac{BTU}{hr-R}$$

With UA determined, we now calculate an approximation for U to determine the length of piping needed to achieve the desired temperature differences.

The overall heat transfer coefficient for this heat exchanger is approximated using a thermal resistive network. The geometry in this network "stretches out" the heat exchanger into a single tube with square fins spaced regularly along the tube's length. This assumption is justifiable if thermal/fluid interactions between adjacent tubes are minimal and if the number of tube rows is minimal compared to the overall heat exchanger width/height. A review of the final design parameters in Table 1 and the CFD model results will prove this assumption to be justified. A visual representation of the thermal resistive network geometry and of the thermal resistive network of a typical tube segment and fin is shown in Figures 2 and 3, respectively.

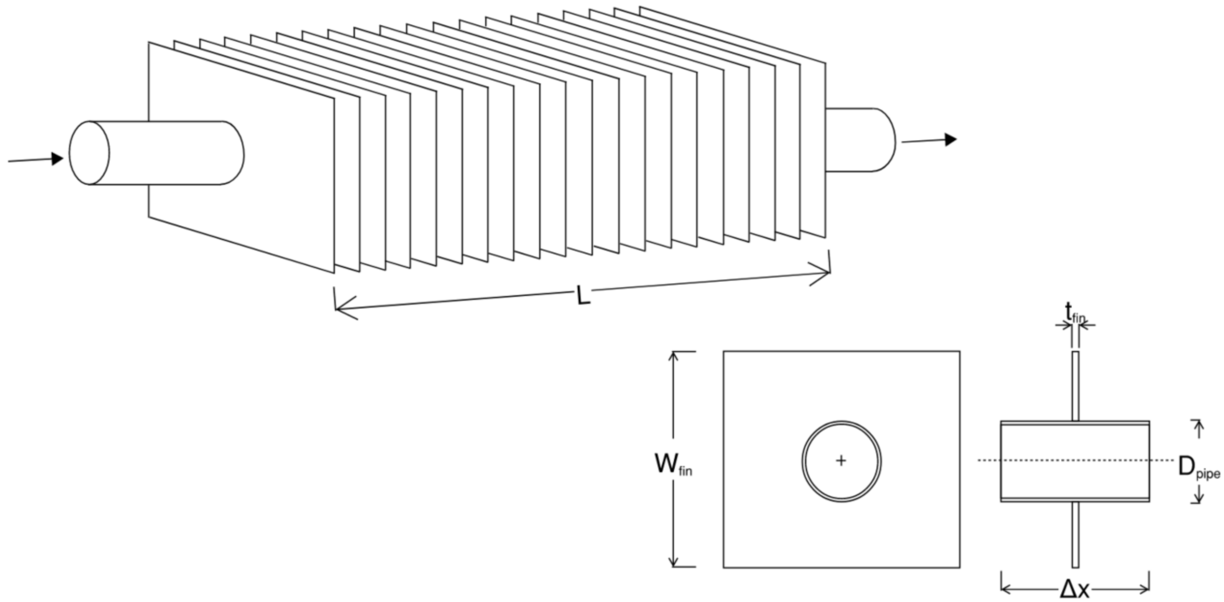


Figure 2: Thermal resistive network simplified geometry

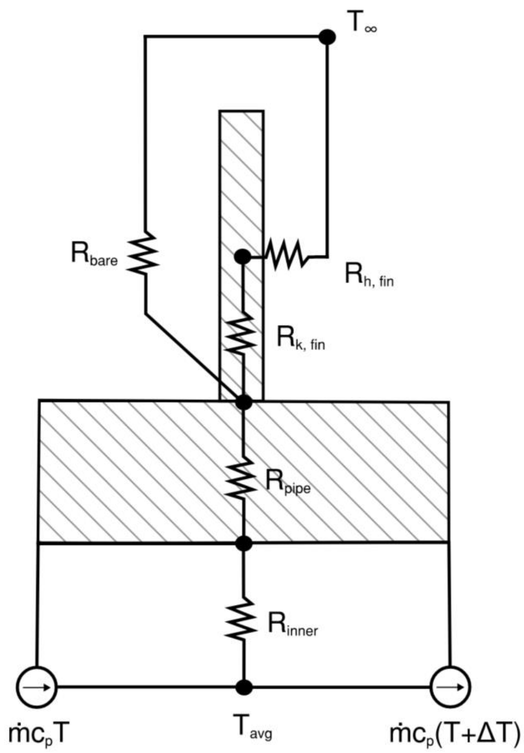


Figure 3: Thermal resistive circuit for sample fin

Based on the thermal resistive network shown, the total thermal resistance per fin can be given as:

$$R_{total} = R_{inner} + R_{pipe} + \frac{1}{\frac{1}{R_{bare}} + \frac{1}{R_{k,fin}+R_{h,fin}}},$$

where

$$R_{inner} = \frac{1}{h_{inner}A_{inner}}$$

$$R_{pipe} = \frac{\ln\left(\frac{D_{outer}}{D_{inner}}\right)}{2\pi k_{pipe}(\Delta x_{fin} - t_{fin})}$$

$$R_{bare} = \frac{1}{h_{outer,bare}A_{outer,bare}}$$

$$R_{k,fin} = \frac{\frac{L_{fin}-D_{outer}}{2}}{k_{fin}A_{fin}}$$

$$R_{k,fin} = \frac{1}{h_{outer,fin}A_{fin}}$$

At this point, there are three values to solve for: h_{inner}, $h_{outer,bare}$, and $h_{outer,fin}$. These values are derived from Nusselt number correlations for fully developed internal flow, external flow over a cylinder, and external flow over a flat plate, respectively. Each correlation is discussed in detail in the sections below.

Fully Developed Internal Flow:

The average fully developed internal flow Nusselt number correlation is given by the Dittus-Boelter equation. All relevant calculations are provided below:

$$Re = \frac{4\dot{m}}{\pi\mu D_{inner}} = \frac{4(53.664\frac{lb}{ft^3})(5.3gpm)(0.134\frac{ft^3}{gal})(\frac{1}{60}\frac{min}{s})}{\pi(0.02318\frac{lb}{s-ft})(\frac{0.18}{12}ft)} = 2326$$

$$Pr = \frac{\mu c_{p,oil}}{k_{oil}} = \frac{(0.02318\frac{lb}{s-ft})(0.5\frac{BTU}{lb-R})(3600\frac{s}{hr})}{0.15\frac{BTU}{hr-ft-R}} = 278$$

$$Nu = 0.023Re^{0.8}Pr^{0.3} = 61.4$$

$$h_{inner} = \frac{(Nu)(k_{oil})}{D_{inner}} = 614\frac{BTU}{hr-ft^2-R}$$

In these calculations, the tube inner diameter and oil volumetric flow rate are the only design parameters that can be varied. Because the Nusselt number for laminar flow is much lower than that for turbulent flow (Nu = 3.66 for laminar flow), the tube outer diameter, tube thickness, and oil volumetric flow rate were selected such that the Reynolds number for the internal flow is as close to 2300 (the transition to turbulent flow in a tube) as possible. While the goal of this design project was only to optimize the heat exchanger for size and weight and not for pressure drop, an effort was made to keep pressure values as reasonable as possible, primarily to prevent material rupture under maximum load conditions.

External Flow Over a Cylinder:
The average external flow over a cylinder Nusselt number correlation is given by Zukauskas et al. [5]. All relevant calculations are provided below:

$$Re = \frac{V_{air} D_{outer}}{\nu} = \frac{(25mph)(88\frac{fpm}{mph})(\frac{1}{60}\frac{min}{s})(22in)(12in)(\frac{0.25}{12}ft)}{(1.85\times10^{-4}\frac{ft^2}{s})(8in)(8in)} = 17033$$

$$Pr = \frac{\mu c_{p,air}}{k_{air}} = \frac{(1.293\times10^{-5}\frac{lb}{s-ft})(0.24\frac{BTU}{lb-R})(3600\frac{s}{hr})}{0.0157\frac{BTU}{hr-ft-R}} = 0.71$$

$$Nu = 0.193 Re^{0.618} Pr^{\frac{1}{3}} = 71$$

$$h_{outer,bare} = \frac{(Nu)(k_{air})}{D_{outer}} = 53.5\frac{BTU}{hr-ft^2-R}$$

In these calculations, the heat exchanger height and width and the tube outer diameter are the only design parameters that can be varied. As the Reynolds number grows, the Nusselt number grows commensurately; however, there are diminishing returns to performance because increasing the tube outer diameter to increase the Reynolds number also increases the overall heat exchanger weight. With this in mind, we elected to aim for the second highest group of Nusselt number correlations to get reasonably high external convection coefficients while keeping weight and overall size down.

External Flow Over a Flat Plate:
The average external flow over a flat plate Nusselt number correlation and all relevant calculations are given by:

$$Re = \frac{V_{air} w_{fin}}{\nu} = \frac{(25mph)(88\frac{fpm}{mph})(\frac{1}{60}\frac{min}{s})(22in)(12in)(\frac{0.5}{12}ft)}{(1.85 \times 10^{-4}\frac{ft^2}{s})(8in)(8in)} = 34066$$

$$Pr = \frac{\mu c_{p,air}}{k_{air}} = \frac{(1.293 \times 10^{-5}\frac{lb}{s-ft})(0.24\frac{BTU}{lb-R})(3600\frac{s}{hr})}{0.0157\frac{BTU}{hr-ft-R}} = 0.71$$

$$Nu = 0.664 Re^{0.5} Pr^{\frac{1}{3}} = 109$$

$$h_{outer,fin} = \frac{(Nu)(k_{air})}{w_{fin}} = 41\frac{BTU}{hr - ft^2 - R}$$

$$\delta_t = \frac{5 w_{fin}}{Re^{0.5} Pr^{\frac{1}{3}}} = 0.015in$$

$$\Delta x = 2\delta_t + t_{fin} = 0.04in$$

Note that these calculations include both an approximation for the convection coefficient between the fin and air and a calculation for fin spacing based on the thermal boundary layer thickness. We assume that the minimum distance between fins is double the thermal boundary layer thickness at the end of the fin; this assumption is validated via subsequent CFD simulations. In these calculations, the heat exchanger height and width, the fin thickness, and the width of the fin are the only design parameters that can be varied. As the Reynolds number grows, the Nusselt number grows commensurately; however, there are diminishing returns to performance because increasing the width of the fin to increase the Reynolds number also increases the overall heat exchanger weight. The overall weight of the heat exchanger would be larger if the flow was turbulent; we elected to maintain the flow as laminar for this correlation to keep the overall heat exchanger weight minimized.

Summary of Thermal Resistances and Thermal Parameter Results:

A summary of all thermal resistance values based on the calculated convection coefficients is summarized in Table 7 below:

Table 7: Thermal Resistance Summary

Resistance	Value
R_{inner}	10.28 R-hr/BTU
R_{pipe}	0.089 R-hr/BTU
R_{bare}	113 R-hr/BTU
$R_{k,\ fin}$	0.016 R-hr/BTU
$R_{h,\ fin}$	8.69 R-hr/BTU
R_{total} (per fin)	18.45 R-hr/BTU

Using the internal surface area as a reference, the overall heat transfer coefficient is:

$$U_i = \frac{1}{R_{total}A_i} = 342\frac{BTU}{hr - ft^2 - R}$$

Using the UA factor defined above, the resulting tube length needed to reach the desired temperature differences in the heat exchanger is 1028 in. Dividing this length by the fin spacing and accounting for the fact that most of these fins are actually shared among multiple tubes, 133 unique fins are needed along the width of the heat exchanger. With all design parameters defined, the weight of the heat exchanger is measured volumetrically in SolidWorks to be 24.33 lbs.

Pressure Drops:

The oil pressure drop throughout the heat exchanger can be calculated using a Bernoulli equation calculation:

$$\Delta P_{oil} = \gamma f \frac{L}{D_{inner}} \frac{V^2}{2g},$$

where γ is the specific weight of oil and f is the friction factor for turbulent flow in smooth tubes, determined using the first Petukhov equation [5] as:

$$f = 0.184Re^{-0.2} = 0.039$$

Plugging in,

$$\Delta P_{oil} = (53.664\frac{lb_f}{ft^3})(0.039)\frac{1028in}{0.18in}\frac{(67fps)^2}{2(32.2\frac{ft}{s^2})}(\frac{1}{144}\frac{ft^2}{in^2}) = 5787psi$$

The pressure drop for air in the heat exchanger can be approximated using the pressure drop correlation derived by Zukauskas et al. [5] for tube banks:

$$\Delta P_{air,exchanger} = N_L f \chi \frac{\rho V_{max}^2}{2},$$

where f and χ are determined using the graphs in Figure 4 below:

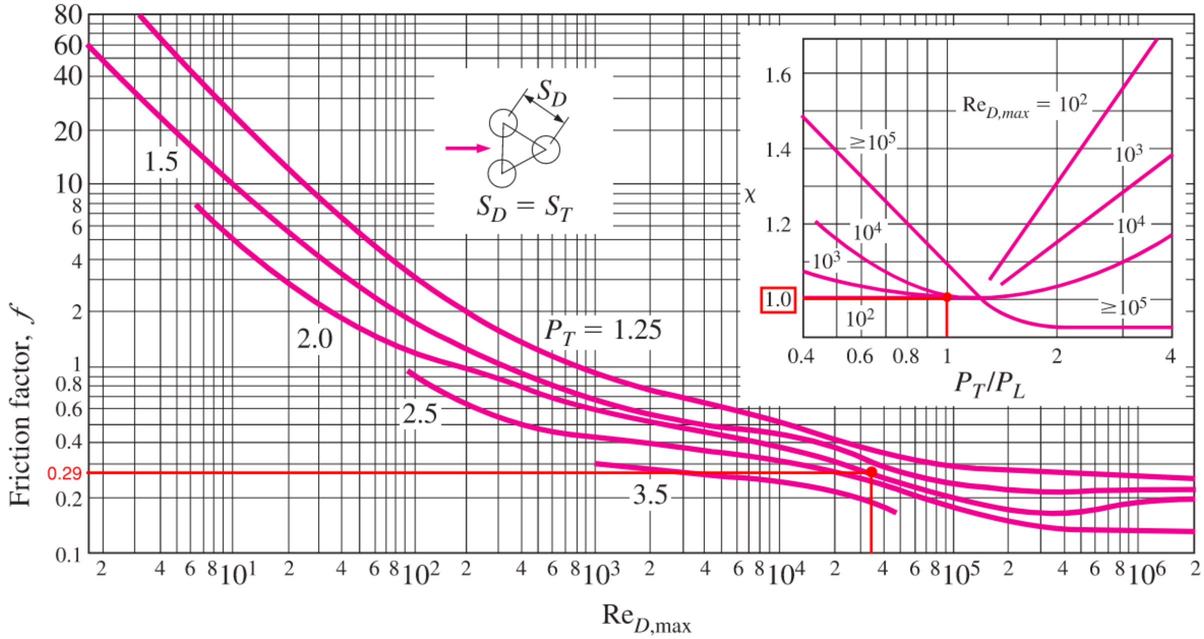


Figure 4: Friction factor and correction factor graphical determination

Plugging in:

$$\Delta P_{air,exchanger} = (16)(0.29)(1)\frac{(0.07\frac{lb}{ft^3})(302.5fps)^2}{2}\left(\frac{1}{32.2}\frac{lb_f s^2}{lbft}\right)\left(\frac{1}{144}\frac{ft^2}{in^2}\right) = 3.2psi$$

The pressure drop for air in the duct could be more involved if compressible effects of air are not negligible.A popular criterion for ignoring compressible effects is if the Mach number is below 0.3 throughout the channel. As a preliminary check, we will determine the Mach numbers at the inlet and outlet to gauge whether compressible flow functions are appropriate for this calculation.

The speed of sound at the inlet of the duct can be calculated as:

$$c_1 = \sqrt{kRT} = \sqrt{(1.4)(53.35\frac{lb_f ft}{lbR})(32.2\frac{lbft}{s^2 lb_f})(567.69R)} = 1,168.5fps$$

The Mach number at the inlet can be calculated as:

$$M_1 = \frac{V_1}{c_1} = \frac{(25mph)(88\frac{fpm}{mph})(\frac{1}{60}\frac{min}{s})}{1,168.5fps} = 0.03 << 0.3$$

Using a lookup table for isentropic compressible flow functions [6], the ratio of inlet area to critical area (where M = 1) is equal to 19.3005. Therefore, the critical area can be calculated as:

$$A^* = \frac{A_1}{19.3005} = \frac{(22in)(12in)}{19.3005} = 13.68in^2$$

The ratio of outlet area to critical area is:

$$\frac{A_2}{A^*} = \frac{(8in)(8in)}{13.68in^2} = 4.679$$

Using the same lookup table for isentropic compressible flow functions, the Mach number at this ratio of area to critical area is $M_2 = 0.125$. At both the inlet and outlet, the Mach number is below 0.3; incompressible/Bernoulli equations can be used in lieu of compressible flow functions for the duct.

The pressure drop across the duct can be calculated similarly to the pressure drop for oil in the heat exchanger using a Bernoulli equation calculation:

$$\Delta P_{air,duct} = \gamma\frac{(V_2^2 - V_1^2)}{2g} = (0.07\frac{lb_f}{ft^3})\frac{(151.25fps)^2 - (36.67fps)^2}{2(32.2\frac{ft}{s^2})}(\frac{1}{144}\frac{ft^2}{in^2}) = 0.162psi$$

Heat Exchanger Efficiency:

The efficiency of a counterflow heat exchanger that transfers heat from oil to air is given by [7]:

$$\eta = \frac{1 - e^{-\alpha}}{1 - \frac{\dot{m}_{oil}c_{p,oil}}{\dot{m}_{air}c_{p,air}}e^{-\alpha}}$$

, and

$$\alpha = U_i\pi DL(\frac{1}{\dot{m}_{oil}c_{p,oil}} - \frac{1}{\dot{m}_{air}c_{p,air}})$$

Using the average heat transfer coefficient and overall length calculated above, the efficiency of this heat exchanger is calculated to be 66%.

Heat Exchanger Design: Two Fin Model

Motivation:
This section develops and explains a lightweight model of two fins along with the section of tube it's attached to. The purpose of this model is to validate the fin thickness, fin spacing, tube outer diameter, and tube thickness calculated in the section above. To achieve this, a model was created in ANSYS Fluent.

Geometry and Mesh:
The same basic geometry is used for all studies. To reduce the size and complexity of the model, only a short section of tube, equal to the spacing between the fins plus the thickness of two fins in addition to buffer segments outside of the heat transfer zone to allow for fully developed flow, is modeled. For these studies, the tube and fins are modeled explicitly using solid regions. A polyhedral mesh is used to ensure good quality cells are maintained around the fins and tube. A summary of meshing parameters and the resulting geometry is shown below:

Table 8: Two Fin Model Meshing Parameters

Meshing Parameter	Value
Fin local sizing	0.02 in.
Tube local sizing	0.04 in.
Oil local sizing	0.005 in.
Air local sizing	0.03 in.
Air refinement region near tube	0.0075 in.
Number of air boundary layers	5
Number of solid boundary layers	1
Volume mesh size	0.03 in.
Volume mesh growth rate	1.1
Total number of cells	1,329,289

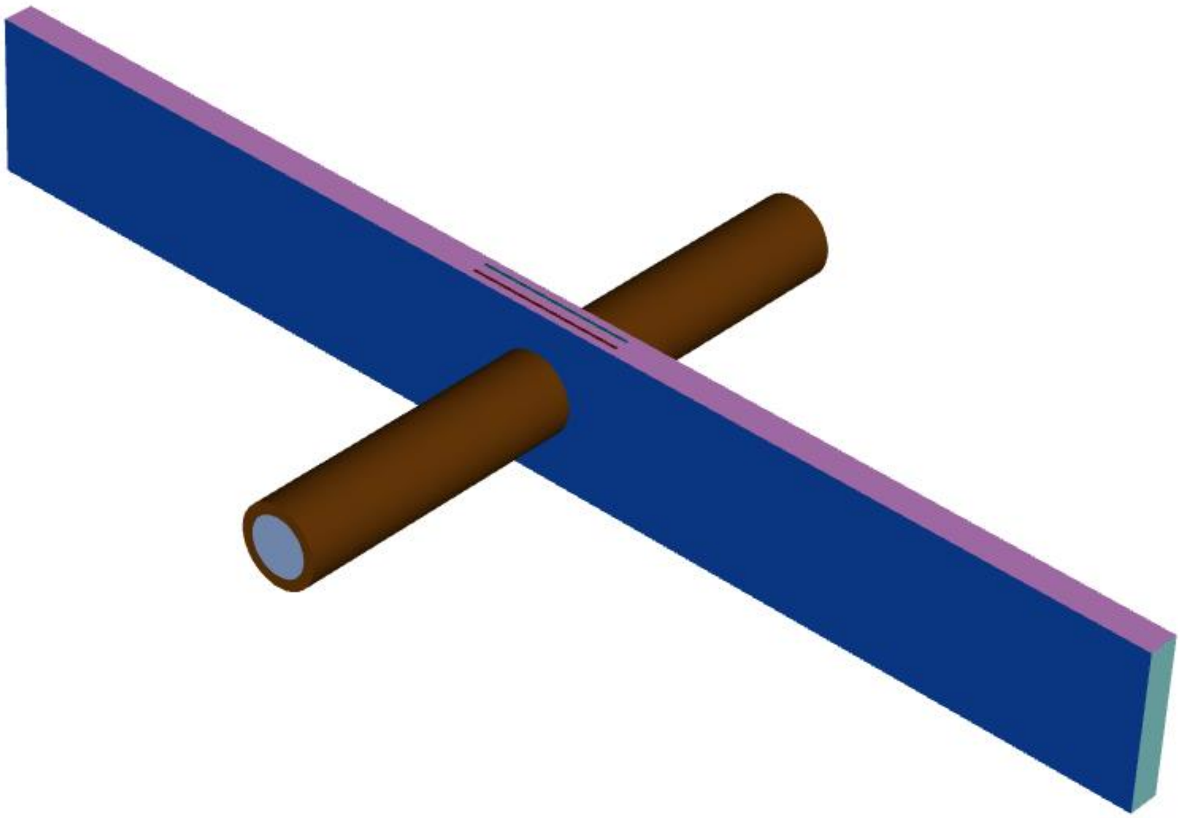


Figure 5: Geometry of the two fin model

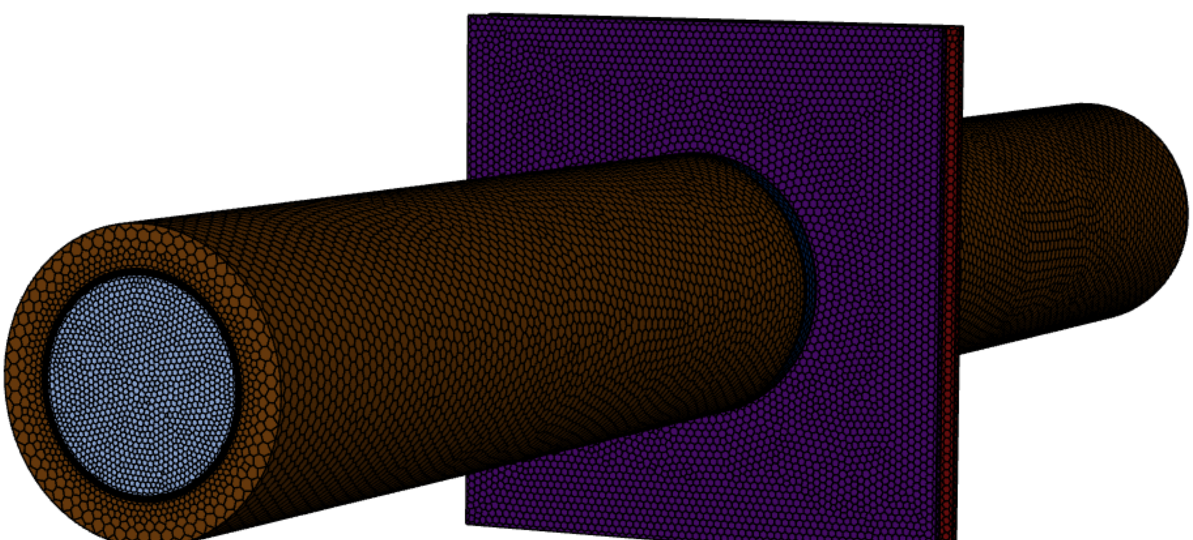


Figure 6: Mesh of the fin and tube

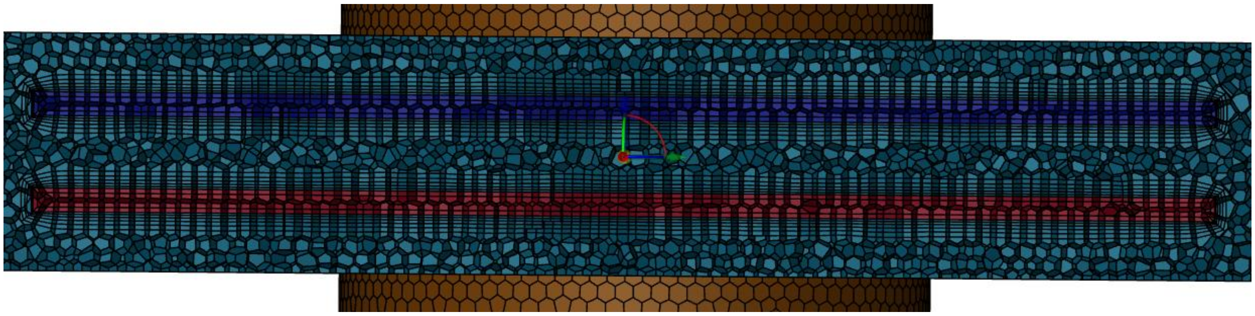


Figure 7: Boundary layers surrounding each fin

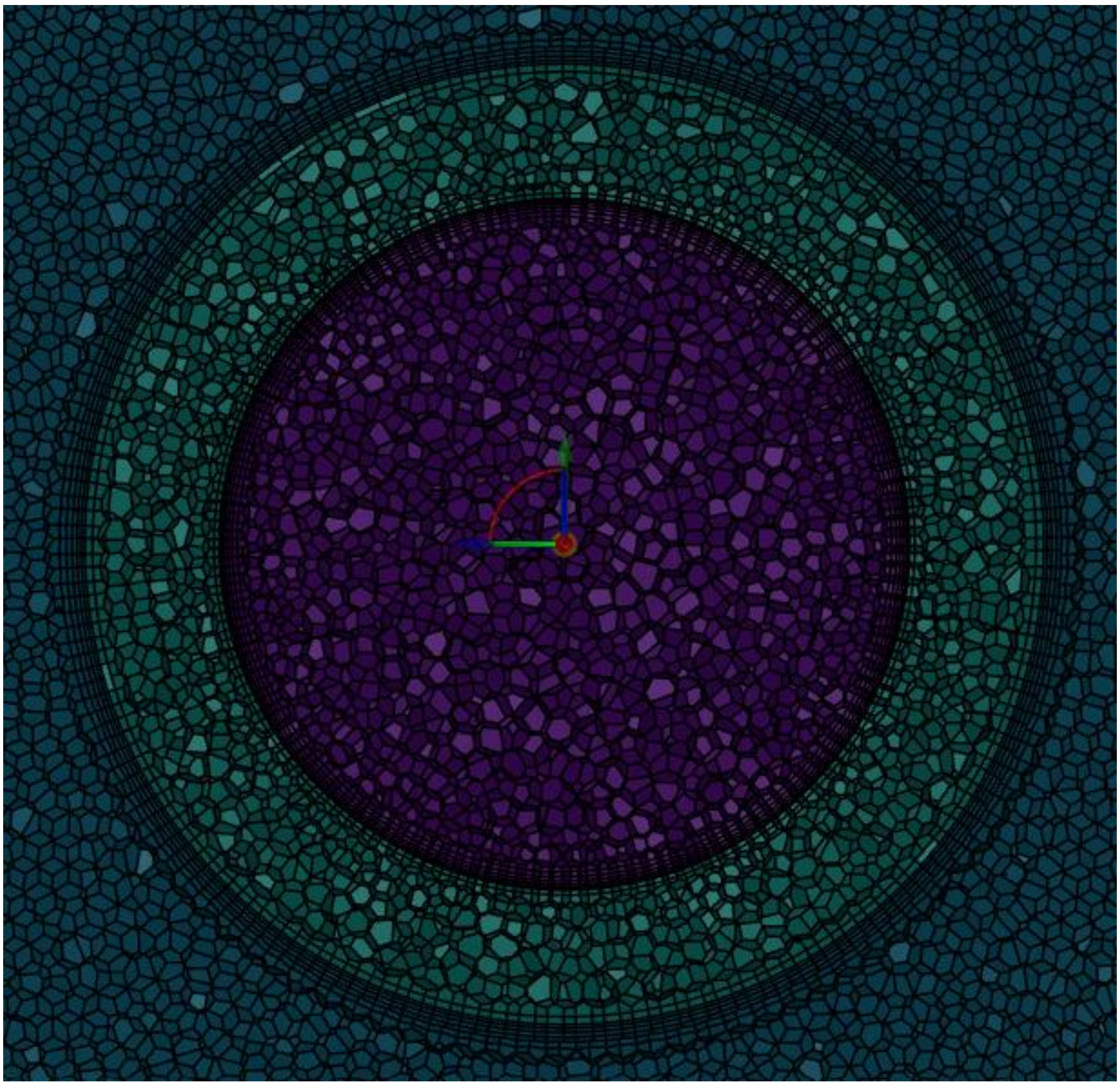


Figure 8: Boundary layers surrounding the tube

Boundary Conditions and Solver Settings:

This model uses a unique mass flow inlet and pressure outlet for both the air and oil regions. The left, right, top, and bottom boundaries of the air are treated as symmetry conditions. All other boundaries are treated as non-slip walls. For all cases, the energy equation is enabled to simulate heat transfer between the two working fluids, as well as the k-ω SST turbulence model. All simulations are allowed to run for 1,000 iterations using the SIMPLE solver with hybrid initialization. Convergence is monitored through monitors for mass flow rate difference between inlet and outlet for air and oil and temperatures at both the air and oil outlets. A summary of the boundary conditions and solver settings is provided in the table below:

Table 9: Two Fin Model Boundary Conditions and Solver Settings

Parameter	Value
Air mass flow rate	0.003671 lb/s
Air inlet/outlet turbulent intensity	4.98%
Air inlet/outlet turbulent length scale	0.000694 ft
Air inlet temperature	108 °F
Oil mass flow rate	0.6352 lb/s
Oil inlet/outlet turbulent intensity	6.07%
Oil inlet/outlet turbulent length scale	0.00075 ft
Oil inlet temperature	350 °F
Air outlet gauge pressure	0 psi
Oil outlet gauge pressure	0 psi
Gradient discretization method	Least Squares Cell Based
Pressure discretization method	Second Order
Momentum discretization method	Second Order Upwind
Turbulent kinetic energy discretization method	Second Order Upwind
Specific dissipation rate discretization method	Second Order Upwind
Energy discretization method	Second Order Upwind

Results:

After 1,000 iterations, all cases reach convergence. Both outlet temperatures are stable, and residuals drop sufficiently.

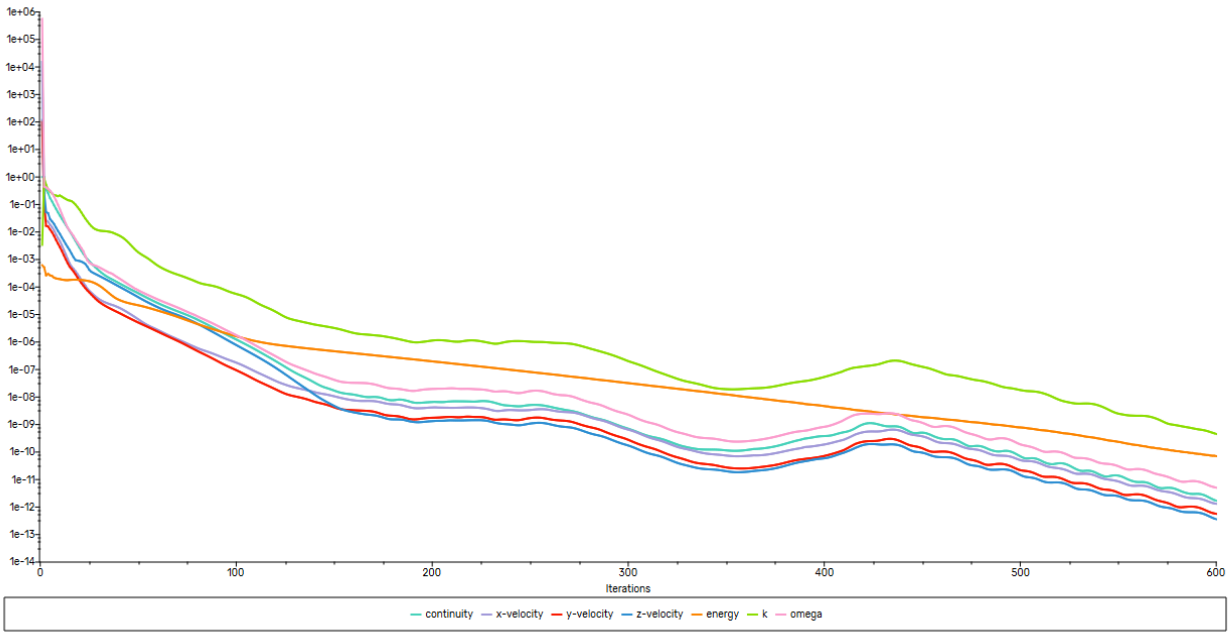


Figure 9: Plot of residuals

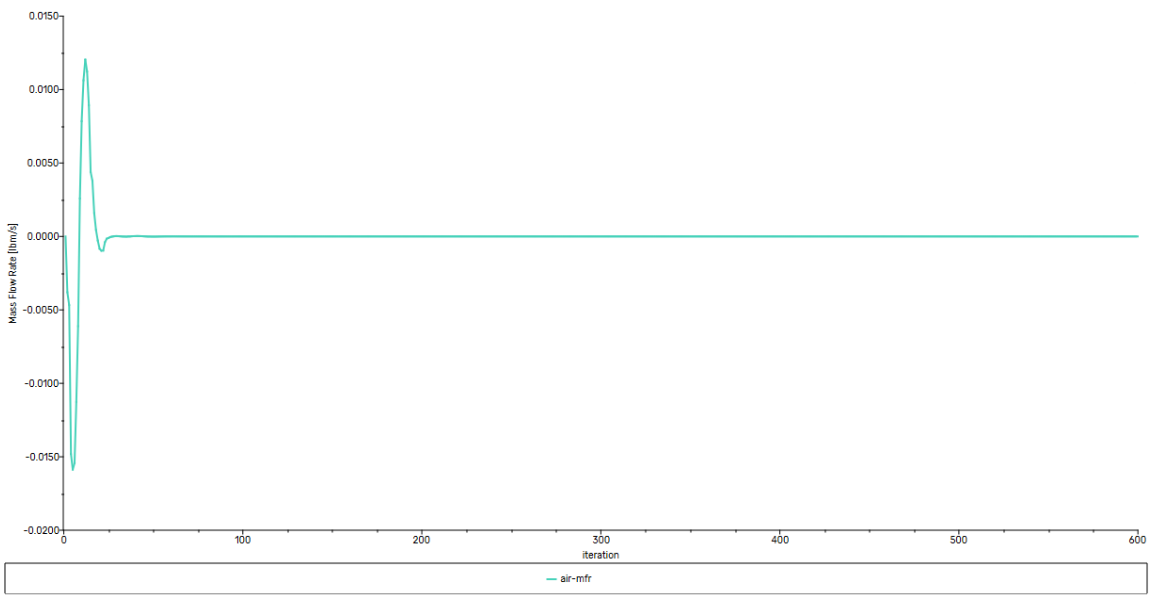


Figure 10: Convergence history for air mass flow rate

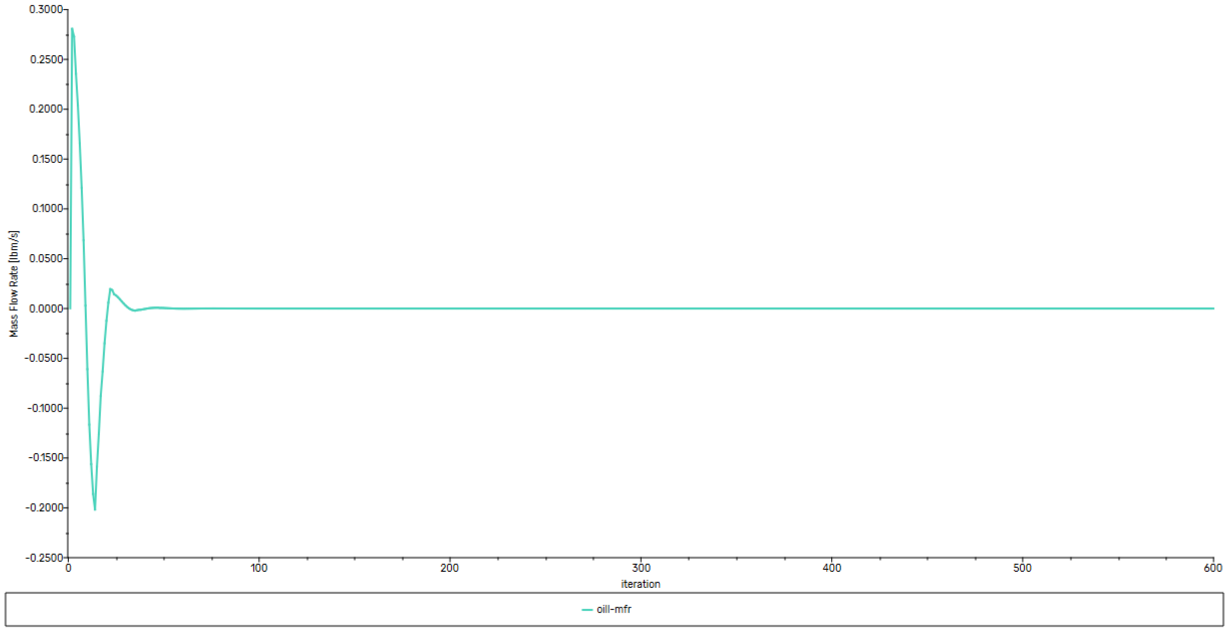


Figure 11: Convergence history for oil mass flow rate

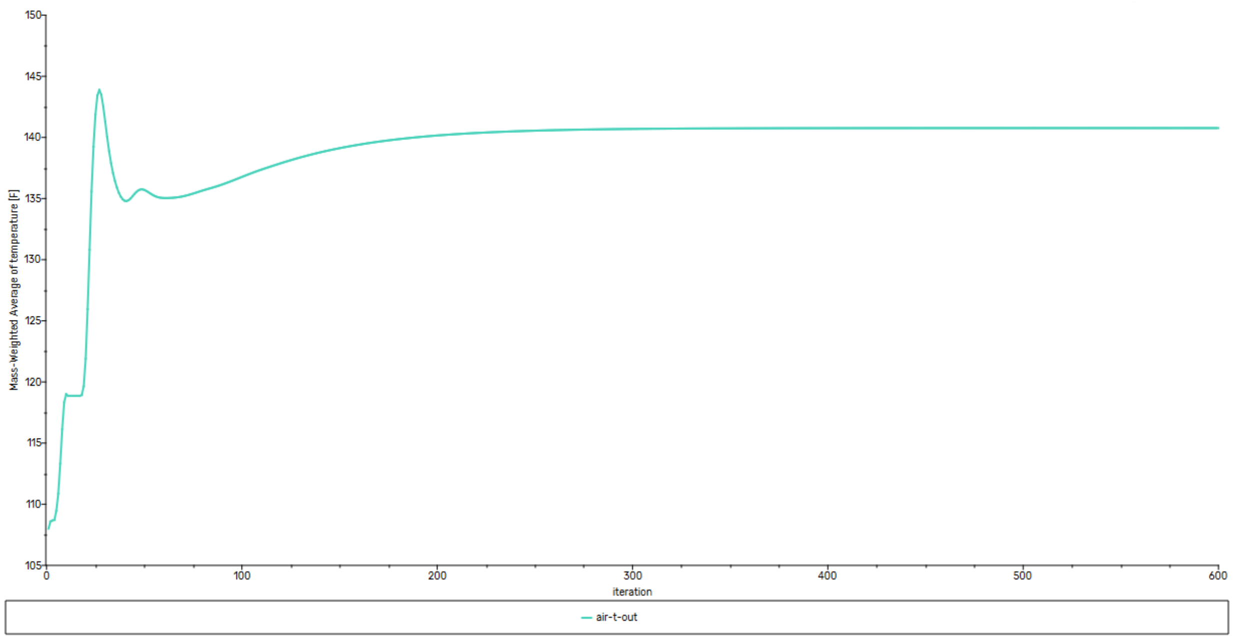


Figure 12: Convergence history for air outlet mass-weighted average temperature

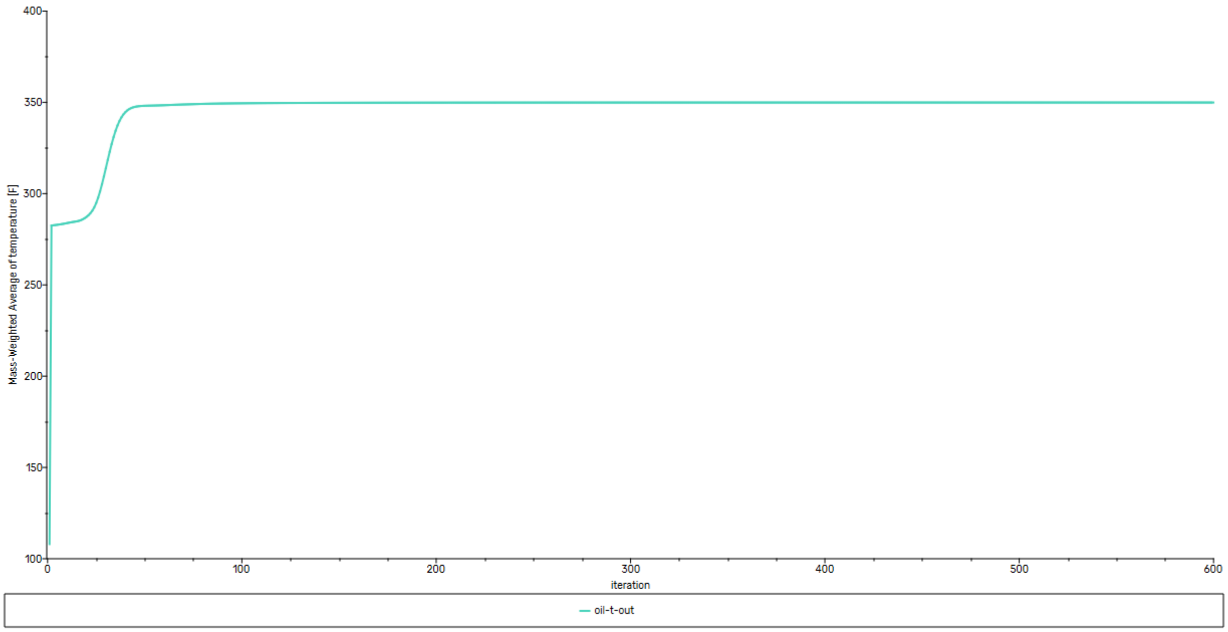


Figure 13: Convergence history for oil outlet mass-weighted average temperature

One assumption made in the preliminary design hand calculations above was that the minimum distance between fins was equal to exactly twice the thermal boundary layer at the end of the fins. This assumption can be validated in simulation if the thermal profile is symmetric about the midplane of the model and if the temperature profile comes to a minimum point at this midplane; this signifies that the thermal boundary layers have converged at the end of the fin and that the fin is not too short or too long such that the thermal profile looks "flat" near the midplane. To obtain the temperature profile at the end of the fin, a line probe is used at a height above the tube so the thermal influence of the tube is not taken into account. The location of this line probe, along with the temperature profile along this line, are shown in the figures below.

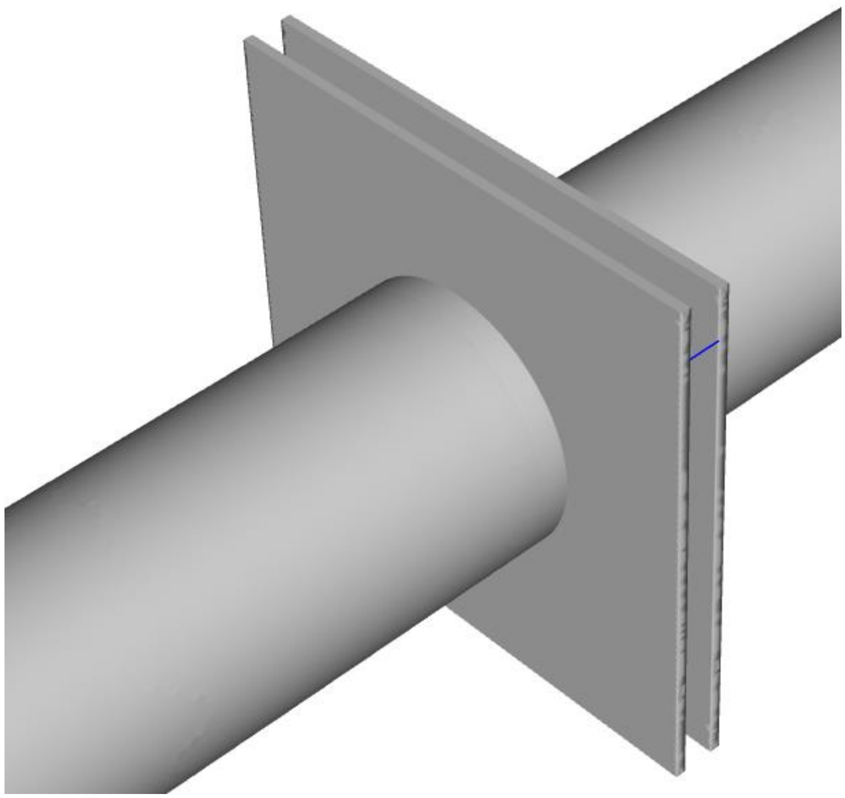


Figure 14: Location of line probe to measure temperature (blue line)

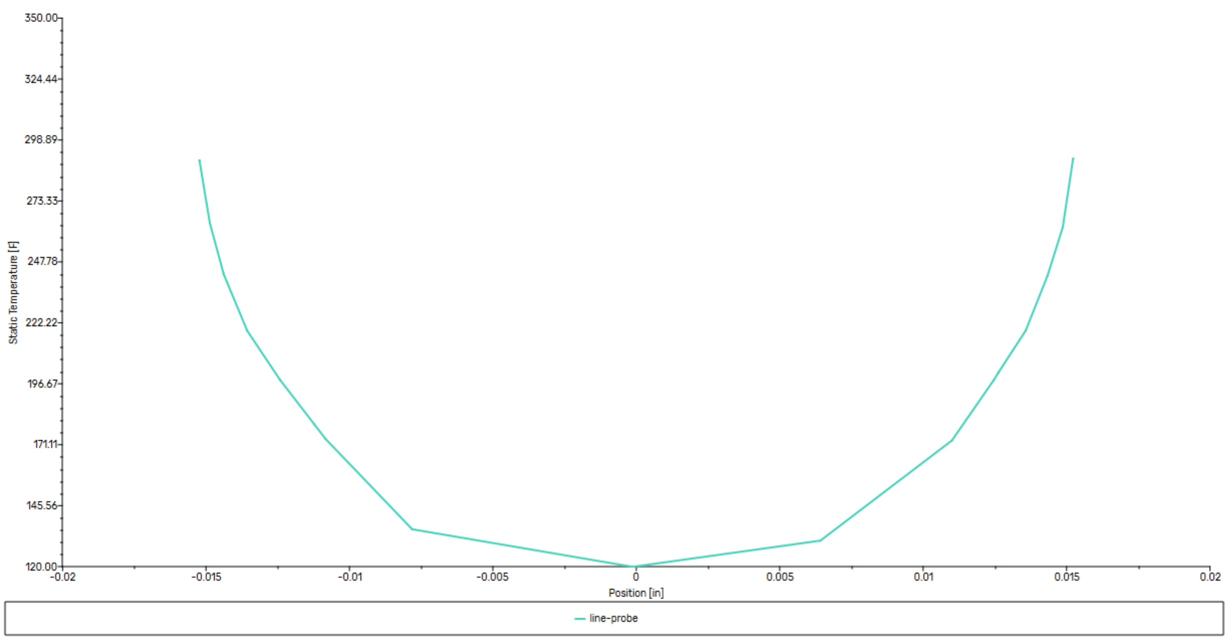


Figure 15: Temperature profile along line probe

The symmetry and concavity of the temperature profile is in good agreement with the hand calculation assumptions, validating the calculation for fin spacing. Other results for velocity distribution, temperature distribution, and pressure drop are shown in the table and figures below.

Table 10: Two Fin Model Results Summary

Parameter	Value
Air outlet temperature	140.77 °F
Oil outlet temperature	349.91 °F
Air pressure drop	0.781 psi
Oil pressure drop	28 psi

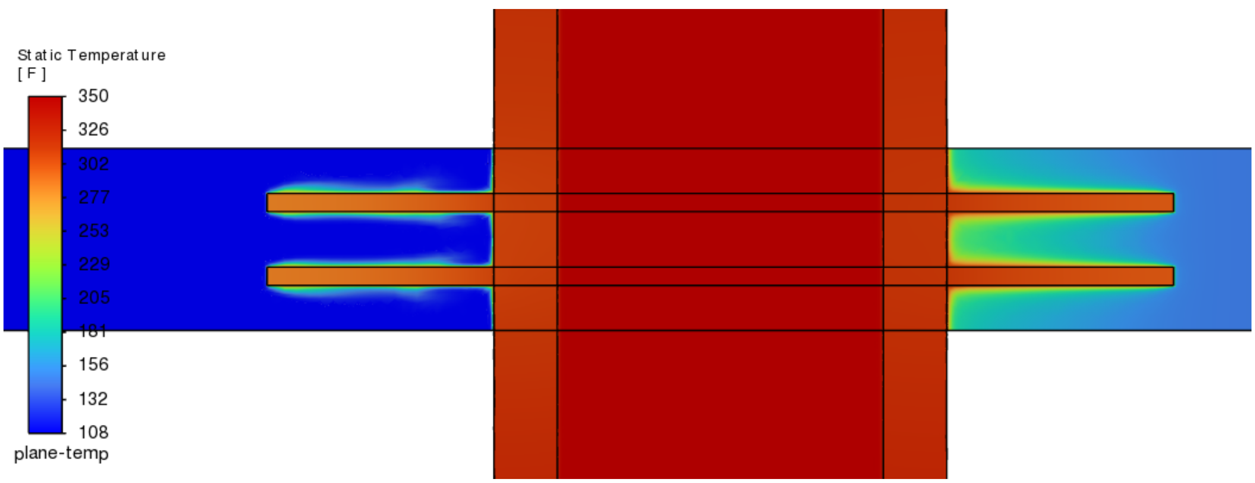


Figure 16: Midplane temperature distribution

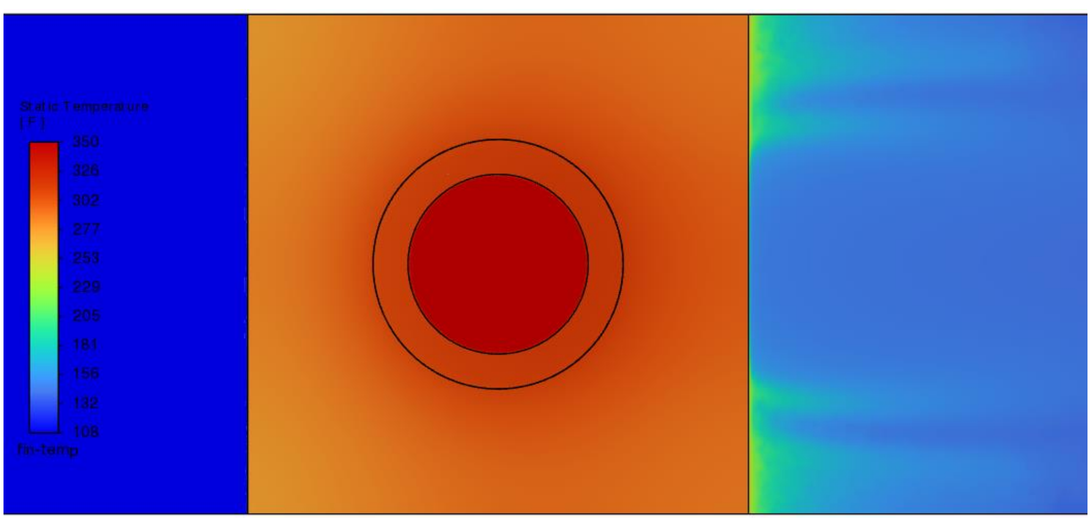


Figure 17: Air temperature distribution (including fin)

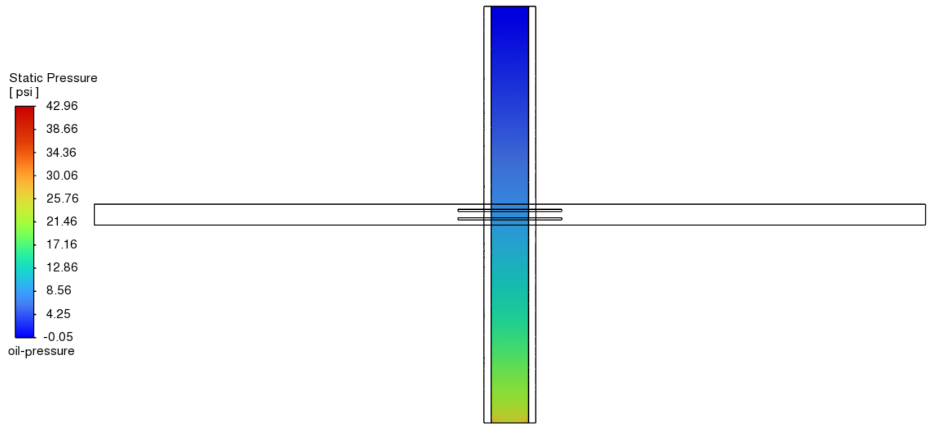


Figure 18: Oil pressure distribution

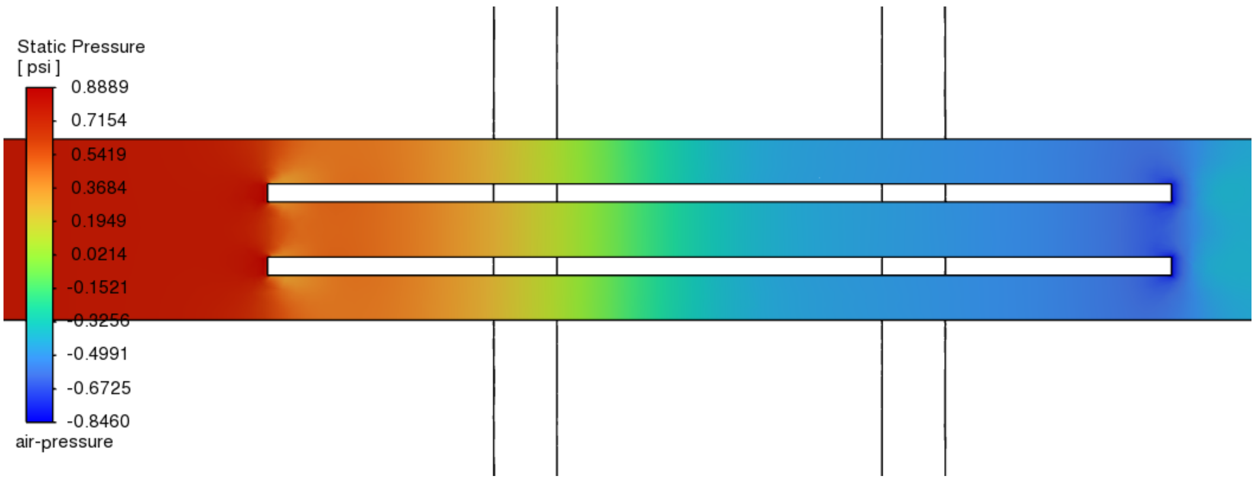


Figure 19: Air pressure distribution

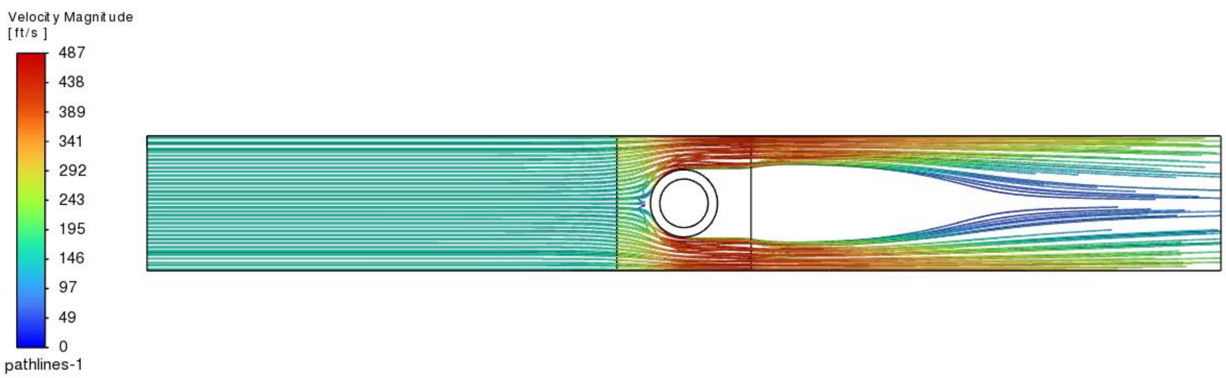


Figure 20: Air velocity streamlines

Hand Calculations for Validation:

To confirm the accuracy of this simulation for the short segment modeled, we provide additional hand calculations for outlet temperature and pressure drop for oil and air.

Since both the outlet temperatures are unknown, we opted to use effectiveness-NTU calculations for this model. Knowing both mass flow rates and specific heats for both the oil and air, we can define an NTU as:

$$NTU = \frac{U_i A_i}{(\dot{m}c_p)_{min}} = 0.127$$

Similarly, we can define the heat capacity ratio as:

$$C_r = \frac{(\dot{m}c_p)_{min}}{(\dot{m}c_p)_{max}} = 0.00277$$

These parameters can be used to calculate the heat exchanger effectiveness directly [3] as:

$$\epsilon = 1 - e^{\frac{NTU^{0.22}}{C_r}(e^{-C_r NTU^{0.78}} - 1)} = 0.119$$

With this, the outlet temperatures can be calculated:

$$T_{oil,exit} = T_{oil,entrance} - \frac{(\dot{m}c_p)_{min}(T_{oil,entrance} - T_{air,entrance})\epsilon}{(\dot{m}c_p)_{oil}} = 349.92^{\circ}F$$

$$T_{air,exit} = T_{air,entrance} + \frac{(\dot{m}c_p)_{min}(T_{oil,entrance} - T_{air,entrance})\epsilon}{(\dot{m}c_p)_{air}} = 136.87^{\circ}F$$

The pressure drops for oil and air can be calculated similarly to how they were calculated in the *Design Hand Calculations* section above:

$$\Delta P_{oil} = \gamma f \frac{L}{D} \frac{V^2}{2g} = 11.26 psi$$

$$\Delta P_{air} = N_L f \chi \frac{\rho V_{max}^2}{2} = 0.2 psi$$

Results are summarized and compared to simulation results in the table below:

Table 11: Two Fin Model and Calculation Comparison

Parameter	**Simulation Value**	**Hand Calculation Value**
Air outlet temperature	140.77 °F	136.87 °F
Oil outlet temperature	349.91 °F	349.92 °F
Air pressure drop	0.781 psi	0.2 psi
Oil pressure drop	28 psi	11.26 psi

Heat Exchanger Design: Tube Bank Model

Motivation:
This section expands the two-fin model by adding a second tube with a bend. The purpose of this model is to validate the pressure drop along tube bend and spacing between adjacent tubes.

Geometry and Mesh:
To maximize heat exchanger compactness, a staggered tube bank has been chosen. This arrangement has a relatively high packing factor compared to the alternative unstaggered arrangement. The distance between tubes is 0.5 in. vertically and horizontally. To fit the geometry of the duct outlet, a total of 8 tubes per air pass are able to be placed, fully characterizing the length and width of the heat exchanger. If the tubes thermally interact with each other similarly between air passes, the total temperature drop for oil per air pass can be estimated through a simple log mean temperature difference calculation. Note that to fit both tubes in the model while maintaining symmetry boundary conditions on all air boundaries, the air height must be doubled.

The geometry for this model follows directly from the geometry of the two-fin model; aside from another tube being added to simulate the addition of a banked tube to the system and using only the ideal geometry and spacing assumed in the previous Heat Exchanger Design sections, the model is unchanged. A summary of meshing parameters and the resulting geometry is shown below:

Table 12: Tube Bank Model Meshing Parameters

Meshing Parameter	Value
Fin local sizing	0.02 in.
Tube local sizing	0.04 in.
Oil local sizing	0.005 in.
Air local sizing	0.03 in.
Air refinement region near tube	0.0075 in.
Number of air boundary layers	5
Number of solid boundary layers	1
Volume mesh size	0.03 in.
Volume mesh growth rate	1.1
Total number of cells	3,226,130

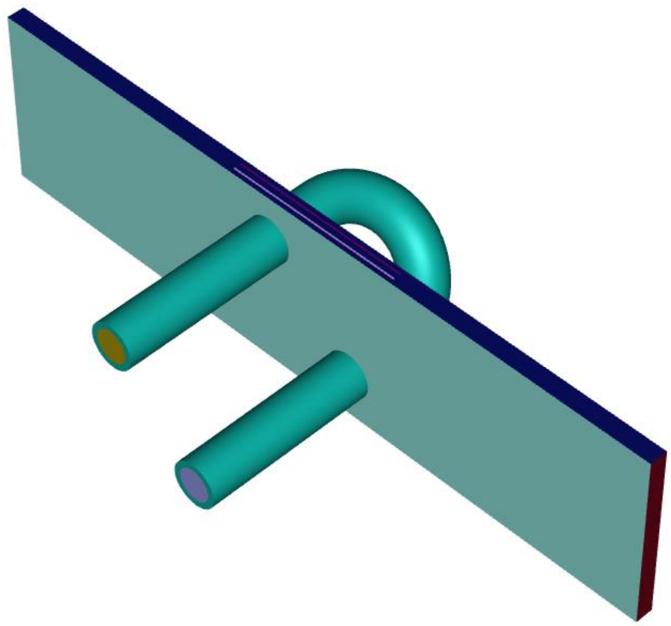


Figure 21: Geometry used for tube bank model (isometric view)

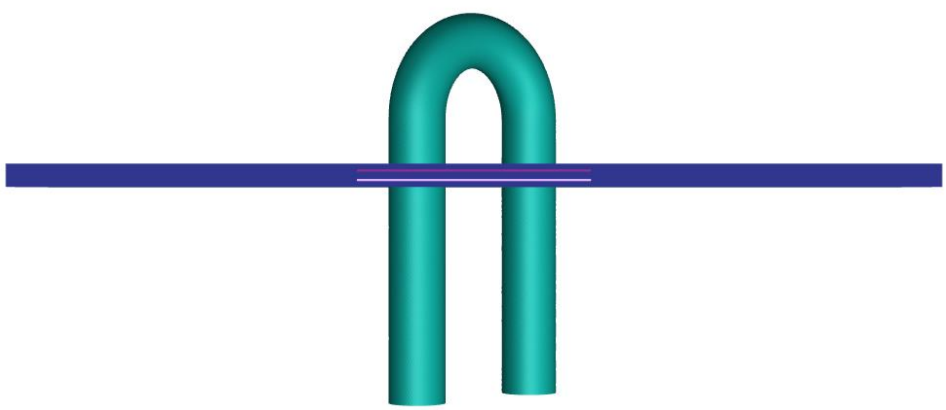


Figure 22: Geometry used for tube bank model (top view)

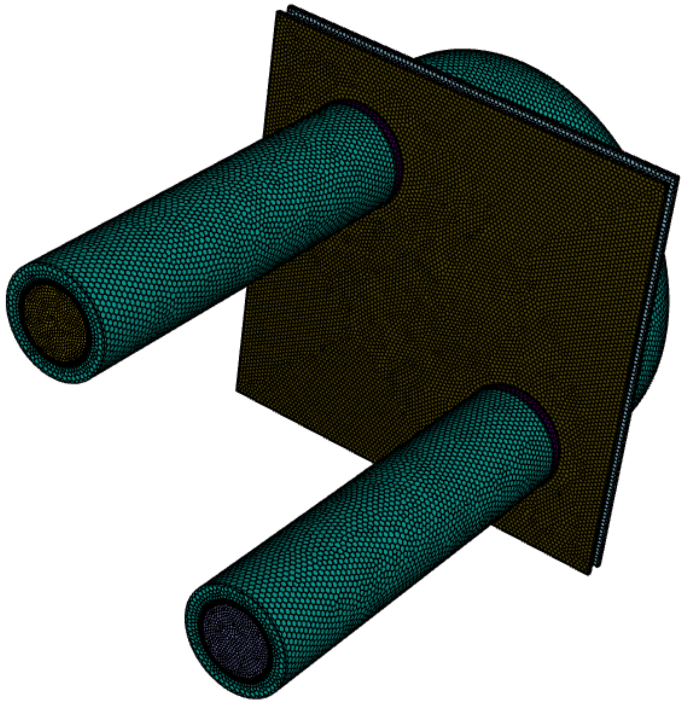


Figure 23: Mesh used for tube bank model

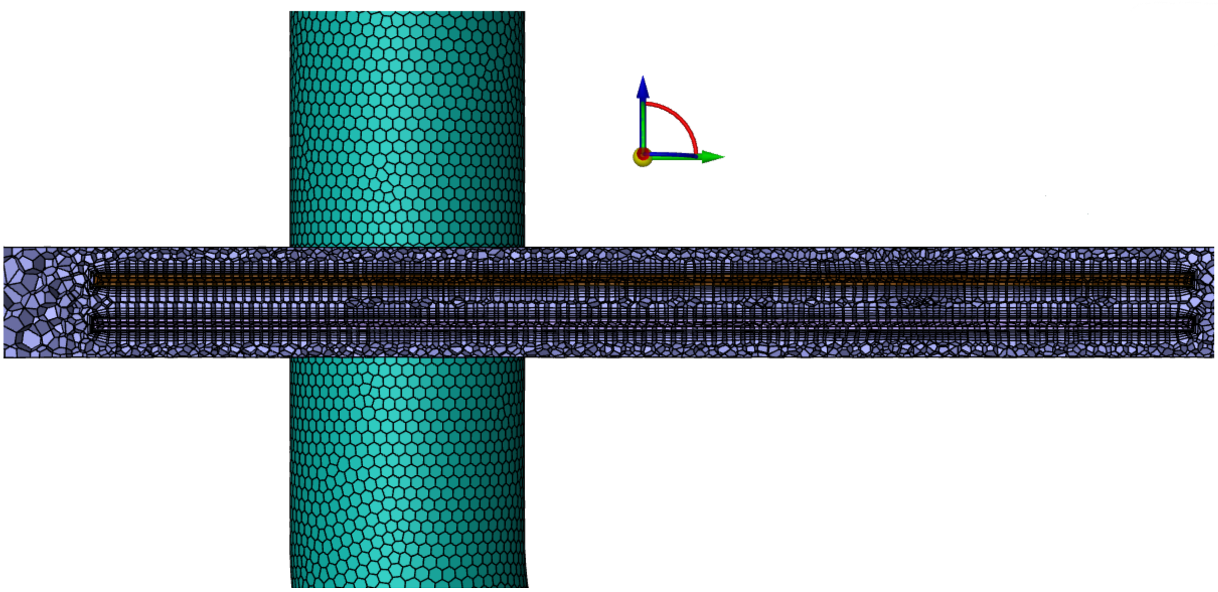


Figure 24: Boundary layers surrounding each fin

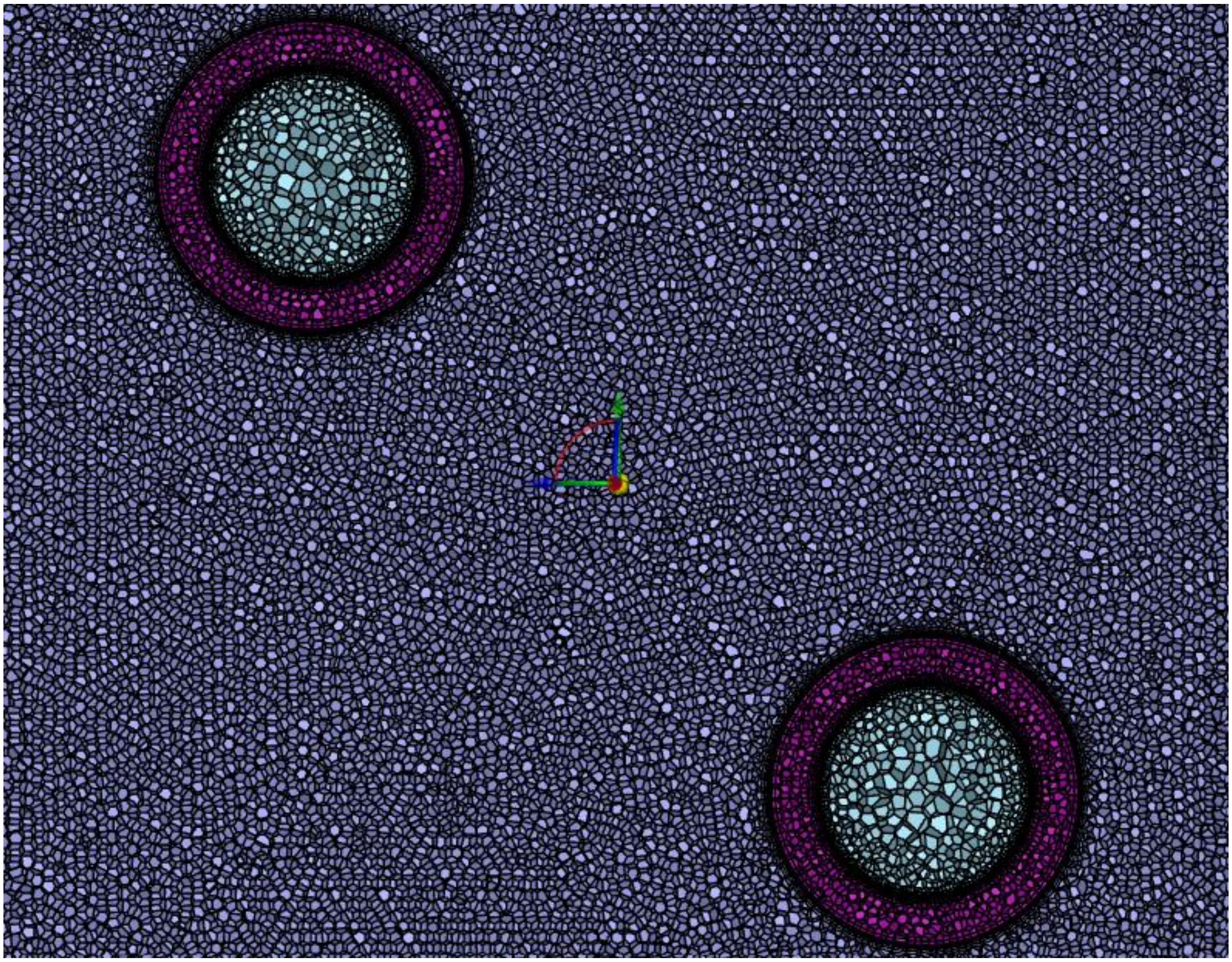


Figure 25: Boundary layers surrounding the tubes

Boundary Conditions and Solver Settings:

This model uses a unique mass flow inlet and pressure outlet for both the air and oil regions. The left, right, top, and bottom boundaries of the air are treated as symmetry conditions. All other boundaries are treated as non-slip walls. For all cases, the energy equation is enabled to simulate heat transfer between the two working fluids, as well as the k-ω SST turbulence model. All simulations are allowed to run for 1,000 iterations using the SIMPLE solver with hybrid initialization. Convergence is monitored through monitors for mass flow rate difference between inlet and outlet for air and oil and temperatures at both the air and oil outlets. A summary of the boundary conditions and solver settings is provided in the table below:

Table 13: Two Fin Model Boundary Conditions and Solver Settings

Parameter	Value
Air mass flow rate	0.007341 lb/s
Air inlet/outlet turbulent intensity	4.93%
Air inlet/outlet turbulent length scale	0.000758 ft
Air inlet temperature	108 °F
Oil mass flow rate	0.6352 lb/s
Oil inlet/outlet turbulent intensity	6.07%
Oil inlet/outlet turbulent length scale	0.00075 ft
Oil inlet temperature	350 °F
Air outlet gauge pressure	0 psi
Oil outlet gauge pressure	0 psi
Gradient discretization method	Least Squares Cell Based
Pressure discretization method	Second Order
Momentum discretization method	Second Order Upwind
Turbulent kinetic energy discretization method	Second Order Upwind
Specific dissipation rate discretization method	Second Order Upwind
Energy discretization method	Second Order Upwind

Results:

After 800 iterations, all monitors converge. Residuals drop sufficiently and the air and oil outlet temperature monitors converge. It is critical that these monitors reach convergence, as the outlet temperatures are used to evaluate the performance of the heat exchanger. Sample plots of the residuals and convergence histories are shown below for the model.

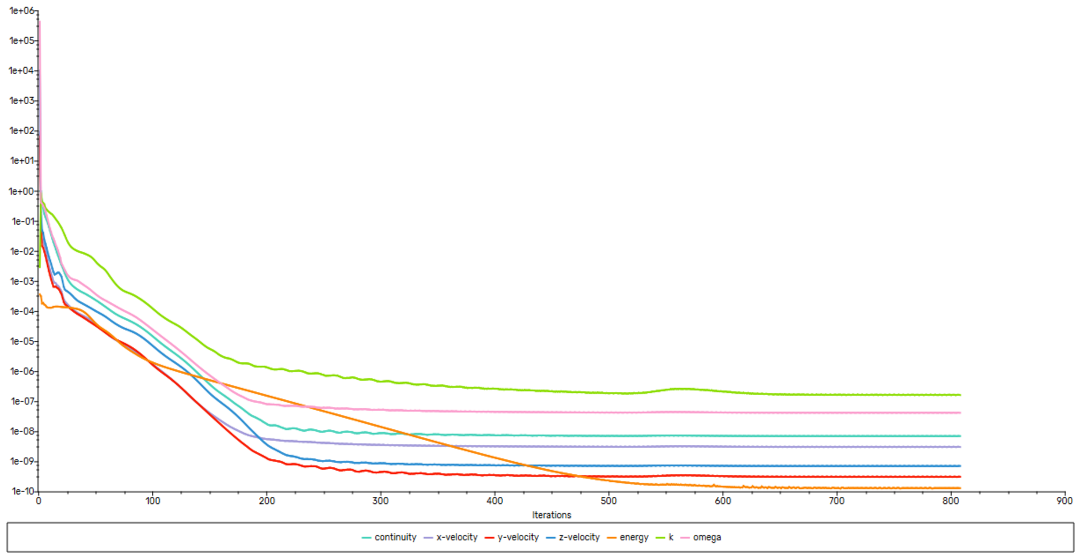


Figure 26: Plot of residuals

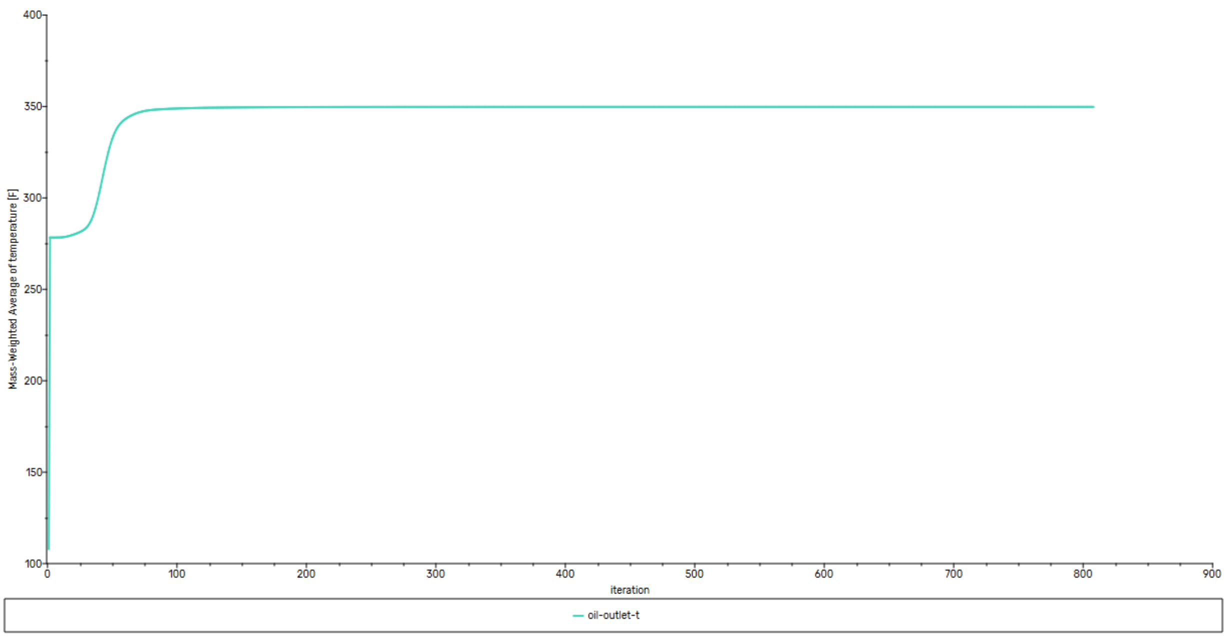


Figure 27: Convergence history for oil mass-weighted average temperature

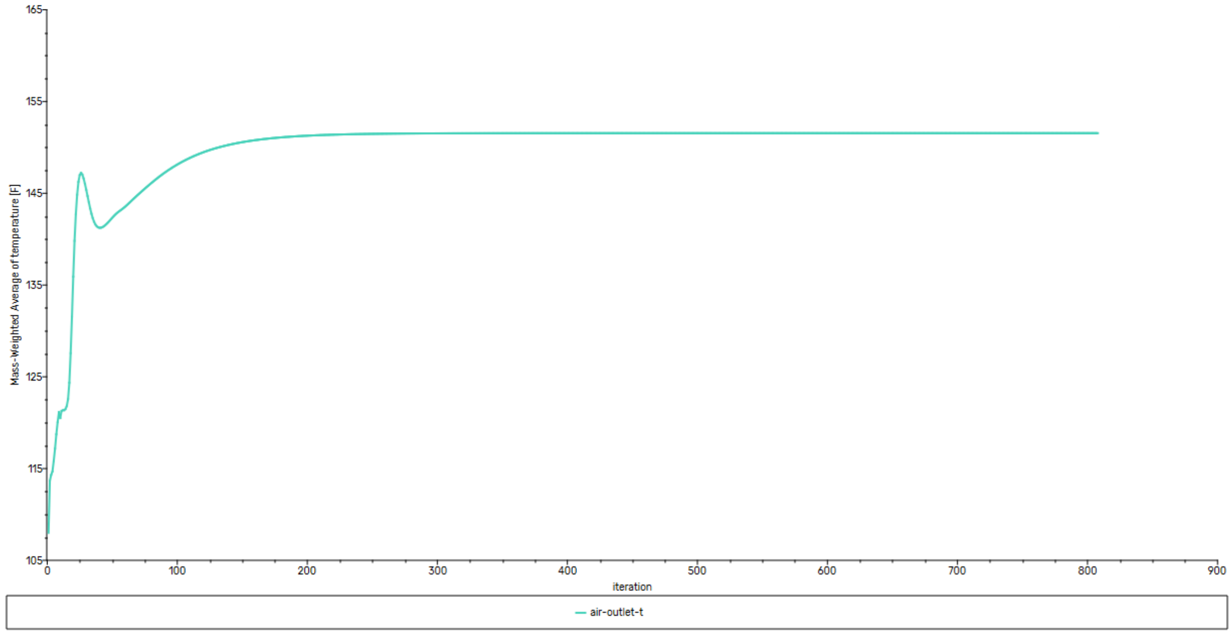


Figure 28: Convergence history for air mass-weighted average temperature

Results for velocity distribution, temperature distribution, and pressure drop are shown in the table and figures below.

Table 14: Tube Bank Model Results Summary

Parameter	**Value**
Air outlet temperature	151.55 °F
Oil outlet temperature	349.76 °F
Air pressure drop	0.693 psi
Oil overall pressure drop	53.5 psi
Oil tube bend pressure drop	25.89 psi

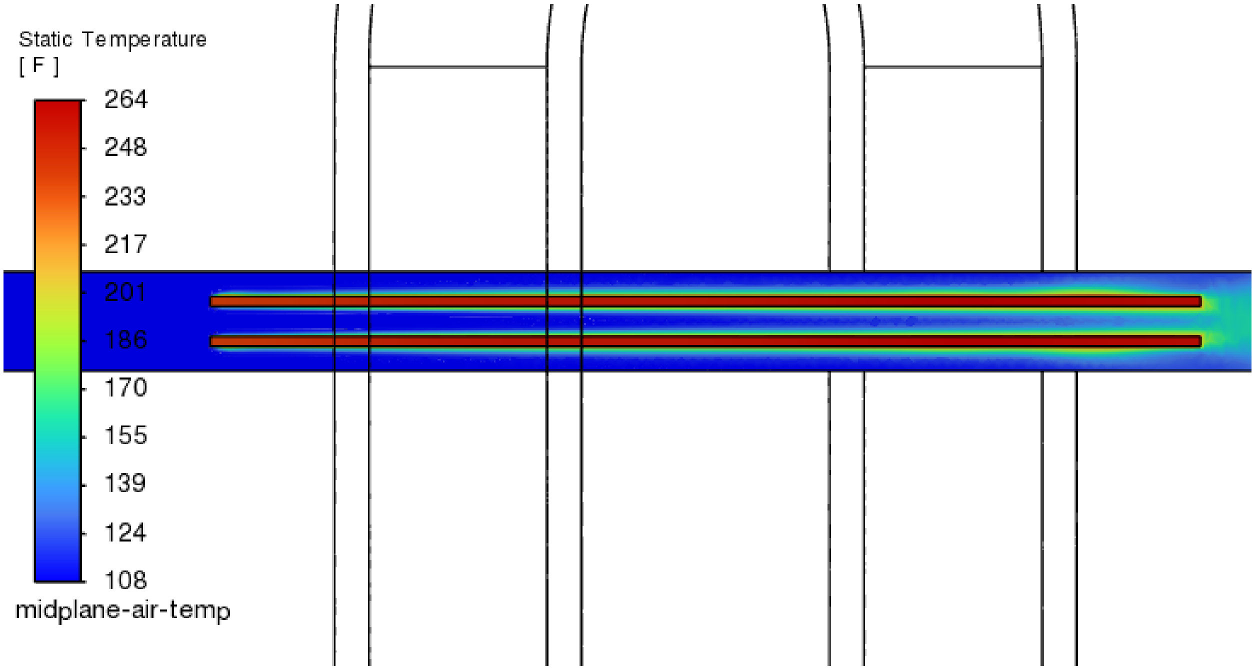


Figure 29: Midplane temperature distribution

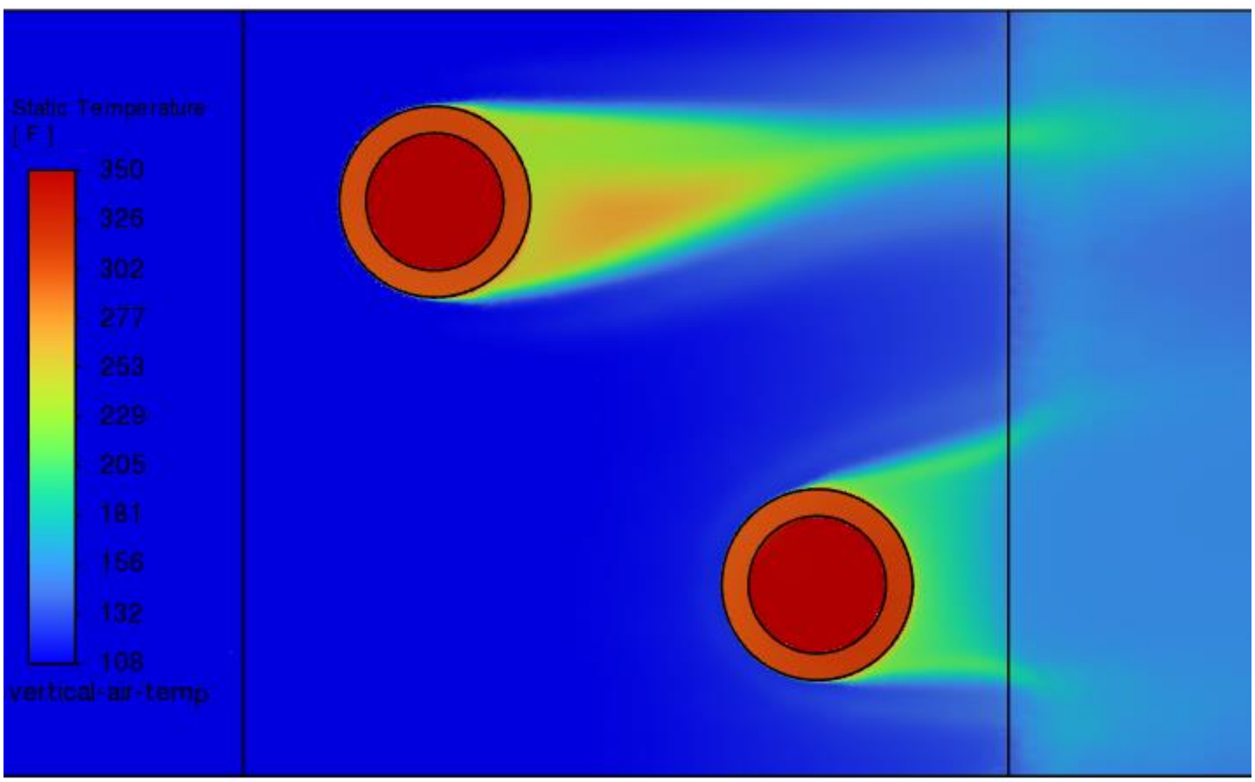


Figure 30: Midplane temperature distribution (between fins)

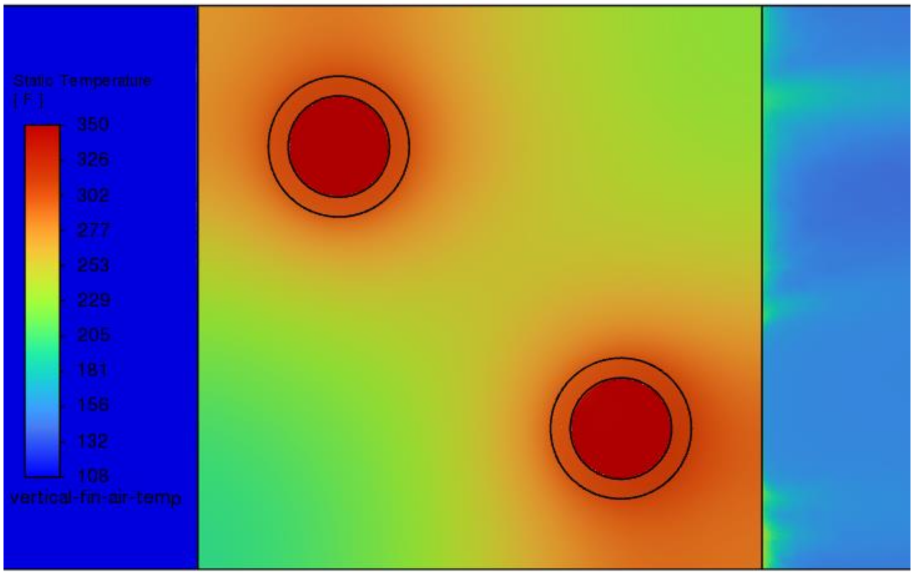


Figure 31: Midplane temperature distribution (at fin)

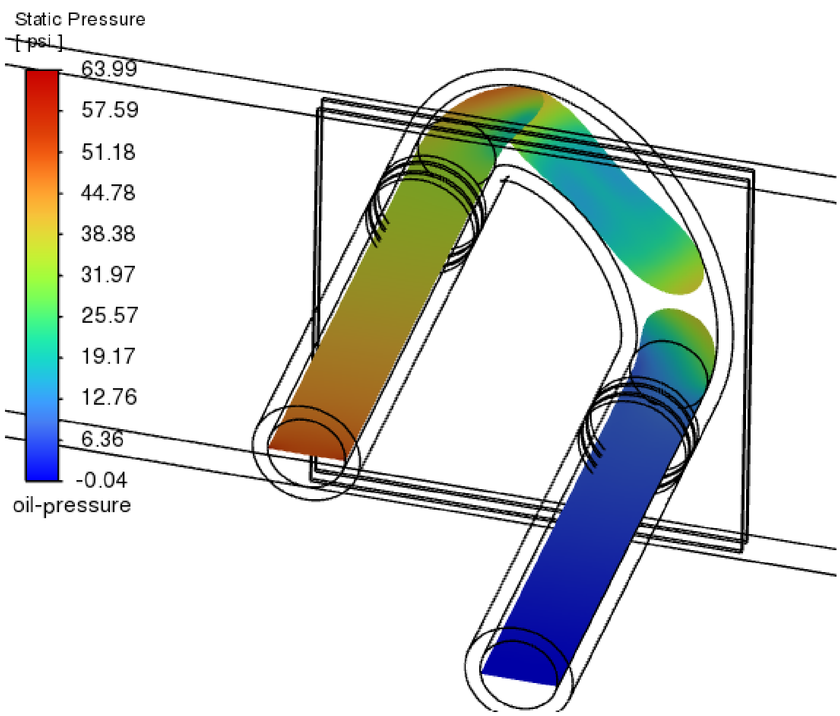


Figure 32: Oil pressure distribution

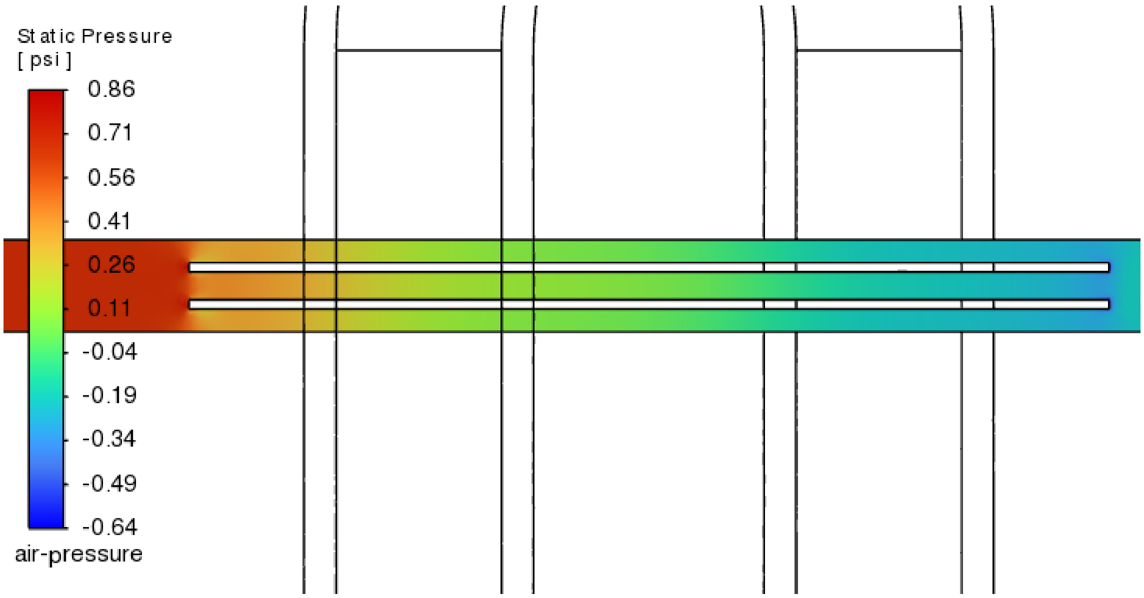


Figure 33: Air pressure distribution

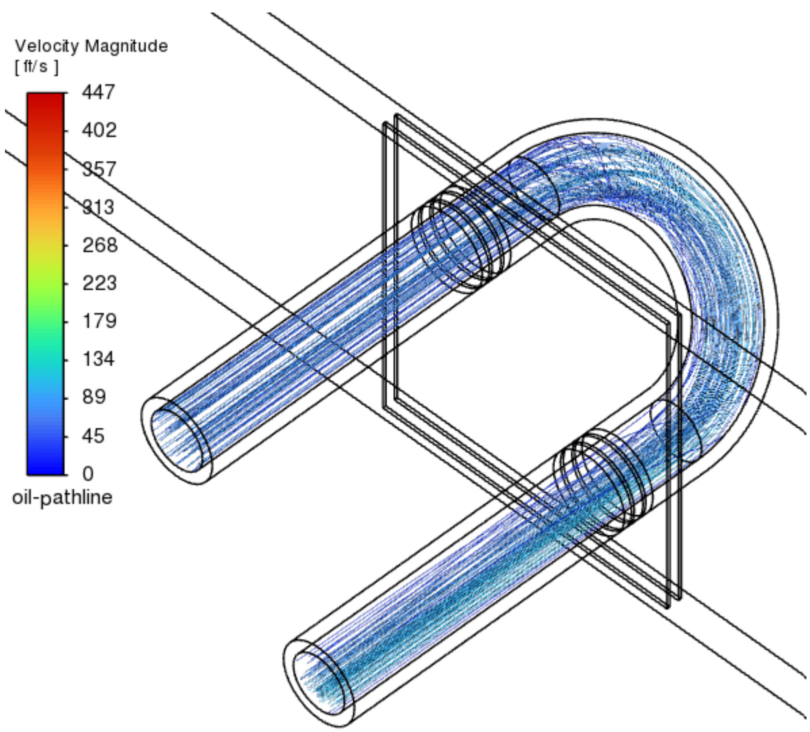


Figure 34: Oil velocity streamline

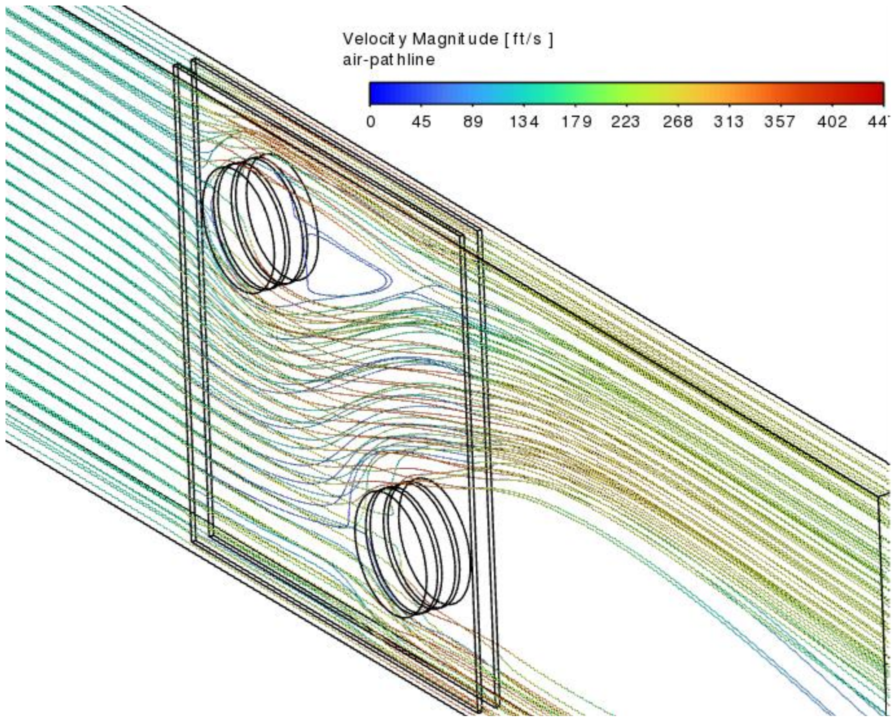


Figure 35: Air velocity streamline

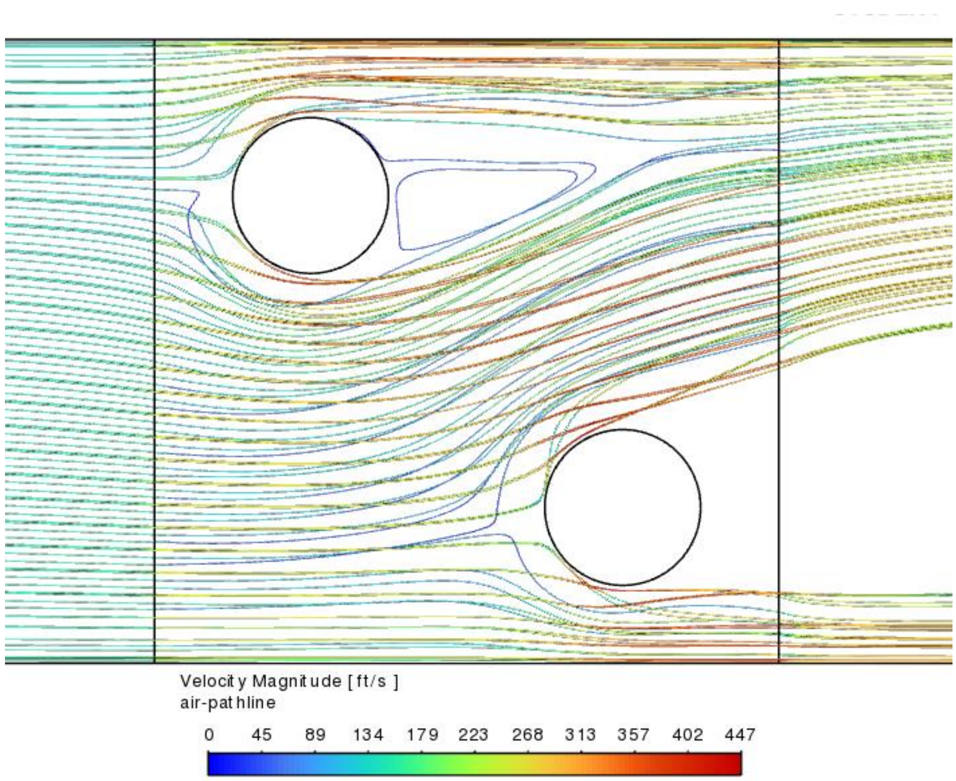


Figure 36: Air velocity streamline

Hand Calculations for Validation:
To confirm the accuracy of this simulation for the short segment modeled, we provide additional hand calculations for outlet temperature and pressure drop for oil and air.

Since both the outlet temperatures are unknown, we opted to use effectiveness-NTU calculations for this model. Knowing both mass flow rates and specific heats for both the oil and air, we can define an NTU as:

$$NTU = \frac{U_i A_i}{(\dot{m}c_p)_{min}} = 0.233$$

Similarly, we can define the heat capacity ratio as:

$$C_r = \frac{(\dot{m}c_p)_{min}}{(\dot{m}c_p)_{max}} = 0.00555$$

These parameters can be used to calculate the heat exchanger effectiveness directly [3] as:

$$\epsilon = 1 - e^{\frac{NTU^{0.22}}{C_r}(e^{-C_r NTU^{0.78}} - 1)} = 0.119$$

With this, the outlet temperatures can be calculated:

$$T_{oil,exit} = T_{oil,entrance} - \frac{(\dot{m}c_p)_{min}(T_{oil,entrance} - T_{air,entrance})\epsilon}{(\dot{m}c_p)_{oil}} = 349.77^\circ F$$

$$T_{air,exit} = T_{air,entrance} + \frac{(\dot{m}c_p)_{min}(T_{oil,entrance} - T_{air,entrance})\epsilon}{(\dot{m}c_p)_{air}} = 149.97^\circ F$$

The pressure drops for oil and air can be calculated similarly to how they were calculated in the *Design Hand Calculations* section above, except there is now a K factor for the tube bend to account for. This K factor is determined using a hydraulics lookup table to be 1.2 [3].

$$\Delta P_{oil} = \gamma(f\frac{L}{D} + K)\frac{V^2}{2g} = 53.66psi$$

$$\Delta P_{air} = N_L f \chi \frac{\rho V_{max}^2}{2} = 0.4psi$$

Results are summarized and compared to simulation results in the table below:

Table 15: Tube Bank Model and Calculation Comparison

Parameter	Simulation Value	Hand Calculation Value
Air outlet temperature	151.55 °F	149.97 °F
Oil outlet temperature	349.76 °F	349.77 °F
Air pressure drop	0.693 psi	0.4 psi
Oil overall pressure drop	53.5 psi	53.68 psi

Heat Exchanger Design: Duct Design

Motivation:
To model the duct that brings outside air to the heat exchanger. The purpose is to determine the velocity profile of the heat exchanger inlet and also to verify the pressure drop of air.

Geometry and Mesh:
To ensure that the velocity is turbulent at the inlet of the heat exchanger, the geometry of the duct must be determined. As previously mentioned, the duct inlet size is 22 in. x 12 in. and velocity is 25 mph. The duct outlet area is chosen to result in a velocity that creates turbulent flow over the fins of the heat exchanger. The Reynolds number for external flow over a flat plate is used to determine the type of flow. This procedure is discussed in detail in the *Design Hand Calculations* section of the report. Using conservation of mass and the velocity necessary for turbulence, an outlet area can be calculated. Duct outlet dimensions of 8 in. x 8 in. are chosen. This area results in a similar velocity at the duct outlet to the velocity necessary for turbulence, as will be discussed below.

As required, the duct outlet must be 60” downstream of the inlet. In order to ensure that a fully developed flow profile reaches the heat exchanger, the first half of the duct length is used for the contraction and the second half of the length is kept at a constant area. The duct has a rectangular cross-section throughout its length, as it is the most convenient cross section for a fin and tube heat exchanger. Note that the inlet area at the contraction has been extended to ensure a fully developed profile throughout the duct. A summary of meshing parameters and the resulting geometry is shown below:

Table 16: Duct Model Meshing Parameters

Meshing Parameter	Value
Surface mesh size	0.4 in.
Number of air boundary layers	5
Volume mesh size	0.6 in.
Volume mesh growth rate	1.2
Total number of cells	343,666

Figure 37: Duct geometry

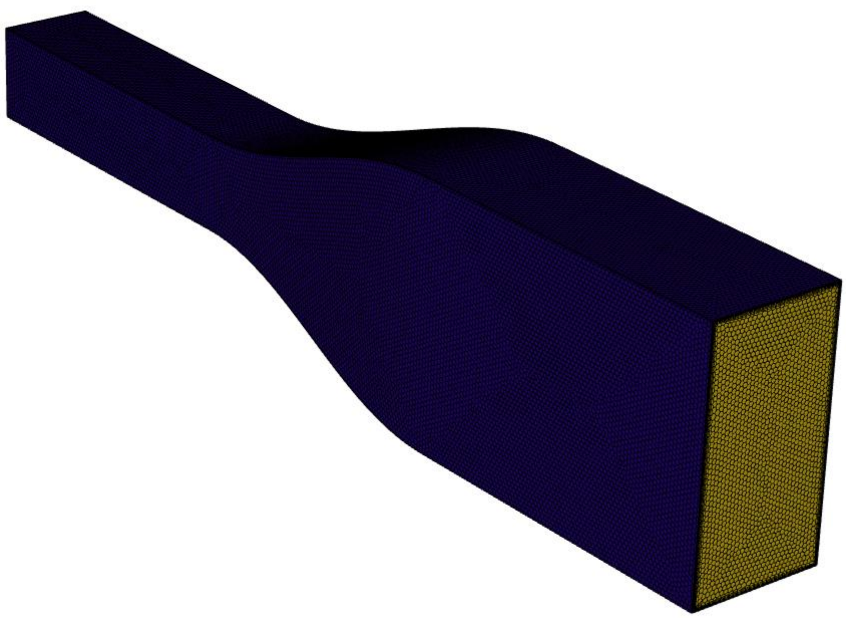


Figure 38: Duct mesh

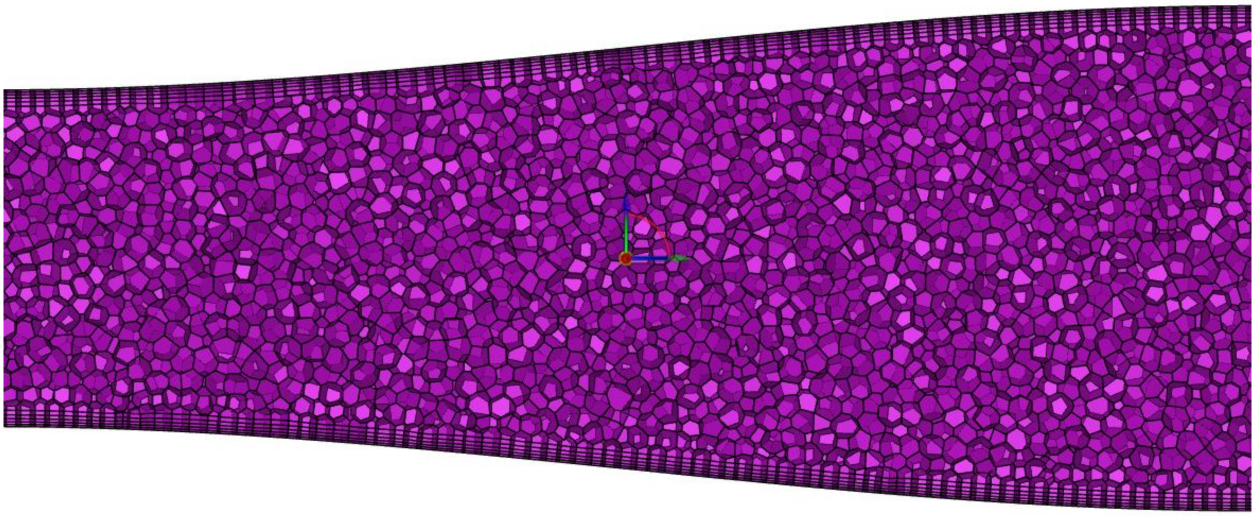


Figure 39: Duct mesh cross section

Boundary Conditions and Solver Settings:

This model uses a velocity inlet and pressure outlet for the air region. All other boundaries are treated as non-slip walls. The energy equation is disabled as heat transfer is ignored as air flows through the duct. The k-ω SST turbulence model is used. The simulation is allowed to run for 1,000 iterations using the SIMPLE solver with hybrid initialization. Convergence is monitored through monitors for mass flow rate difference between inlet and outlet and inlet pressure. A summary of the boundary conditions and solver settings is provided in the table below:

Table 17: Duct Model Boundary Conditions and Solver Settings

Parameter	Value
Air inlet velocity	25 mph
Air inlet turbulent intensity	3.37%
Air inlet turbulent length scale	0.0647 ft
Air outlet gauge pressure	0 psi
Air outlet turbulent intensity	3.66%
Air outlet turbulent length scale	0.0333 ft
Gradient discretization method	Least Squares Cell Based
Pressure discretization method	Second Order
Momentum discretization method	Second Order Upwind
Turbulent kinetic energy discretization method	Second Order Upwind
Specific dissipation rate discretization method	Second Order Upwind
Energy discretization method	Second Order Upwind

Results:

After 1,000 iterations, both residuals and convergence monitors indicate that the solution has reached convergence. Plots of residuals and the convergence monitors are reproduced below.

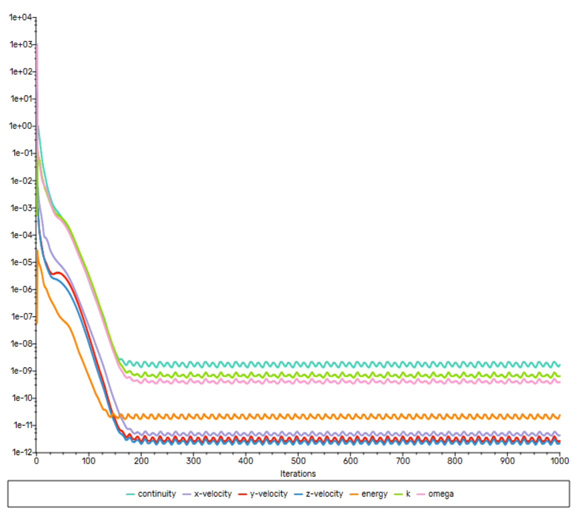


Figure 40: Plot of residuals

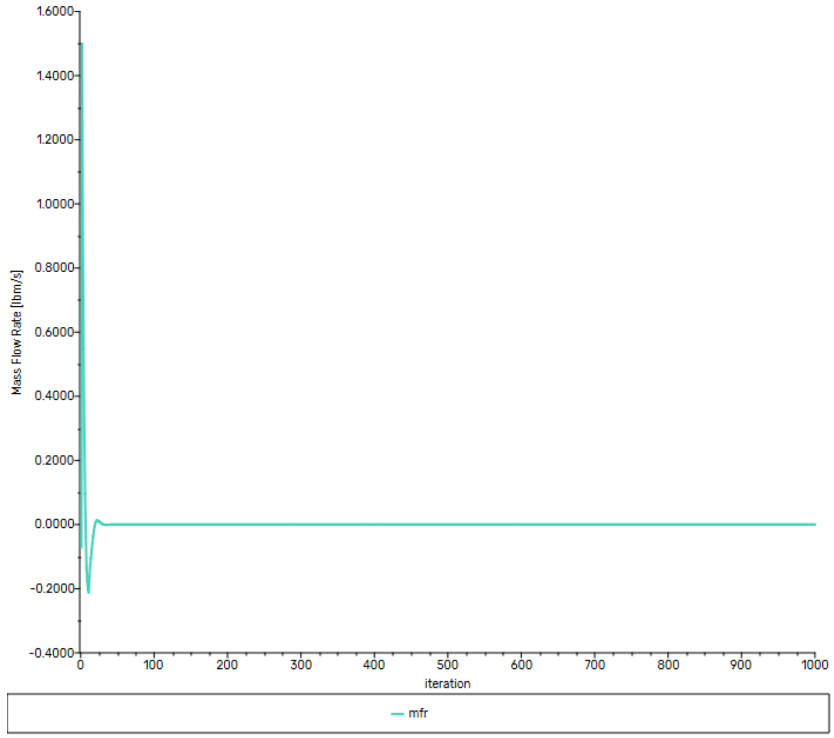


Figure 41: Convergence history for duct mass flow rate

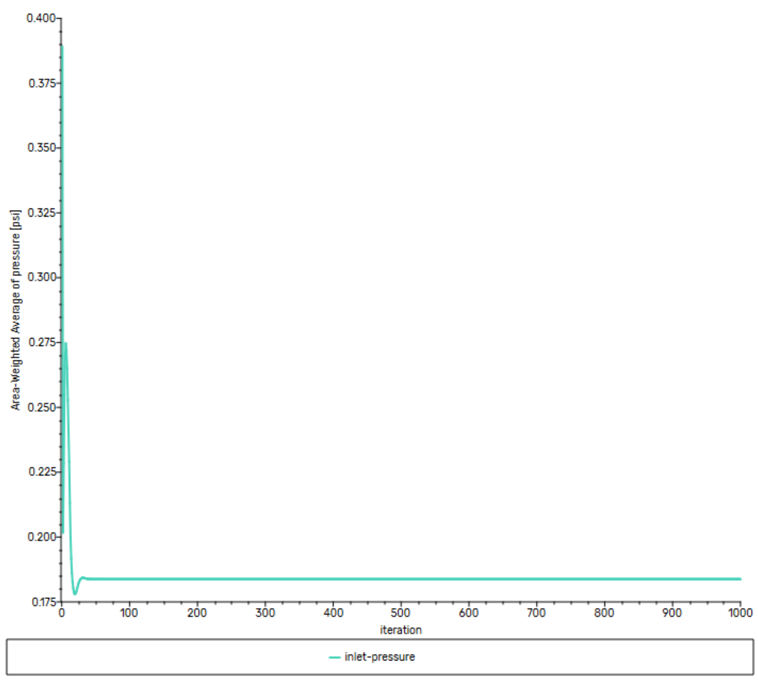


Figure 42: Convergence history for duct inlet pressure

To demonstrate how air flows through the device, velocity streamlines of air passing through the duct are provided below:

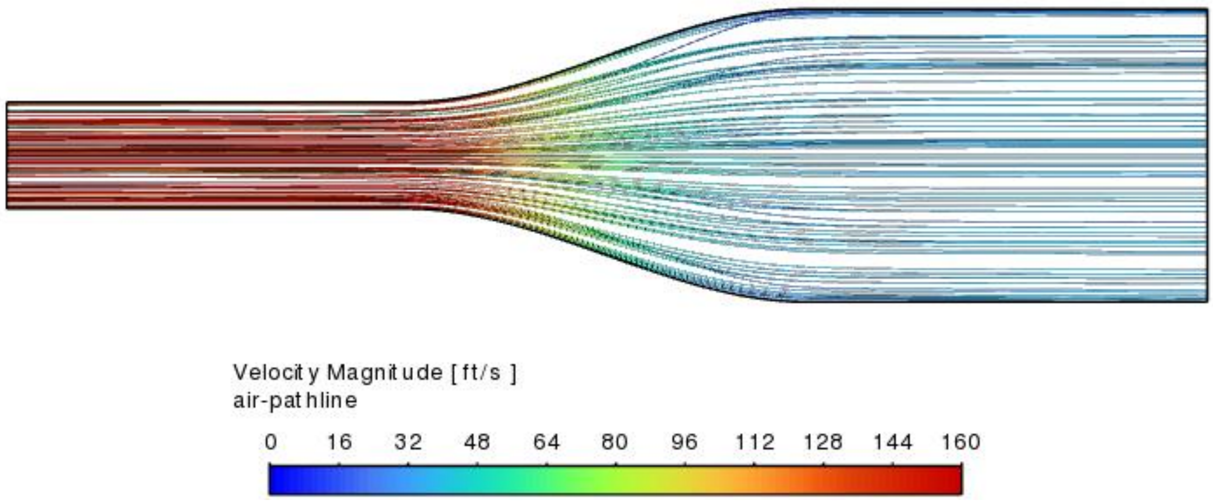


Figure 43: Velocity streamlines of air through duct

Additionally, a pressure contour and pressure drop has been calculated for the duct to show how the contraction affects the overall pressure distribution for air:

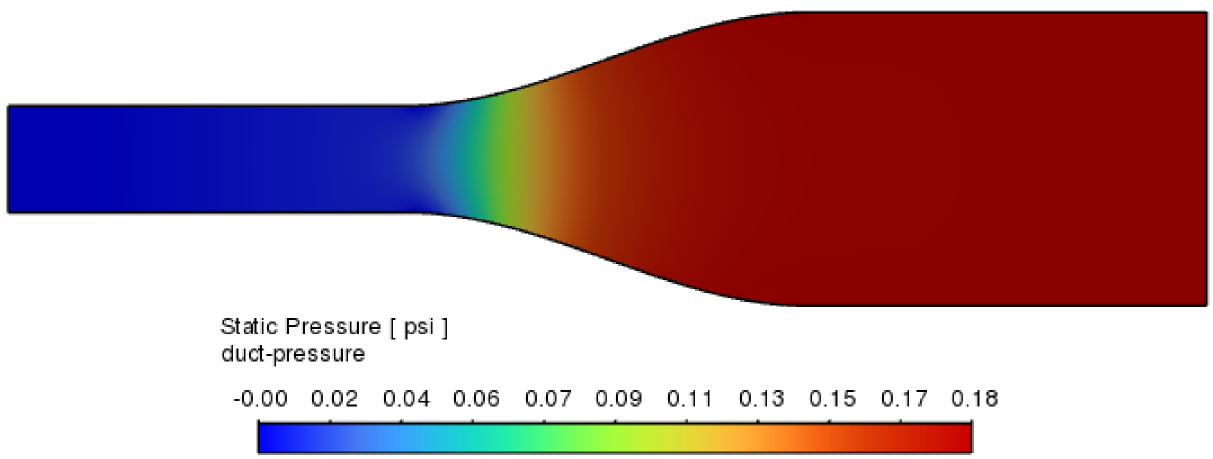


Figure 44: Pressure contour of duct midsurface

The duct has a total pressure drop of 0.18 psi across the duct according to the CFD model.

Hand Calculation for Verification:
The pressure drop across the duct can be calculated similarly to how is was calculated in the *Design Hand Calculations* section above:

$$\Delta P_{air,duct} = \gamma \frac{(V_2^2 - V_1^2)}{2g} = (0.07 \frac{lb_f}{ft^3}) \frac{(151.25fps)^2 - (36.67fps)^2}{2(32.2\frac{ft}{s^2})} (\frac{1}{144} \frac{ft^2}{in^2}) = 0.162psi$$

As discussed in the *Design Hand Calculations* section, the air flow is considered incompressible across the duct; therefore, the velocity at the duct outlet can be calculated as:

$$V_{outlet} = \frac{V_{inlet}A_{inlet}}{A_{outlet}} = \frac{(25mph)(88\,\frac{fpm}{mph})(\frac{1}{60}\,\frac{min}{s})(22in)(12in)}{(8in)(8in)} = 151.25fps$$

Results are summarized and compared to simulation results in the table below:

Table 18: Duct Model and Calculation Comparison

Parameter	**Simulation Value**	**Hand Calculation Value**
Air pressure drop	0.693 psi	0.4 psi
Outlet velocity	154.31 fps	151.25 fps

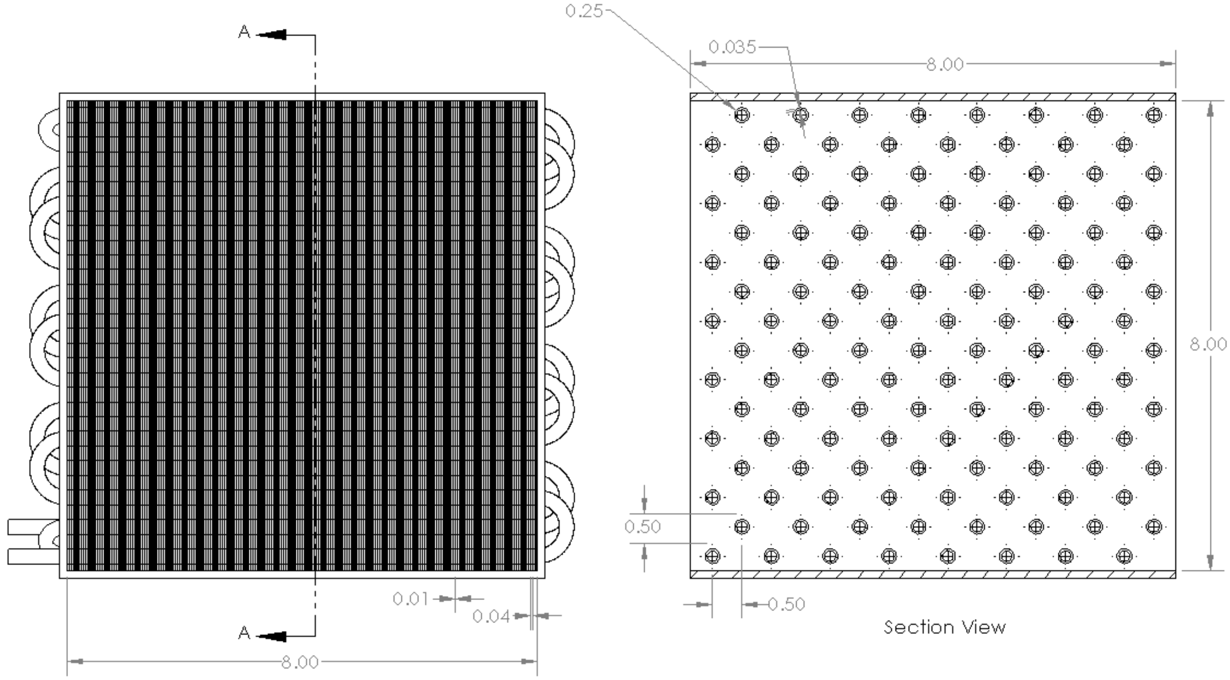
Design Summary

The benefits of the fin tube design include weight savings, packaging, and modularity. This radiator dry weight estimation is 30.15 lbs. The inlet section is made out of 26-gauge sheet aluminum, which facilitates manufacturing without compromising weight. The radiator is very compact, with a footprint of 8 in. x 8 in. x 8 in. Finally, the radiator is able to meet the stated performance requirements with a relatively low oil side pressure drop of 5.8 ksi, which could be handled by a properly sized pump.

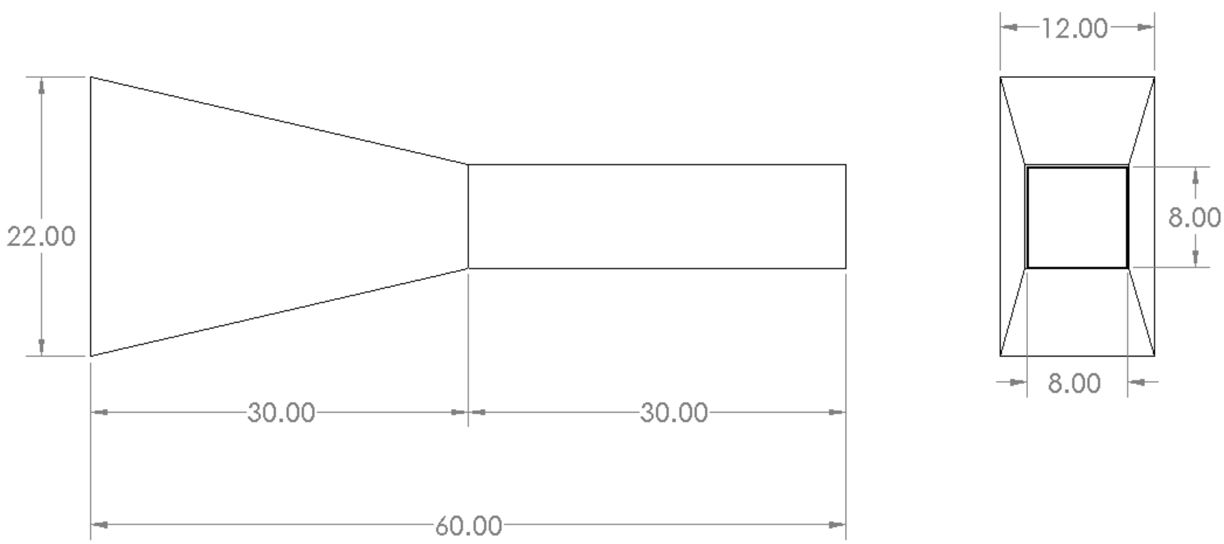
Design Time Estimate:
212 hours total

Scaled Drawings (all dimensions in inches):

Heat Exchanger:



Air Duct:



Citations

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6. Report 1135: Equations, Tables, and Charts for Compressible Flow, Ames Research Staff, Ames Aeronautical Laboratory, Moffett Field, Calif., 1953. https://www.nasa.gov/sites/default/files/734673main_Equations-Tables-Charts-CompressibleFlow-Report-1135.pdf

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CFD MIDTERM: HEAT EXCHANGER DESIGN

DEFAULT (INPUT) PARAMETERS

Oil SG	SG	0.86	
Oil viscosity	muoil	34.5	cP
Oil thermal conductivity	koil	0.15	BTU/hr-ft-R
Oil specific heat	cpoil	0.5	BTU/lb-R
Air specific heat	cpair	0.24	BTU/lb-R
Duct inlet height	hin_duct	22	in
Duct inlet width	win_duct	12	in
Duct outlet height	hin	8	in
Duct outlet width	win	8	in
Pipe outer diameter	Dpipe	0.25	in
Pipe thickness	tpipe	0.035	in
Inner Diameter	Dinner	0.180	in
Fin thickness	tfin	0.01	in
Fin width	wfin	0.5	in
Air mass flow rate	mdot_air	4.698367938	lb/s
Air inlet temperature	Tin_air	108	F
Oil inlet temperature	Tin_oil	350	F
Oil outlet temperature	Tout_oil	195	F
Air outlet temperature	Tout_air	151.6572017	F
Vehicle speed	V	25	mph
Oil volumetric flow rate	Vdot_oil	5.3	gpm
Copper thermal conductivity	kcopper	233	BTU/hr-ft-R
Copper density	rhocopper	557	lb/ft^3
Water density	rhowater	62.4	lb/ft^3
Cross flow heat exchanger correction factor	F	0.95	
LMTD	LMTD	135.110208	F
UA	UA	1380.717705	BTU/hr-R
Q	Q	177221.6035	BTU/hr

Convection coefficients (h)

Pipe - internal flow of water through	hi	614.1734422	BTU/hr-ft^2-R
Pipe - external flow of air across cylinder	hop	53.50234545	BTU/hr-ft^2-R
Fins - external flow of air across flat plate	hofin	41.22649369	BTU/hr-ft^2-R

RESULTS (DERIVED IN NEXT WORKSHEETS)

Overall Heat Transfer Coefficient	Ui	342.091264	BTU/hr-ft^2-R
Fin spacing	deltax	0.040343982	in
Pipe Length	L	1027.786774	in
Number of fins in straight line approximation	nfins_straight	25476	-
Actual number of fins	nfins	133	-
Total weight	W	24.33067391	lb
Heat exchanger efficiency alpha parameter	alpha	0.86746181	-
Heat exchanger efficiency	eta	0.657802791	-
Oil pipe friction factor	f, oil pipe	0.039039303	-
Air pipe friction factor	f, air pipe	0.29	-
Air pipe pressure correction factor	chi, air	1	-
Oil pressure drop	dP, oil	5787.370538	psi
Air pressure drop (heat exchanger)	dP, air (heat exchanger)	3.2	psi
Air pressure drop (duct)	dP, air (duct)	0.162282822	psi

DUCT MODEL INPUTS AND VALIDATION

Inlet velocity	V, in	25	mph
Inlet hydraulic diameter	Dh, in	15.52941176	in
Inlet Reynolds number	Re, in	256496.1342	-
Inlet turbulence intensity	T_i, in	0.033727557	-
Inlet turbulence length scale	L, in	0.064705882	ft
Outlet hydraulic diameter	Dh, out	8	in
Outlet Reynolds number	Re, out	132134.3722	-
Outlet turbulence intensity	T_i, out	0.036643167	-
Outlet turbulence length scale	L, out	0.033333333	ft
Air pressure drop	dP, air (duct)	0.162282822	psi

2 FIN MODEL INPUTS AND VALIDATION

Air height	h_air	0.5	in
Air width	w_air	0.1	in
Air hydraulic diameter	Dh, air	0.166666667	in
Air mass flow rate	mdot, air	0.0036706	lb/s
Air Reynolds number	Re, air	11355.29761	-
Air turbulence intensity	T_i, air	0.049798948	-
Air turbulence length scale	L, air	0.000694444	ft
Oil hydraulic diameter	Dh,oil	0.18	in
Oil mass flow rate	mdot, oil	0.63520288	lb/s
Oil Reynolds number	Re, oil	2325.834752	-
Oil turbulence intensity	T_i, oil	0.060715227	-
Oil turbulence length scale	L, oil	0.00075	ft
Model inner surface area	Ai_model	0.001178097	ft^2
Air heat capacity	Cair	3.171398358	BTU/hr-R
Oil heat capacity	Coil	1143.365184	BTU/hr-R
Number of transfer units	NTU	0.127078572	-
Heat capacity ratio	Cr	0.002773741	-
Heat exchanger effectiveness	epsilon	0.119304479	-
Maximum heat transfer rate	Qmax	767.4784027	BTU/hr
Actual heat transfer rate	Q	91.56361081	BTU/hr
Oil outlet temperature	Toil_out	349.9199174	F
Air outlet temperature	Tair_out	136.8716839	F
Oil pressure drop	dP, oil	11.26181166	psi
Air pressure drop	dP, air	0.2	psi

TUBE BANK MODEL INPUTS AND VALIDATION

Air height	h_air	1	in
Air width	w_air	0.1	in
Air hydraulic diameter	Dh, air	0.181818182	in
Air mass flow rate	mdot, air	0.0073412	lb/s
Air Reynolds number	Re, air	12387.59739	-
Air turbulence intensity	T_i, air	0.049260248	-
Air turbulence length scale	L, air	0.000757576	ft
Oil hydraulic diameter	Dh,oil	0.18	in
Oil mass flow rate	mdot, oil	0.63520288	lb/s
Oil Reynolds number	Re, oil	2325.834752	-
Oil turbulence intensity	T_i, oil	0.060715227	-
Oil turbulence length scale	L, oil	0.00075	ft
Model inner surface area	Ai_model	0.003534292	ft^2
Air heat capacity	Cair	6.342796717	BTU/hr-R
Oil heat capacity	Coil	1143.365184	BTU/hr-R
Number of transfer units	NTU	0.190617858	-
Heat capacity ratio	Cr	0.005547481	-
Heat exchanger effectiveness	epsilon	0.173431762	-
Maximum heat transfer rate	Qmax	1534.956805	BTU/hr
Actual heat transfer rate	Q	266.2102629	BTU/hr
Oil outlet temperature	Toil_out	349.7671695	F
Air outlet temperature	Tair_out	149.9704863	F
Tube bend K factor	Kbend	1.2	-
Oil pressure drop	dP, oil	53.67878173	psi
Air pressure drop	dP, air	0.4	psi

Convection Coefficient at Fins and Fin Spacing

DEFAULT (INPUT) PARAMETERS

Air crossflow velocity	Vair	151.25	fps
Fin thickness	tfin	0.01	in
Fin width	wfin	0.5	in

FLUID PROPERTIES

	Air	Units
k	0.0157	BTU/hr-ft-R
rho	0.069893	lb/ft^3
cp	0.24	BTU/lb-R
mu (dynamic viscosity)	0.00001293	lb/s-ft
v (kinematic viscosity)	0.000184997	ft^2/s
alpha	0.935953455	ft^2/hr

Reynolds Number (Re)	Re	34065.89282
Prandtl Number (Pr)	Pr	0.711561783

Convection coefficient at fins	h	41.22649369	BTU/hr-ft^2-R
	del_h	0.013545035	in
Boundary Layer thickness	del_t	0.015171991	in

Fin spacing	delta_x	0.040343982	in

Convection Coefficient at Bare Pipe

DEFAULT (INPUT) PARAMETERS

Air crossflow velocity	Vair	151.25	fps
Pipe outer diameter	Dpipe	0.25	in

FLUID PROPERTIES

	Air	
k	0.0157	BTU/hr-ft-R
rho	0.070	lb/ft^3
cp	0.24	BTU/lb-R
mu (dynamic viscocity)	0.00001293	lb/s-ft

Derived Parameters

v (kinematic viscosity)	0.000184997	ft^2/s
alpha	0.935953455	ft^2/hr
Reynolds Number (Re)	17032.94641	-
Prandtl Number (Pr)	0.711561783	-

C	0.193	-
n	0.618	-

Convection coefficient h	53.50234545	BTU/hr-ft^2-R

Reynolds Number Range

Low	**High**	**C**	**n**
0.4	4	0.989	0.33
4	35	0.911	0.385
35	4083	0.683	0.466
4083	40045	0.193	0.618
40045	400000	0.0266	0.805

Convection Coefficient Inside Pipe

DEFAULT (INPUT) PARAMETERS

Mass flow rate	mdot	0.63520288	lb/s
Pipe outer diameter	Dpipe	0.25	in
Pipe thickness	tpipe	0.035	in
Pipe inner diameter	Dinner	0.18	in

FLUID PROPERTIES

	Oil	
k	0.15	BTU/hr-ft-R
rho	53.664	lb/ft^3
cp	0.5	BTU/lb-R
mu (dynamic viscosity)	0.023182083	lb/s-ft

Derived Parameters

v (kinematic viscosity)	0.000431986	ft^2/s
alpha	0.00559034	ft^2/hr
Reynold's Number (Re)	2325.834752	-
Prandtl's Number (Pr)	278.185	-

Nusselt Number (Nu)	61.41734422

Convection Coefficient	614.1734422	BTU/hr-ft^2-R

Total Resistance Calculation

Geometries

Pipe outer diameter	Dpipe	0.25	in
Pipe thickness	tpipe	0.035	in
Pipe inner diameter	Dinner	0.18	in
Fin thickness	tfin	0.01	in
Fin width	wfin	0.5	in
Fin spacing	deltax	0.04034398	in

Thermal conductivities (k)

Pipe thermal conductivity	kpipe	233	BTU/hr-ft-R

Convection coefficients (h)

Pipe - internal flow of water through	hi	614.173442	BTU/hr-ft^2-R
Pipe - external flow of air across cylinder	hop	53.5023455	BTU/hr-ft^2-R
Fins - external flow of air across flat plate	hofin	41.2264937	BTU/hr-ft^2-R

Areas

Pipe inner (per fin)	Ai,pipe	0.02281398	in^2
Pipe outer (per fin)	Ao,pipe	0.02383211	in^2
Fin (flat plate with hole)	Afin	0.40182523	in^2

Resistances (R)

Advective resistance from water to pipe	Rinner	10.2770937	R-hr/BTU
Conductive resistance through pipe	Rpipe	0.08873882	R-hr/BTU
Convective resistance across bare pipe	Rbare	112.93465	R-hr/BTU
Conductive resistance through fin	Rk,fin	0.01602131	R-hr/BTU
Convective resistance across fin (flat plate)	Rh,fin	8.6925839	R-hr/BTU

Rtotal per fin	18.4509769	R-hr/BTU