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Effect of Engine Oil in the Aftercooler of a 2HA4TERS Horizontally Balanced Opposed Air Compressor

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Abstract

Thermal designing of an aftercooler typically includes the determination of heat transfer area ,number of tubes, tube length and tube diameter ,tube layout, number of shell and tube passes ,tube pitch, number of baffles, its type and size. CFD model of the after cooler had been developed and it is validated. Unexpected heat is generated at the after cooler of a 2HA4TERS, horizontally balanced opposed air compressor. This causes overheating and burning of the after cooler. This paper aims to determine the effect of variation in oil discharge in to the cylinder of the compressor by analyzing the CFD model of the after cooler. LMTD, Heat flow Q, Weight flow W, Clearance between the tubes C1,Baffle space B, Flow area a, Mass velocity G, Average temperature of the cold fluid ta, Average temperature of the hot fluid Ta, Reynolds number for heat transfer Re, Heat transfer coefficient in general h, for inside fluid hi, and for out side fluid ho, Value of hi when referred to the tube outside diameter hio, Clean overall coefficient of heat transfer Uc, Design overall heat transfer coefficient Ud and dirt factor Rd are determined by using Kerns method.

Keywords: aftercooler, heat transfer area, shell and tube passes, tube pitch, baffles, CFD model.

Introduction

The paper reveals the determination of Dirt factor R_d of the aftercooler by determining the following: heat balance between the air and water flowing through the after cooler. LMTD of after cooler, mass velocity of after cooler fluids, Reynolds's number of after cooler fluids, factor for heat transfer j_h, weight flow of after cooler fluids, the heat transfer co-efficient.

An aftercooler is a shell and tube heat exchanger which consists of a shell (a large pressure vessel) with a bundle of tubes inside it. One fluid runs through the tubes and another fluid flows over the tubes (through the shell) to transfer heat between the two fluids. The sets of tubes is called a tube bundle. It is the most common type of heat exchanger in oil refineries and other large chemical processes, and is suited for high pressure applications.

Nomenclature

B	Baffle spacing
C	Clearance between tubes, m
C	Specific heat, J/kgK
D	Inside diameter of tubes, m
De	Equivalent diameter, m
F_T	Temperature difference factor
G	Mass velocity ,kg/sm^2
h, hi, ho	Heat transfer coefficient in general, for inside and for outside fluid respectively, W/m^{2}K
Hio	Value of hi when referred to the tube OD, W/m^{2}K
Jh	Factor for heat transfer
K	Thermal conductivity, W/mK
Pt	Tube pitch,m
Rd	Dirt factor,mk/W
Ta	Average temperature of cold fluid ,K
ΔT	True temperature difference K
Uc, Ud	Clean and design overall coefficient of heat W/m^{2}K

Literature Review

Ahmad Fakheri [11] in his paper shows that how to calculate the efficiency of the heat exchangers based on the second law of thermodynamics. He says that corresponding to every heat exchanger there is an ideal balanced counter flow heat exchanger which has the properties of same UA, same AMTD and minimum entropy generation corresponding to minimum losses and irreversibility. The efficiency of the heat exchanger may be calculated by comparing the heat transfer capability of actual heat exchanger with that of the ideal heat exchanger

Rajeev Mukherjee [12] explains the basics of exchanger thermal design, covering such topics as: STHE components; classification of STHEs according to construction and according to service; data needed for thermal design; tube side design; shell side design, including tube layout, baffling, and shell side pressure drop; and mean temperature difference. The basic equations for tube side and shell side heat transfer and pressure drop. Correlations for optimal condition are also focused and explained with some tabulated data. This paper gives overall idea to design optimal shell and tube heat exchanger.

The optimized thermal design can be done by sophisticated computer software however a good understanding of the underlying principles of exchanger designs needed to use this software effectively.

Jiangfeng Guo et. al [13] took some geometrical parameters of the shell-and-tube heat exchanger as the design variables and the genetic algorithm is applied to solve the associated optimization problem. It is shown that for the case that the heat duty is given, not only can the optimization design increase the heat exchanger effectiveness significantly, but also decrease the pumping power dramatically.

A. Pignotti [14] in his paper established relationship between the effectiveness of two heat exchanger configurations which differ from each other in the inversion of either one of two fluids.

M. S. Bohn [15] in his article presents a method of calculating the electric power generated by a thermoelectric heat exchanger. The method presented in this paper is an extension of the NTU method used to calculate heat-exchanger's heat-transfer effectiveness. The effectiveness of thermoelectric power generation is expressed as the ratio of the actual power generated to the power that would be generated if the entire heat-exchanger area were operating at the inlet fluid temperatures.

P.S. Gowthaman and S. Sathish [1] in their work a comparison is made by analyzing the segmental and helical baffle in a heat exchanger .They found that higher heat transfer and lower pressure drop is achieved in a helical baffle compared to segmental baffle.

Amarjit Singh and Satbir S. Seghval [2] had studied the different effects in Shell and Tube heat exchanger by increasing Reynolds no. with segmental baffles at 0^0,30^0 and 60^0. The model is studied with four segmental baffles. They found that heat transfer coefficient increases with increase in Reynolds no., Nusselt No. increases with increase in Reynolds no.

S.N. Hossain and S. Bari [3] had conducted an experimentally connecting a shell and tube heat exchanger at an exit of diesel engine having specification engine 13B, Tayota made ,4 cylinder Water cooled diesel engine, 102 mm bore and 105 mm stroke, compression ratio 17.6, Torque 217 Nm at 200 rpm. The experiments had been conducted by using HFC, 134a, and ammonia as the working fluid. It is found that it can increase the overall efficiency of diesel engine.

Andre L.H, Costa and Eduardo. m. Queiroz [4] described a method for the optimization of SHTE. The formation of the problem seeks the minimization of thermal surface of the equipment, for certain minimum excess area and maximum pressure drops, considering discrete decision variables.

K. Anand, V.K. Pravin and P.H. Veena [5] had designed the SHTE based on Bell Delaware method.

Chandrkant B Kothare [6] had developed a sophisticated and user-friendly computer software using visual Basic 6.0 (As a primary programming language) for the hydraulic design of SHTE based on the D.Q. Kern method.

Vindhya Vasiny Prasad Dubey, Raj Rajat Varma and Piyush Shankar Varma [7] had designed a simplified model of shell and tube heat exchanger using kerns method to cool the water from 550 c to 450 c by using water at room temperature, and carried out steady state analysis on ANSYS -14, to justify the design.

Lutcha and Nemcansky [8] upon investigation of the flow field patterns generated by various helix angles used in helical baffle geometry found that the flow patterns obtained in their study are similar to plug flow condition which is expected to decline pressure at shell side and increase heat transfer process significantly.

Stehlik [8] studied the effect of optimized segmental baffles and helical baffles in heat exchanger based on Bell-Delaware method and demonstrated the heat transfer and pressure decline correction factors for a heat exchanger.

Oil- Water Shell and Tube Heat Exchangers with various baffle geometries of 5 continuous helical baffles and one segmental baffle and test results were compared for performance with respect to their heat transfer coefficient and pressure decline values at shell side by Kral [9] When they have made comprehensive comparison on the most important geometric factor of helix angle, 40^0 helix angle outperformed the other angles with respect to the heat transfer per unit shell side fluid pumping power or unit shell side fluid pressure decline.

The continuous helical baffles when designed well can prevent the flow induced vibration and fouling in the shell side. Similar results on fouling were reported by Murugesan and Balasubramanian [10].

Methodology

The 3-D model is developed by UNIGRAPHICS and it is analyzed by ANSYS FLUENT-15. RNG k-ε model is chosen for the present study considering the computational time also. The SIMPLE algorithm is used to simultaneously solve the velocity and pressure equations. Kerns method is applied for the thermal designing of the aftercooler. The analysis is conducted by varying the mass flow of the oil from 0.001 kg/s to 0.007 kg/s.

Two Equation Models

Two-Equation Models of turbulence have served as the foundation for much of the turbulence model research during the past two decades. These models provide methods for not only the computation of kinetic energy k, but also for that of turbulence length scale or equivalent.

Consequently, two-equation models are complete, i.e., they can be used to predict properties of a given turbulent flow with no prior knowledge of the turbulence structure. They are, in fact, the simplest complete model of turbulence.

The basic expression for turbulence kinetic energy for the two equation models are as follows:

$$\rho\frac{\partial k}{\partial t}+\rho\bar{u}_j\frac{\partial k}{\partial x_j}=-\tau_{ij}\frac{\partial \bar{u}_i}{\partial x_j}-\rho\epsilon+\frac{\partial\left[\left(\mu+\frac{\mu_T}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right]}{\partial x_j}$$

where τ_{ij} is the turbulence stress tensor, ϵ is the dissipation rate of k.

k-ε Model

The k-ε model focuses on the mechanisms that affect the turbulent kinetic energy (per unit mass) k. The instantaneous kinetic energy $k(t)$ of a turbulent flow is the sum of mean kinetic energy $\bar{k}$ and turbulent kinetic energy k:

$$k(t)=\bar{k}+k$$

where $\bar{k}=\frac{1}{2}(\bar{u}^2+\bar{v}^2+\bar{w}^2)$ and $k=\frac{1}{2}\left(\overline{\acute{u}^2}+\overline{\acute{v}^2}+\overline{\acute{w}^2}\right)$

The dissipation rate of k, ϵ can be written as

$$\epsilon=2v\overline{e'_{ij}e'_{ij}}$$

By assuming suitable closure coefficients**,** ϵ is calculated.

Turbulent dynamic viscosity can be calculated once k and ϵ are known

$$\mu_t=c_\mu\frac{k^2}{\epsilon}$$

Reynolds stresses are calculated further to complete the model closure.

RNG k-ε Model

NG is Renormalized Group. In this model, k- ε equations are derived from the application of a rigorous statistical technique (Renormalization Group Method) to the instantaneous Navier-Stokes equations. They are similar in form to the standard k- ε equations, but include an additional term in ε equation for interaction between turbulence dissipation and mean shear. The effect of swirl on turbulence is considered in this model. Analytical formula for turbulent Prandtl number is additionally included.

For steady, incompressible boundary layers, all of these models can be written compactly as follows: $\bar{u}\frac{\partial k}{\partial x}+\bar{v}\frac{\partial k}{\partial y}=-\vartheta_T\left(\frac{\partial \bar{u}}{\partial y}\right)^2-\epsilon+\frac{\partial\left[\left(\vartheta+\frac{\vartheta_T}{\sigma_k}\right)\frac{\partial k}{\partial y}\right]}{\partial y}$

$$\bar{u}\frac{\partial\tilde{\epsilon}}{\partial x}+\bar{v}\frac{\partial\tilde{\epsilon}}{\partial y}=-c_{\epsilon 1}f_1\frac{\tilde{\epsilon}}{k}\vartheta_T\left(\frac{\partial\bar{u}}{\partial y}\right)^2-c_{\epsilon 2}f_2\frac{\tilde{\epsilon}^2}{k}+E+\frac{\partial\left[\left(\vartheta+\frac{\vartheta_T}{\sigma_\epsilon}\right)\frac{\partial\tilde{\epsilon}}{\partial y}\right]}{\partial y}$$

where ϑ is the kinematic viscosity, $\tilde{\epsilon}$ is defined as $\tilde{\epsilon}=\epsilon-\epsilon_0$, ϵ_0 is the dissipation rate at y=0, $c_{\epsilon 1}$ and $c_{\epsilon 2}$ are the closure coefficients and f_1 and f_2 are the damping functions.

The empirical viscous damping functions in the above equations depends on the following dimensionless parameters,

$$Re_T=\frac{k^2}{\tilde{\epsilon}\gamma},\quad R_y=\frac{k^{1/2}y}{\gamma},y^+=\frac{u_T y}{\gamma}$$

Calculations

HOT FLUID	COLD FLUID
Inlet temperature=403K	Inlet temperature=306.6K
Outlet temperature=303K	Outlet temperature=301K

Inside diameter=251.5 mm, Baffle space=281 mm, Number of passes=1, Number of tubes=199, Length of the tube=1405 mm, Outer diameter=0.9525 cm, BWG=0.9144 mm, Pitch=14 mm, Number of passes=1.

R=$\frac{(T1-T2)}{t1-t2}$, =265.145K, S=$\frac{t2-t1}{T1-t1}$, =255.26K, LMTD=$\frac{\Delta t2-\Delta t1}{2.3\log(\frac{\Delta t2}{\Delta t1})}$, =279.60K, W=5.583kg/s, Q,Air=5.583X60X60 (320.82-315.22), =112553.28W/hr., Q,Water=0.3126x60x60(144.2-44.22), =112513.49W/hr, C^1=Pitch-OD=1.397-0.9525=0.445cm, B=28.09cm, As=$\frac{IDXC1B}{144XPT}$, =0.073m^2

At=0.0610m^2Gs=W/as=20098.8/0.073=275326.02kg/hr.m^2Gt=60466.55kg/hr.m^2V=13.97 m/s, Ta=318.02K, $\boldsymbol{\mu}$,water=.7kg/m.hr, De=0.0139m, Res=(0.0139x275326)/0.7=5467, Ta=367.2K, $\boldsymbol{\mu}$,air=0.076kg/m.hr, D=8.059536x10^{-3} m, Ret=$\frac{DxGt}{\mu}$=(8.59536x10^{-3} x60466.5)/0.0754 =6892, K=6.375x10^{-3} W/cm^2 (^{0}c/cm)=2.335x10^{-3} W/Mk, Jh=$\frac{h0De}{k}\left(\frac{C\mu}{k}\right)^{(1/3)}$ $(\boldsymbol{\mu}/\boldsymbol{\mu}w)^{-.34}$,H0=0.297W/Mk, Hi=$\frac{hixD}{k}$ $(C\boldsymbol{\mu}/k)^{-(1/3)}$ $(\boldsymbol{\mu}/\boldsymbol{\mu}_w)^{-.14}$,Hi=5.46x10^{-3} W/Mk, Hio=4.93x10^{-3}W/Mk,Uc=$\frac{hioxho}{hio+ho}$, =4.854x10^{-3}W/mk, Ud=(Q/A$\boldsymbol{\Delta}$t)=4.76x10^{-3}W/mk

Rd=(Uc-Ud)/(UcxUd)=4.068mk/W

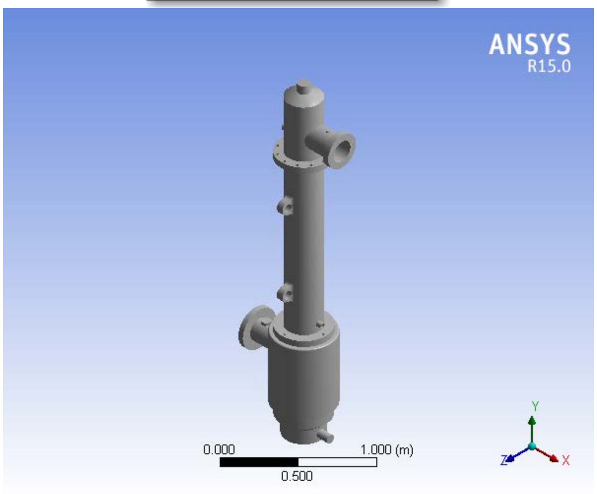

Fig. 1. Aftercooler as Solid

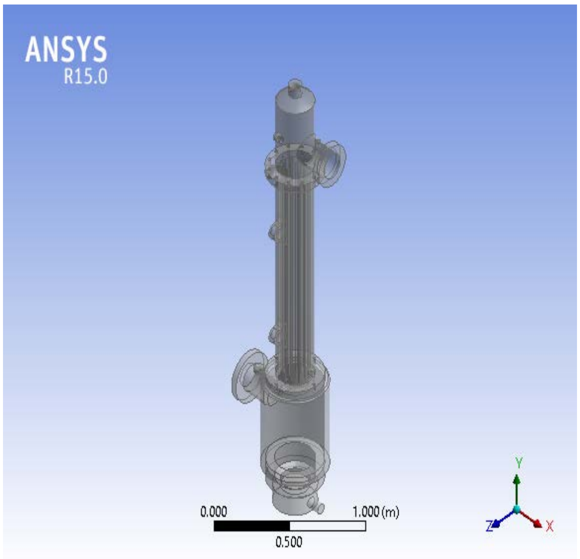

Fig. 2. Aftercooler as Transparent

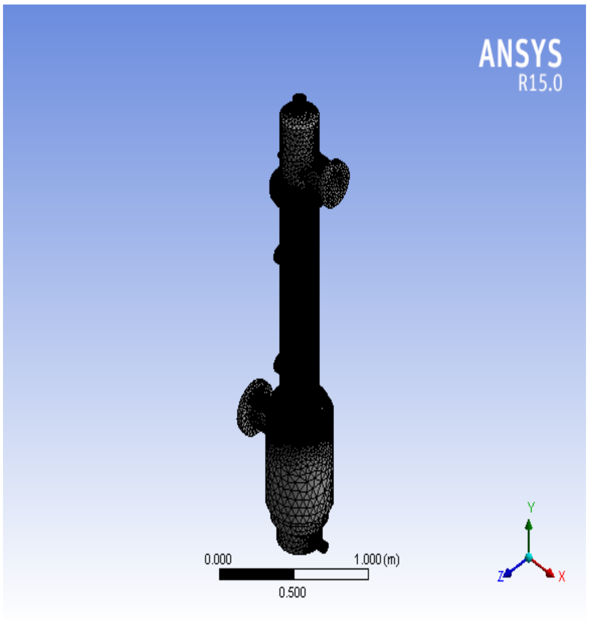

Fig. 3. After cooler in mesh

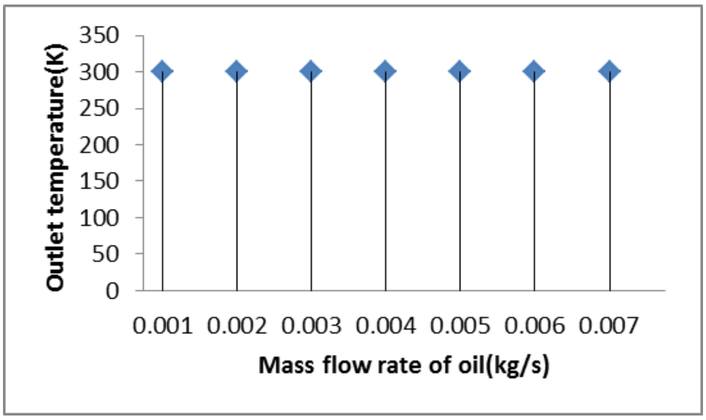

Fig. 4. Mass flowrateVsOutlet temperature

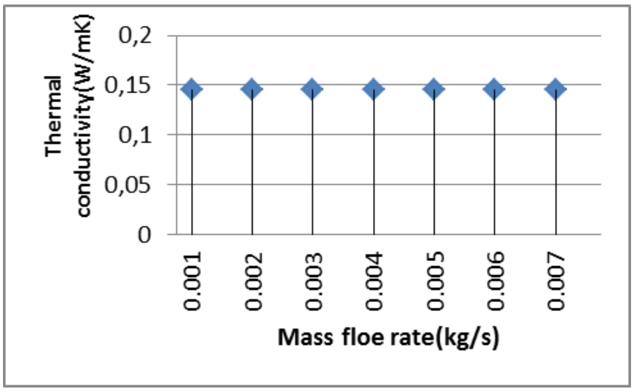

Fig. 5. Mass flow rate Vs Thermal conductivity

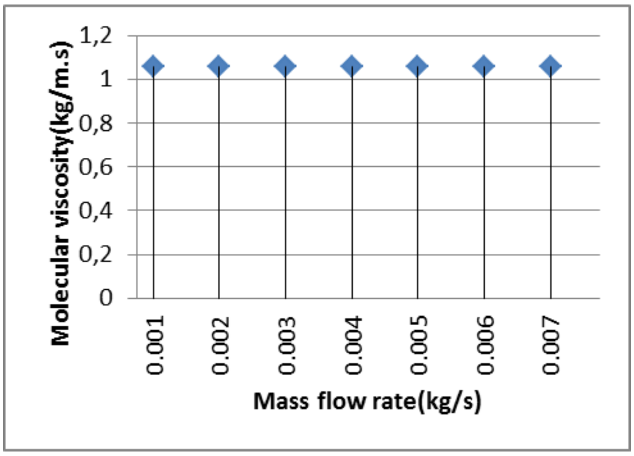

Fig. 6. Mass flow rate Vs Molecular viscosity

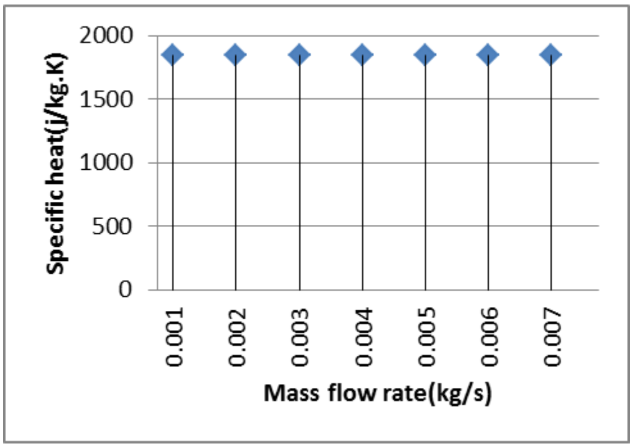

Fig. 7. Mass flow rate Vs Specific heat

Masss flow rate of oil (kg/s)	X-Vorticity (m/s)	Y-Vorticity (m/s)	Z-Vorticity(m/s)	Temperature at out let(K)	Thermal conductivity (k),W/m-k
0.001	1.41E-05	4.68E-05	5.30E-05	300.173	0.145
0.002	1.41E-05	4.68E-05	5.30E-05	300.173	0.145
0.003	1.41E-05	4.68E-05	5.30E-05	300.173	0.145
0.004	1.41E-05	4.68E-05	5.30E-05	300.173	0.145
0.005	1.41E-05	4.68E-05	5.30E-05	300.173	0.145
0.006	1.41E-05	4.68E-05	5.30E-05	300.173	0.145
0.007	1.41E-05	4.68E-05	5.30E-05	300.173	0.145

Molecular viscosity (Kg/m-s)	Specific heat(Cp); j/kg-K	Prandlt No.	Radial velocity (m/s)	Tangentialv elocity (m/s)
1.06	1845.001	13487.59	5.91E-05	1.69E-05
1.06	1845.001	13487.59	5.91E-05	1.69E-05
1.06	1845.001	13487.59	5.91E-05	1.69E-05
1.06	1845.001	13487.59	5.91E-05	1.69E-05
1.06	1845.001	13487.59	5.91E-05	1.69E-05
1.06	1845.001	13487.59	5.91E-05	1.69E-05
1.06	1845.001	13487.59	5.91E-05	1.69E-05

Chart. 2

Results

The thermal designing of the aftercooler of a 2HA 4TERS horizontally balanced opposed air compressor is done. It is obtained that heat flow of air, Qa=112.553KW/hr. Clearance between the tubes c^1=.445 cm. Mass velocity of air Gs=275326.02kg/hr.m^2. Mass velocity of water Gt=60466.55kg/hr.m^2. Reynolds No. of the shell fluid=5467.Reynolds No. of the tube fluid =6892. Thermal conductivity K=2.335x10^{-3} W/mk. Outside heat transfer coefficient ho=.297W/mk. Inside heat transfer coefficient hi=5.46x10^{-3}W/mk .Clean overall heat transfer coefficient Uc=4.854x10^{-3}W/mk. Design overall heat transfer coefficient Ud=4.76x10^{-3} W/mk. Dirt factor Rd=4.068mk/W. From the design data book it is found that the value of Rd obtained is well in agreement. There for the design is correct. Also the effect of the oil content is studied by varying the mass flow rate from 0.001 to 0.007 kg/s. This is done by designing the aftercooler using the 3-D designing software Unigraphics and analyzing it by ANSYS FLUENT-15. The out let temperature of the cold water obtained in this method is 300.179K, which is well in agreement with Kerns method of designing. Also other properties like Thermal conductivity, Molecular viscocity, Specific heat, Prandlt No, Radial velocity, Tangential velocity, Helicity, Vorticity, Enthalpy, Total energy, Turbulent intensity and turbulent dissipation rate is also determined. It is found that there is no effect for oil content towards these parameters, since graph obtained in these cases are a straight line. Different graphs are plotted between the mass flow rate of oil and various outlet properties of the aftercooler.

Conclusion

Design parameters of the aftercooler is verified by Kerns method and also it is analyzed by ANSYS FLUENT. The results obtained by these methods are compared and found that, both of these are well in agreement. The analysis shows that there is no effect for mass flow rate of the oil towards the heat transfer properties of the aftercooler. Further studies can be conducted in the areas of air receiver and the design parameters in order to study the heat transfer properties. The carbon deposit formed inside the tubes of the aftercooler as result of the combustion of air oil mixture and also the hardness of water have the effect on thermal transport properties of the aftercooler. Further studies can be conducted by redesigning and analyzing the air receiver and the aftercooler in order to reduce these effects by optimizing the design.

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