INVESTIGATION OF INTERCOOLER FINS OF A TWO STAGE RECIPROCATING AIR COMPRESSOR

Surisetty VENKATAIAH[1], Gudimalla STHITHAPRAGNA[1], Nagarapu NAGARAJU[1], Kanneboina VIVEK YADAV[1]

This study primarily aims to investigate and enhance the heat transfer rate through intercooler fins of a Two Stage Reciprocating Air Compressor by modifying the fin configuration. In this paper, an identical model of the existing intercooler with annular fin configuration is modeled in NX 12.0 and later analyzed in Ansys by implementing a steady state thermal analysis. The fluid domain of the intercooler is subjected to a spatial temperature variation because of the flow of air along its length, with its inlet and outlet temperatures being 115 °C and 94 °C, for the air delivery pressures 1 bar, 1.7 bar, and 6 bar at the inlet of the LP (Low-Pressure) cylinder, the outlet of the LP cylinder and outlet of the HP (High-Pressure) cylinder, respectively. A mathematical model is adopted to analyze the annular fins of the existing intercooler system and the same results are used to validate with the results of steady state thermal analysis to evaluate its temperature and heat flux distributions. The validation results revealed that the percentage of temperature depreciation between theoretical and analytical values was in the range of 0.00966 to 0.01949, which is considered quite little in practice. It has been also observed that the rectangular fin configuration of the intercooler yields a 37% higher heat flux than the annular fin configuration because of its larger surface area. In addition to this, different materials of Aluminum alloys and Copper alloys are assigned to the fins of intercooler, allowing for further optimization of various fin configurations.

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Keywords: Fins, Intercooler Fins, Ansys Mechanical, Air Compressor

1. Introduction

A Reciprocating Air Compressor is a positive displacement device in which a volume of air is enclosed and is elevated to a higher pressure. They operate by means of a piston in a cylinder following the first inversion of a single slider-crank mechanism. Two Staging in a Reciprocating Air Compressor can be accomplished with the help of an intake cooling device known as an intercooler. The intercooler of a Reciprocating Air Compressor is adjoined between the two stages of its compression processes i.e., between the LP Cylinder and the HP Cylinder. The heat transfer rate can be improved by facilitating a maximum surface area to the intercooler across which the convection and radiation occur. So, the intercooler surface is provided with a series of fins that extend from the wall into the

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surrounding fluid that enhances the heat transfer rate majorly by means of convection and nominally by means of radiation. So, the effect of radiation is ignored in many cases. The intercooler fins are subjected to forced convection with the help of a fan equipped to the air compressor. It is positioned such that the airflow is precisely focused onto the intercooler's fins. This fan is driven with the help of a belt drive by means of a squirrel induction motor that also runs the air compressor. Fins are extended surfaces that are used to improve the rate of heat transfer from the surfaces of various heat transfer systems. They are commonly used to attribute a case involving heat transfer by conduction within the solid and heat transfer by both convection and radiation from the boundary surfaces of the solid. A fin material is desired to have a very high thermal conductivity for a rapid thermal response. The fin with the high thermal conductivity is subjected to a flowing fluid, which cools or heats it, allowing the heat to be conducted from the wall through the fin. Ideally, a fin material is chosen such that it is anti-corrosive, less expensive, and possesses high strength. The various kinds of fins available are flat fins, annular fins, pin fins, rectangular fins, etc., of both uniform and non-uniform cross-sections. Different configurations are used for different applications depending upon the suitability, space, and weight limitations. In order to improve a fin body's effectiveness for heat transmission, Kota Leela Sai Bharath *et al.* [1] investigated on different cases like fins with varying thickness, rectangular perforation, rectangular extension, and varying material using Catia. The results of the temperature gradient along the length of the fin and heat flux were compared using Ansys software. Abdullah *et al.* [2] investigated a horizontal rectangular fin embedded with equilateral triangular perforations to enhance the free convection heat transfer. The heat dissipation rate of a perforated fin is compared to that of an equivalent solid fin. The influence of the perforated fin's geometrical dimensions and thermal qualities is also investigated in depth. B. T. F. Chung *et al.* [3] discussed a methodology to identify the ideal dimensions for cylindrical pin fins and rectangular longitudinal fins by taking the effects of variable heat transfer coefficient under transverse heat transfer condition. Pravin *et al.* [4] evaluated the performance of aluminium and copper fin tube intercooler used in multistage reciprocating air compressor and results were validated with Ansys - Fluent - 18.2 software. Senthil Kumar *et al.* [5] analyzed and optimized the compressor intercooler fins in turbocharger using Taguchi's design of experiment by taking three parameters i.e., the material of the fin, the shape of the fin, and the thickness of the pipe into account. Kannan *et al*. [6] carried out CFD analysis on different perforated fins to study density variation, velocity, and temperature drop. In comparison to its dimensionally equivalent solid fin, the perforated fin was found to have a higher heat transfer rate. Sane *et al*. [7] carried out both CFD analysis and experimentation to compare the natural convection heat transfer rate in a rectangular notched fin with an unnotched fin. Dipankar Bhanja *et al.* [8] studied the performance parameters, and the heat transfer rate through a porous pin fin under free convection conditions. P. Ragupathi *et al.* [9] used CFD to achieve the instantaneous pattern of local heat transfer coefficients and local velocities over the surface of the compressor. It has been found that temperatures in areas under the influence of fan has fewer values than the rear side. By modifying the fin configuration, the temperature difference between the front and rear sides of the compressor cylinder was minimized. Saeed Tiari *et al.* [10] employed annular fins on a latent heat thermal energy storage system to investigate its performance. A fin-configuration of twenty in number with uniform length was the most successful in terms of both charging and discharging time, reducing the overall time by 76.3 percent. Ratnesh Kumar Yadav *et al.* [11] reviewed on heat transfer from fins of different profiles. It has been observed that cubic pin fin arrays have been found to have better heat transfer rates than elliptical pin fin arrays. S. Benjamin Franklin *et al*. [12] investigated the efficiency of various shapes of fins (annular, square, and hexagonal) with a rectangular duct pin-fin apparatus. Due to its large area of contact, it has been determined that the hexagonal fin is best for its highest heat transfer rate and the paramount efficiency. Xiangrui Meng *et al.* [13] investigated a straight fin heat sink under natural convection conditions at different mounting angles with constant heating power conditions. It has been observed that the heat sink performs the worst at a mounting angle of 15°, and the best at a mounting angle of 90°. P Moorthy *et al.* [14] studied the effect of various shapes of fins on the performance of compact finned flat tube heat exchangers. In this study, the rectangular fin has been found to be the most effective with respect to the higher heat transmission. However, the disadvantage of the large friction factor causes the fin to be the least efficient of all. The plain fin, on the other hand, had the least heat transmission but the maximum efficiency. K Shahril *et al.* [15] studied the simulation of heat transfer of motorcycle fins under varying climatic conditions. Fins confine the air and retain the heat of the engine block, which can lead to overheating. The extent of overheating is largely influenced by the design of fins. However, in the present project, as there exists a spatial temperature variation along the length of fluid domain in the cavity of intercooler at any given point of time, it is thought to implement Steady State Thermal Analysis on the fins of the intercooler which is logically the last step of Transient Thermal Analysis. Some of the above studies did not employ the exact heat transfer coefficient values and also didn't validate the approach used. Due to this reason, this work seeks to suggest, implement, and validate a methodology by using a precise heat transfer coefficient throughout. It is also required to improve the existing model of intercooler. So, different fin configurations like flat fins and rectangular fins are perceived, modeled, and analyzed for optimizing the existing model. Different alloys of aluminium and copper are also thoroughly investigated to assign them as fin material to broaden the scope of optimization.

2. Material and Methods

2.1 Methodology

Initially, experimentation is carried out on the Two-Stage Reciprocating Air Compressor to obtain the inlet and outlet temperatures of the Intercooler for the respective delivery pressures of LP and HP cylinders. Apart from this, the convective heat transfer coefficient of air cross-flowing over the intercooler fins is to be calculated theoretically. The parametric model of intercooler with annular fin configuration is to be analyzed in Ansys by implementing steady-state thermal analysis on it by giving the specified boundary conditions of spatial temperature variation and convection. *Table 1* below shows the material and thermal properties of the intercooler with annular fin configuration.

Table 1

Material and Thermal Properties of the Intercooler with Annular Fin Configuration

Material	PROPERTY		
	Density [kg/m^3]	Tensile Yield Strength [MPa]	Isotropic Thermal Conductivity [W/m-K]
Copper-Nickel Alloy	8300	280	40
Aluminium Alloy 204	3000	220	120
Air	1.177	-	0.0242

2.2 Experimentation

Fig. 1 Experimentation Setup

To perform the experiment on the Air Compressor, the power supply is connected to it and the system is switched on. The delivery pressures at the inlet and outlet valves of the LP Cylinder and HP Cylinder are set up manually. The orifice of the storage tank of the compressor is opened so as to allow the compressed air to stream out with a uniform discharge rate. Once the entire system gets stabilized, temperatures at the inlet and the outlet of the intercooler are noted down for their respective delivery pressures from the digital temperature indicator.

Table 2

Experimentation Results of the Two Stage Reciprocating Air Compressor

Delivery Pressure [bar]			Intercooler IAT [°C]	Intercooler OAT [°C]
LP CIAP	LP COAP	HP COAP		
1	1.7	6	115	94

Whereas CIAP: Cylinder Inlet Air Pressure, COAP: Cylinder Outlet Air Pressure, IAT: Inlet Air Temperature, OAT: Outlet Air Temperature.

2.3 Calculation of convective heat transfer coefficient

The Convective Heat Transfer Coefficient (h) of air cross-flowing over the fins of the intercooler can be obtained by the use of the following empirical correlation.

$$h = 12.12 - 1.16\ V + 11.6\sqrt{V} \tag{1}$$

So, in order to determine the convective heat transfer coefficient of the air, the velocity of air from the fan cross-flowing over the fins of the intercooler is to be known. The calculation of air velocity could be possibly done by bringing in the concept of power generation from wind turbines where the wind energy is converted into mechanical energy that rotates the fan of the wind turbine. Juxtaposing the same scenario here, energy in the form of air is produced by rotation of the fan which is quite opposite as in the case of power generation from the wind turbine. Further, the Principle of Conservation of Energy is used to theoretically determine the velocity of air from the fan [16]. Let us assume that the mass of air in motion flowing with a velocity 'V' and the corresponding density of air is 'ρ'. Let 'A' be the swept area of the fan and is given by $A = \pi r^2$, where 'r' is the radius that is equal to the blade length of the fan. The equation of power in the air is given by

$$P = \frac{1}{2}\rho A V^3 \tag{2}$$

Power transmitted from the motor to fan (P_0) is given by Motor rating multiplied by efficiency of motor and efficiency of transmission (*where* η_{motor} = *Mechanical efficiency of the motor of the Air Compressor* = 0.85 *and* $\eta_{transmission}$= *Transmission efficiency of the Belt Drive* = 0.9). From *the Principle of Conservation of Energy*, power in the air equals the power transmitted from the motor to the fan which gives the velocity of air (V) as 17.9518 m/s. The heat transfer coefficient can be obtained from the provided empirical relationship and is given as 40.45 W/(m^2K).

2.4 Design and analysis of an annular fin

Initially, using Ansys Workbench, the required geometry of the annular fin is updated from the source, a fluid domain is created in the cavity of the intercooler using the Design Modeler. Now, the model is opened and materials are assigned to their respective geometries, i.e., Cu-Ni to the intercooler body and Aluminum alloy 204 to the annular fin. The given model is meshed with quad and tetrahedral elements with the number of nodes being 53633 and number of elements being 17587. After the completion of meshing, the boundary conditions are given in the segment of steady state thermal analysis. The surfaces of the fin are selected for convection and the Convective Heat Transfer Coefficient is given as 40.45 W/(m^2K) at 27 °C.

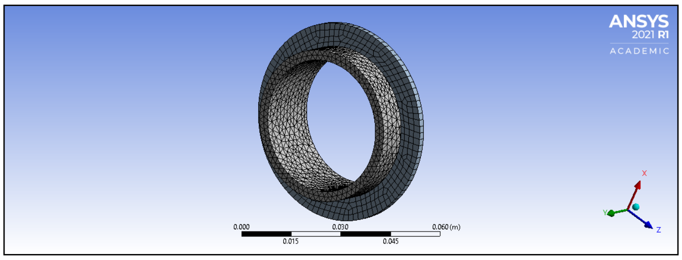
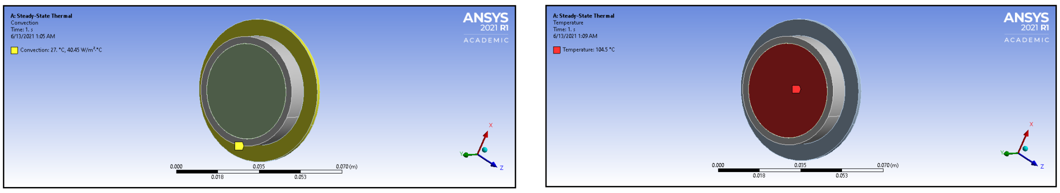


Fig. 2 Meshed Model of the Annular Fin



(a) (b)

Fig. 3 (a) Surfaces of the Annular Fin Subjected to Convection (b) Fluid Domain Subjected to Temperature

The actual fluid domain is subjected to a spatial temperature variation along its length with temperatures at its tip ends being 115 °C and 94 °C. An average temperature of 104.5 °C is considered as the length of fin is very less. The annular fin is solved for Heat Flux and Temperature in the solution segment. In order to get the radial temperature distribution of the annular fin, a path is defined from the inner radius to the outer radius of the annular fin in the segment of Construction Geometry. The defined path is used to accommodate the temperature variation of the annular fin along its radial length.

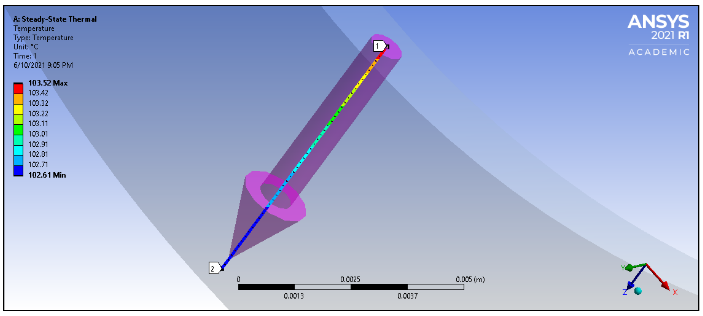
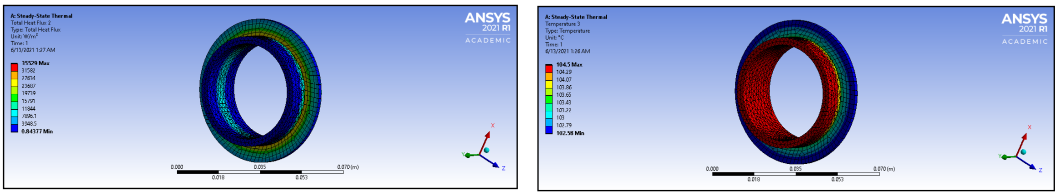


Fig. 4 Path Created using Construction Geometry for Obtaining Temperature Variation along the Radial Length of the Annular Fin



(a) (b)

Fig. 5 Contours of (a) Heat Flux Distribution (b) Temperature Distribution of the Annular Fin

Result	Heat Flux [W/m^2]	Temperature [°C]
Maximum Value	57315	104.5
Minimum Value	2070.4	102.58
Average Value	15009	104.17

Table 3

Results of Steady State Thermal Analysis on the Annular Fin

2.5 Validation of simulation results of the annular fin

In order to theoretically validate the above obtained simulation results from the Ansys, consider an annular fin as shown in Fig. 6. Let us consider an annular fin with inner radius ‘r_i’ and outer radius ‘r_o’ for implementing a general thermal analysis on it.

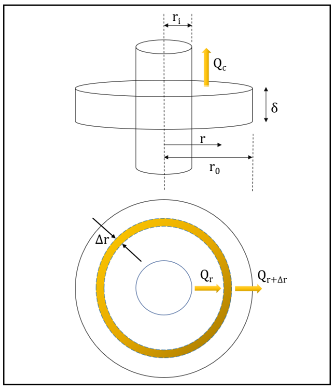


Fig. 6 Schematic of an Annular Fin

Mathematical Model

The General Differential Equation of heat transfer through an annular fin is given by

$$\frac{d}{dr}\left[A(r)\,\frac{dt}{dr}\right] - \frac{h\,P(r)}{K}\,(t - t_\infty) = 0 \qquad (3)$$

Substituting $P(r) = 2 \times 2\pi r$

$$A(r) = 2\pi r \times \delta$$

$$\theta = t - t_\infty$$

P(r) and A(r) are the circumference and cross-sectional area of the annular fin of thickness 'δ' at a radial distance of 'r' respectively.

The equation (3) on solving becomes

$$r^2\,\frac{d^2\theta}{dr^2} + r\,\frac{d\theta}{dr} - m^2 r^2\,\theta = 0 \qquad (4)$$

Here $m^2 = \frac{2h}{K\delta}$

Whereas 'm' is known as Fin Parameter.

Equation (4) is in the form of Modified Bessel's Equation and its general solution is given by

$$\theta(r) = C_1 I_o(mr) + C_2\,K_o(mr) \qquad (5)$$

whereas 'I' and 'K' are termed as Modified Bessel's functions of first and second kind respectively.

They are given by

$$\begin{cases} I_v(x) = \sum_{k=0}^{?} \frac{(x/2)2^{k+v}}{k!\,\Gamma(v+k+1)} \\ K_v(x) = \frac{I_{-v}(x) - I_v(x)}{\sin \Pi v} \end{cases} \tag{6}$$

Boundary conditions are to be applied to the annular fin to obtain the temperature profile equation.

Boundary Condition 1: At $r = r_i$, $t = t_b$, *hence* $\theta = \theta_b$
Whereas t_b is the base temperature of the fin at its inner radius

Boundary Condition 2: The tip of the annular fin when considered to be adiabatic i.e., there is no heat transfer from its tip. So, at $r = r_o$, $\frac{dt}{dr} = 0$, *hence* $\frac{d\theta}{dr} = 0$

Solving equation (5) by applying boundary conditions further gives

$$\begin{cases} C_1 = \frac{\theta_b K_1(mr_o)}{I_o(mr_i) \times K_1(mr_o) + I_1(mr_o) \times K_o(mr_i)} \\ C_2 = C_1 \frac{I_1(mr_o)}{K_1(mr_o)} \end{cases} \tag{7}$$

Substituting the values of C_1 and C_2 in equation (5), the equation of temperature profile is obtained as

$$\frac{\theta}{\theta_b} = \frac{I_o(mr) \times K_1(mr_o) + K_o(mr) \times I_1(mr_o)}{I_o(mr_i) \times K_1(mr_o) + I_1(mr_o) \times K_o(mr_i)} \tag{8}$$

The obtained equation of temperature profile is valid only in the case of annular fin with its tip being adiabatic. But in reality, pertaining to our case of study, the tip is convective in nature i.e., heat transfer occurs across the tip of fin as well. Hence, the obtained equation should be applied to an active convecting tip when the outer radius is replaced by a corrected radius as follows in equation (9) [17].

$$r_{Oc} = r_O + \frac{\delta}{2} \tag{9}$$

The base temperature of the annular fin (t_b) as obtained from the analytical solution is 103.52 °C and the ambient temperature (t_∞) is taken as 27 °C.

Calculation of corrected radius: $r_{Oc} = 0.0301 + \frac{0.002}{2} = 0.0311\ m$
Calculation of Fin parameter:

$$m = \sqrt{\frac{2h}{K\delta}} = \frac{2\times 40.45}{120\times 0.002} = 18.36$$

$$\theta_b = t_b - t_\infty = 103.52 - 27 = 75.52\ °C$$

The values of Bessel's functions could be obtained from the table data of the functions corresponding to the values of (mr). Using the equation (8), the theoretical values of temperatures of the annular fin are obtained for their corresponding radial distances. Path Length is the length measured from the inner radius to the outer radius of the path that was created using construction geometry. The percentage of depreciation could be calculated from the formula shown in the equation (10).

$$\%\ Depreciation = \left|\frac{Theoretical\ Value - Analytical\ Value}{Theoretical\ Value}\right| \times 100 \qquad (10)$$

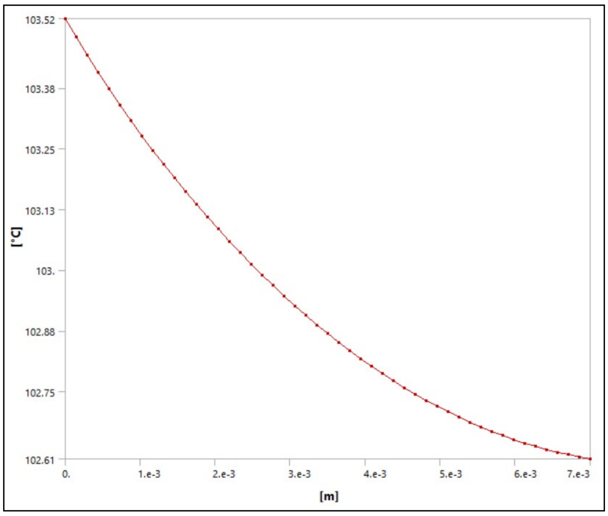
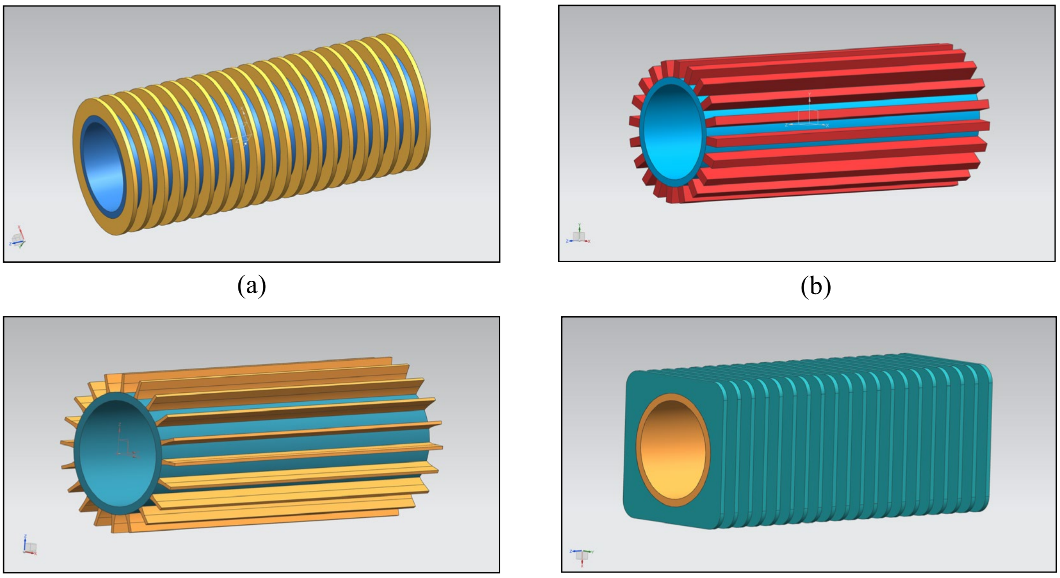


Fig. 7 Graph Plotted Between the Analytical Values of Temperatures Obtained and their Corresponding Distances of the Path Created Radially

From the plot as shown in Fig. 7, it can be understood that the temperature distribution of the annular fin is a radially declining curve that follows the temperature profile equation (8). It is found that the percentage of depreciation between the theoretical and analytical values of temperatures for their corresponding radial distances is in the range of 0.00966 to 0.01949, which is considered to be very minimal. Hence, it is inferred that the model and the method of implementation of steady state thermal analysis in Ansys is precise and the same method could be adopted to analyze the intercooler with various fin configurations.

2.6 Design and analysis of intercooler with various fin configurations

The parametric model of the intercooler with annular fin configuration is designed in NX 12.0 as per specified dimensions. Flat and rectangular fin configurations are considered for the scope of optimization. The fins are modeled in such a way that the space occupied is taken as a constraint in order to accommodate various fin configurations in the same space as that occupied by annular fins for a rational comparability. In reality, convection occurs from the outer surfaces of fins along with the outer surface of the intercooler on which the fins are mounted. So, in the segment of steady state thermal analysis, both the surfaces of fins and intercooler are selected for convection and the calculated Heat Transfer Coefficient is given as an input which is 40.45 W/(m^2K) at an ambient temperature of 27 °C. Apart from applying the convection boundary conditions, the fluid domain that was created inside the cavity of the intercooler is subjected to a spatial temperature variation along its length with temperatures at the ends being 115 °C and 94 °C.



(a) (b) (c) (d)

Fig. 8 Parametric Models of (a) Intercooler with Annular Fin Configuration (b) Intercooler with Flat Fin Configuration of Uniform Cross-Section (c) Intercooler with Flat Fin Configuration of Non-Uniform Cross-Section (d) Intercooler with Rectangular Fin Configuration

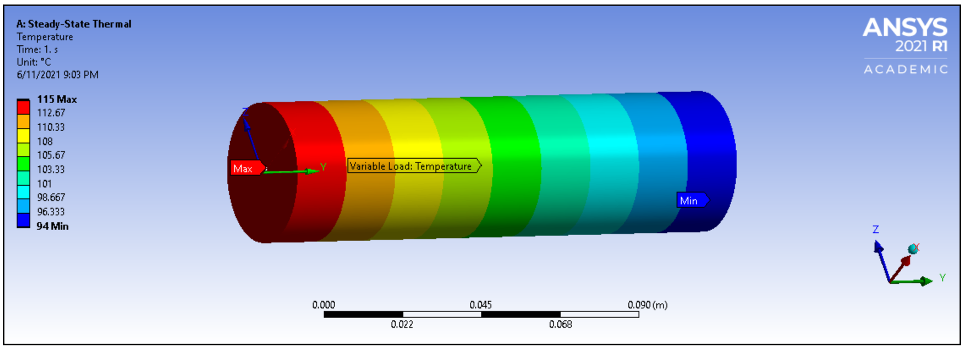


Fig. 9 Fluid Domain Subjected to Spatial Temperature Variation

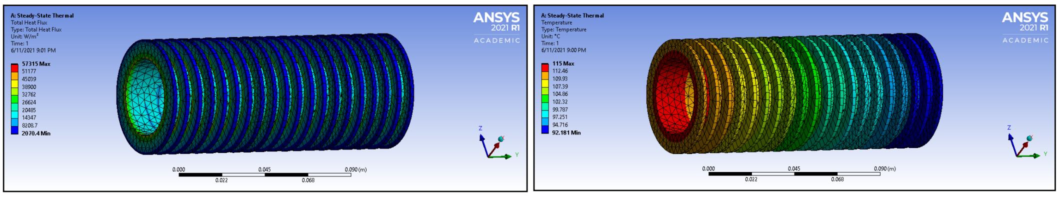
The above problem is solved for Heat flux and Temperature in the solution segment of the Ansys Mechanical. Similarly, the same analysis is implemented on various other designed parametric models of different fin configurations with different materials. The material properties of various alloys of Aluminium and Copper that are used for the scope of optimization are shown in *Table 4* [21].

Table 4

Material Properties of Various alloys used as Fin Material

S.NO	MATERIAL	DENSITY [kg/m^3]	THERMAL CONDUCTIVITY [W/m-K]	YIELD TENSILE STRENGTH [MPa]
1.	Aluminium Alloy 204	3000	120	180-220
2.	Aluminium Alloy 2014 T6	2800	154	414-365
3.	Aluminium Alloy 6061	2700	167	310-124
4.	Aluminium Alloy 7068	2850	190	590
5.	Copper Nickel Beryllium Alloy C17150	8900	210	170-750

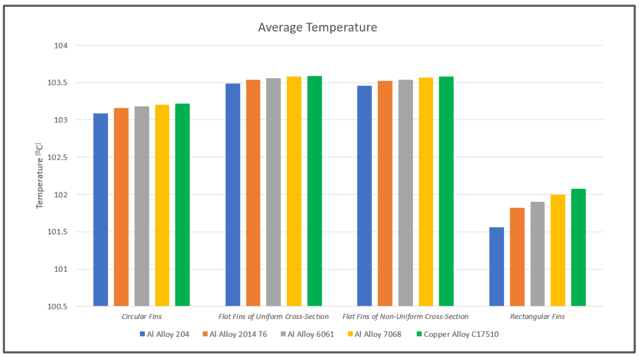
3. Results and discussion



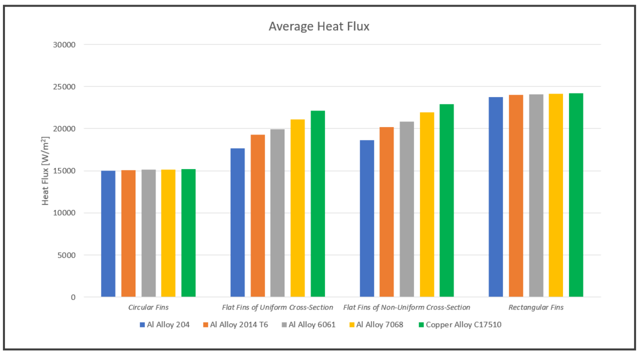
(a) (b)

Fig. 10 Contours of (a) Heat Flux and (b) Temperature Distribution with Aluminium Alloy 204 as fin material

The results of steady state thermal analysis viz., Total Heat Flux and Temperature are shown in Fig. 10. Analysis is implemented on different fin configurations with different materials of the fins by keeping the material of the intercooler tube i.e., Copper Alloy unchanged and the corresponding results are presented in the form of graphs as shown in Fig. 11.



(a)



(b)

Fig. 11 Bar Graphs Showing (a) Average Temperature (b) Average Heat Flux of Different Fin Configurations with Different Materials

4. Conclusions

The different fin configurations of intercooler with different materials are perceived, studied, and later analyzed for enhancing the heat transfer rate. A methodology is proposed to analyze the models and the same is validated with the help of single annular fin.

1. From the analysis of an annular fin, it has been found that the percentage of depreciation between the theoretical and analytical values of Temperatures is in the range of 0.00966 to 0.01949, which is in practice, considered as very minimal. This proves that the model established and methodology adopted to implement the steady state thermal analysis is precise.

2. With use of the existing primary material of the fins i.e., Aluminium alloy 204, steady state thermal Analysis is implemented on different configurations for the same spatial temperature variation along the length of the intercooler. The results show that the rectangular fin configuration of the intercooler yields a heat flux of 102420 W/m^2 which is 37% higher than the annular fin configuration because of its larger surface area. It has been also found that the flat fins of Non-Uniform Cross-Section yields higher heat flux than flat fins of Uniform Cross-Section. Also, by assigning different materials of Aluminium alloys and Copper alloys to various fin configurations of intercooler, it has been discovered that the rectangular fin configuration with Aluminium alloy 7068 and Copper alloy C17510 yield higher heat flux than other alloys.

3. The rectangular fin configuration is found to have the lowest average temperature, followed by the annular fin configuration. A lowest and highest average temperature of 101.56 °C and 102.08 °C is observed for Al alloy 204 and Copper alloy C17510 respectively for rectangular fin configuration.

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