

“Mathematical Validation of Optimized Design of EGR Cooler of CRDI Diesel Engine for NO_x Emission Control using CFD Analysis”

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Abstract

Euro 6 is a significant advancement with regard to NO_x limits. The NO_x limit declines from 0.18 g/ km to 0.08 g/km, a reduction of 56% as compared to Euro 5 norms. This has significant implications for control technologies, requiring for the first time the integration of emission control aftertreatment for NO_x emissions like improvising exhaust gas recirculation (EGR) coolers. Exhaust Gas Recirculation coolers (EGR) have to withstand high thermal stresses due to large temperature gradients between the hot exhaust gas and the coolant. Two different designs of EGR coolers are analyzed using CFD technique to determine effectiveness. The design of EGR cooler is shell and tube type having single entry for coolant in first design and multiple entry for coolant in second design in which the coolant passes through the shell to cool down the exhaust gas inside tubes. The thermo-flow characteristics inside the EGR Cooler and surface temperature have been investigated also by using CFD code ANSYS CFX 14.0. Standard k-epsilon turbulence model is used for analysis.

Keywords — EGR Cooler, NO_x, CFD

I. INTRODUCTION

Engine emissions are toxic gases that cause health problems to humans and damage the environment. To decrease these emissions, a number of laws have been implemented throughout the world, limiting the amount of these emissions that a vehicle may emit. It is possible to reduce the formation of emissions during the combustion to comply with exhaust emission legislation. The raw emissions are reduced and thus no or less aftertreatment is needed. A portion of the exhaust gases is recirculated into the combustion chambers. This can be achieved either internally with the proper valve timing, or externally with some kind of piping, Figure 9 shows this schematically. In a spark-ignited engine running, the recirculated exhaust gases are virtually inert as they contain almost no oxygen. In a diesel engine, the

exhaust gases always contain oxygen but their oxygen concentration is lower than that of fresh air. They have instead a larger concentration of the inert three-atomic gases carbon dioxide and water vapor. By mixing the exhaust gases with the intake air, the oxygen concentration of the cylinder charge is lowered. This leads to a reduction of the combustion temperature by different effects. As the air is diluted with exhaust gas, the mass of a gas portion containing the needed amount of oxygen gets bigger. The fuel molecules need to mix with a larger volume of gas to find the oxygen molecules they need to react.

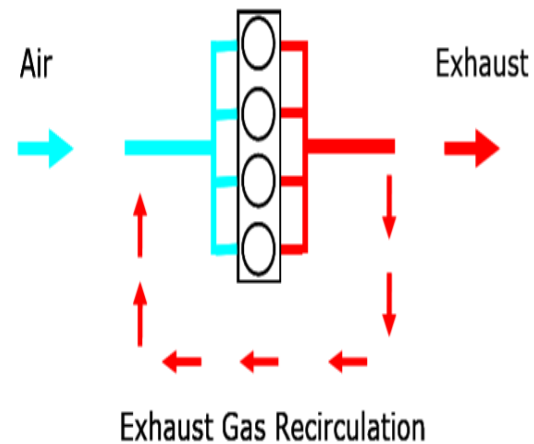


Figure 1: Exhaust Gas recirculation

The energy of the same fuel amount is thus used to heat up a larger gas portion than it would without EGR, resulting in a lower temperature. Another effect is the change in heat capacity.

Exhaust gas has a higher specific heat capacity than air, due to the CO₂-molecule's higher degree of freedom. So for the same amount of combustion heat, a gas mass containing EGR will have a lower temperature than pure air. The lower combustion temperature directly reduces the NO_x formation, as the NO_x formation rate is highly temperature dependent.



Figure 2: EGR cooler

The exhaust gas drawn off for recirculation has a temperature of around 650 degrees Celsius. It is therefore far too hot to be fed directly into the cylinders; it would increase the temperature of the combustion chamber even further, thereby defeating its actual purpose — that of reducing nitrogen oxide formation by lowering the combustion temperature. For this reason, the exhaust gas is first cooled to around 120 degrees Celsius.

II. LITERATURE REVIEW

Simon Reifarhith [1] It gives an overview of the field of EGR and diesel combustion and presenting the methods used in this work. This work provides a simulative comparison of different EGR systems, such as long-route EGR, short-route EGR, hybrid EGR, a system with a reed valve and a system with an EGR pump. Both the steady-state performance and transient performance are compared.

Jaffar Hussain et al [2] studied the effect of EGR on performance and emissions in three cylinders, air cooled and constant speed direct injection diesel engine, which is usually used in agricultural farm machinery. Such engines are normally not operated with EGR. The experiments were performed to experimentally evaluate the performance and emissions for different EGR rates of the engine.

B. Jothithirumal et al [3] conducted experiment on the effect of exhaust gas recirculation on the exhaust gas temperature. The experimental set up for proposed experiments was developed on two-cylinder direct injection air cooled compression ignition engine experiment was conducted for observing the effect of different quantities of EGR on exhaust gas temperature.

Deepak Agarwal et al [4] conducted a test on a single cylinder DI diesel engine and calculated the performance and emission characteristics with rice bran methyl ester (RBME) and its blends as fuel with EGR system. They optimized and reported that 20% biodiesel blends with 15% EGR produce less NO_x, CO and HC emissions and also enhanced thermal efficiency and reduced BSFC.

B. Deet al [5] In this paper an experimental study is carried out on an I.C. engine laboratory single

cylinder, four-stroke VCR, direct injection diesel engine to analyze the performance and emission characteristics of pure diesel, Jatropha oil and Jatropha oil-diesel blended fuels with different blended rates. The measurements are recorded for the compression ratio of 16, 17 and 18 varying the load from idle to rated load of 3.7 kW. Comparative results are given at constant engine speed, variable compression ratio and various engine loads for pure diesel, Jatropha oil and Jatropha oil-diesel blended fuels revealing the effect of diesel, Jatropha oil and Jatropha- diesel blended fuels combustion on engine performance and exhaust emissions.

Zhang et al. [6] investigated the effect of diesel soot deposition on the performance of a small 6-tube shell and tube heat exchanger by operating the engine at medium load which produced an exhaust gas temperature of around 250 °C. They found that fouling increased the thermal resistance and pressure drop by 150% during 12 hours of exposure. The rate of increase of the thermal resistance decreased over this period and approached an asymptotic value.

III. PROPOSED WORK

The overall objective of this research is to improve effectiveness of EGR by varying shell design using techniques of Computational Fluid Dynamics to achieve better heat transfer rate and effectiveness. The software used for analysis is ANSYS CFX and CAD modeling is done using Creo 2.0. The base design for analysis is taken of CRDI engine.

IV. METHODOLOGY

The research analysis is performed using Computational Fluid Dynamics technique where CAD modeled is developed and imported in ANSYS CFX solver where suitable boundary conditions is applied and solved. The analysis involves three stages, first is pre-processing, solution and post processing stages. A pre-processor is used to define the geometry for the computational domain of interest and generate the mesh of control volumes (for calculations). Generally, the finer the mesh in the areas of large changes the more accurate the solution. Fineness of the grid also determines the computer hardware and calculation time needed. The solver makes the calculations using a numerical solution technique, which can use finite difference, finite element, or spectral methods. Most CFD codes use finite volumes, which is a special finite difference method. First the fluid flow equations are integrated over the control volumes (resulting in the exact conservation of relevant properties for each finite volume), then these integral equations are discretized (producing algebraic equations through converting of the integral fluid flow equations), and finally an iterative method is used to solve the algebraic equations. The post-processor provides for visualization of the results and includes the capability

to display the geometry/mesh, create vector, contour, and 2D and 3D surface plots. Particles can be tracked throughout a simulation, and the model can be manipulated (i.e. changed by scaling, rotating, etc.), and all in full coloured animated graphics.

Table 1: Design specification for EGR

Parameters	EGR 1
No. of tubes	28
No. of baffles	3
Tube length	180
Total length	230
Inlet coolant side diameter	19.5
Outlet coolant side diameter	19.5
Inlet exhaust side diameter	30
Outlet exhaust side diameter	30

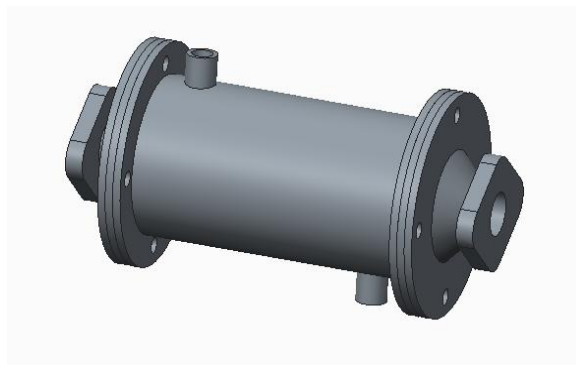


Figure 3: EGR cooler of single entry

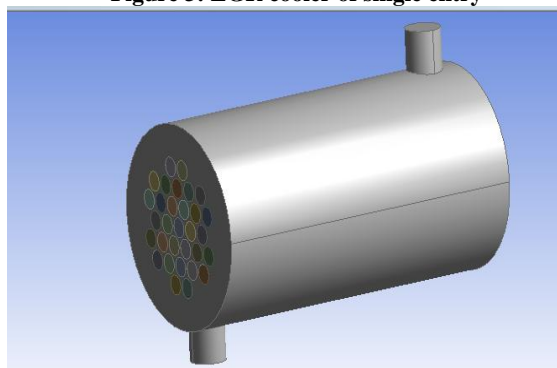


Figure 4: Computational domain of single entry EGR

Table 2: Engine specification for CRDI engine

Parameters	Description
Type	Inline Five cylinder, 4 stroke, CRDI engine
Power	90 KW @ 2950 rpm

Peak torque	320Nm@1400-2400 rpm
Bore	X Stroke 90.5X 90.9
Compression ratio	16.5:1

V. DATA REDUCTION

The parameters which were of key interest were outlet temperature, outlet pressure, velocity outlet. Design of heat exchanger is done by LMTD method and tubular exchangers manufacturers association standard. Also permissible amount of pressure drop is measured by related equation

In the LMTD method heat transfer is calculated by following method

$$A = Q / U_o * \Delta T_m$$

Logarithmic temperature difference can be calculated as :

$$\Delta T_2 = T_{h1} - T_{c2}$$

$$\Delta T_1 = T_{h2} - T_{c1}$$

$$\Delta T_m = \frac{\Delta T_1 - \Delta T_2}{\ln(\frac{\Delta T_1}{\Delta T_2})}$$

Overall heat transfer coefficient can be calculated as:

$$U_o = \frac{1}{\frac{d_o}{d_i} * h_i + d_o * \frac{\ln(\frac{d_o}{d_i})}{2k} + \frac{1}{h_o}}$$

Bell Delaware method is used to calculate heat transfer coefficient. Shell side heat transfer coefficient can be calculated as:

$$h_o = j_i * C_p * (ms/As) * (K/Cp * \mu_s)^{(2/3)} * (\mu/\mu_s * w)^{0.14}$$

Tube side heat transfer coefficient can be calculated as:

$$Nu = \frac{(\frac{f}{2}) * Re * Pr}{1.07 + 12.7 * (\frac{f}{2})^{1/4} * ((Pr^{1/4} * 0.5) - 1)}$$

The CAD model of EGR cooler is modeled using Creo 2 software which is sketch based, feature based parametric 3d modeling software developed by PTC .

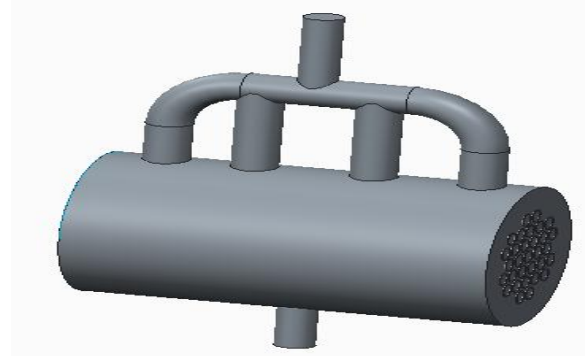


Figure 5: CAD model of multiple entry EGR

The meshing for is carried out using ANSYS mesher. Here element shape is taken as brick and fine sizing with relevance 100. Other meshing parameters such as smoothing is set to medium , transition slow, curvatuve angle 12^0

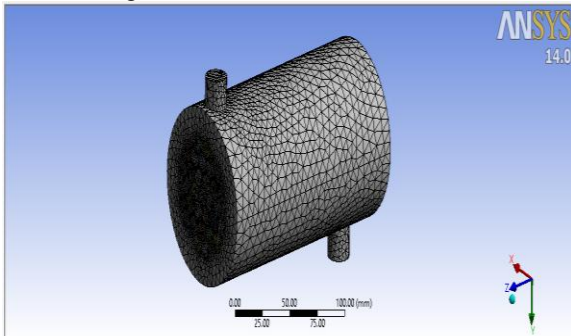


Figure 6: Meshed model of single entry EGR cooler

The upwind scheme “second order upwind scheme” was selected for momentum and energy equations. In order to couple velocity and pressure, the SIMPLE algorithm was applied. Temporal discretization was achieved using the solution method “Implicit integration”. Standard scheme was utilized to interpolate pressure and the relaxation factors for pressure, density, body forces, momentum and energy were maintained at 0.3, 1, 1, 0.7 and 1 respectively. A low convergence criterion of 10^{-4} was chosen for energy equation and 10^{-3} was chosen for continuity and velocity equation for the residuals in order to accurately predict different parameters. The solution was initialized by computing from the inlet using “Standard Initialization” option. After the completion of these settings, the iteration procedure was initiated by clicking on ‘Run Calculation’ button.

VI. RESULTS AND DISCUSSION

The CFD analysis conducted on EGR cooler with single entry and multiple entry is simulated and contour plot of temperature, pressure and velocity are plotted as shown in section below. As engine works at different RPM thereby resulting in different emission gas temperatures. To analyze the cooling of EGR cooler different operating temperatures has been taken viz 419°C , 589°C , 688°C , 786.2°C .

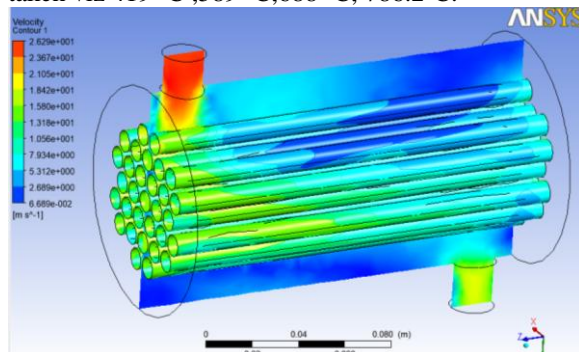


Figure 7: Velocity plot for 1st design EGR cooler

The velocity contour along with mid-section plane is shown in fig 7 above. The velocity of EGR

coolant is highest at inlet section of EGR cooler which reduces considerably as we move towards shell side of EGR cooler and increases towards exit shown by yellow colour. Velocity increases and then decreases as we move across the shell. The pressure contour along with mid-section plane is shown in fig 8. The pressure value of EGR is highest at inlet section of EGR cooler which reduces considerably as we move towards shell side of EGR cooler and increases towards exit shown by blue colour. The pressure value remains almost constant towards middle right zone of shell.

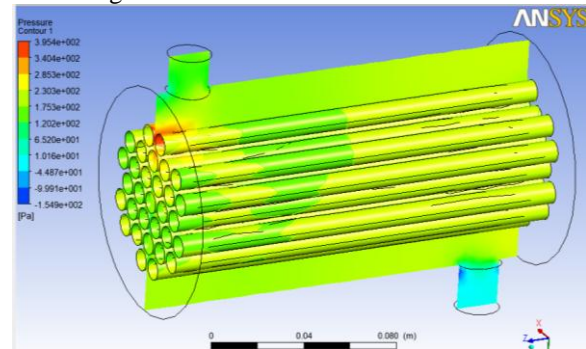


Figure 8: Pressure plot for 1st design EGR cooler

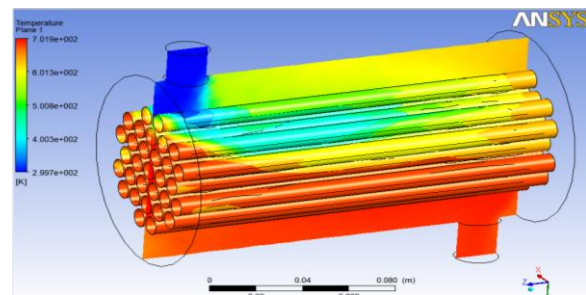


Figure 9: Temperature plot for 1st design EGR cooler

The temperature contour along with mid-section plane is shown in fig 9 above. The portion of tubes containing EGR gas which is in immediate contact with coolant shown by dark blue colour and light blue colour has the lowest temperature while the portion of pipe away from coolant inlet shows higher temperature shown by dark red colour. Subsequent CFD analysis is conducted on 2nd design EGR cooler with multiple inlet entries and contours are plotted.

At low proportions of entrance diameter to shell diameter (D_1/D_2), flow percent decreases at side tubes and as a result, more flow is passed into centrals. This causes pressure reduction at central tubes and cooling water temperature increment near them, which can make boiling at section. This local boiling causes vibration and stresses. Also flow rate decrement and subsequently turbulence decrement at side tubes cause local heat transfer rate reduction. This could result in reduced effectiveness and lifetime of cooler. Therefore, we can't add more inlet tubes by reducing diameter. The maximum number of tubes fitted for inlet is four. The subsequent analysis is conducted using 4 inlet tubes discussed in next section.

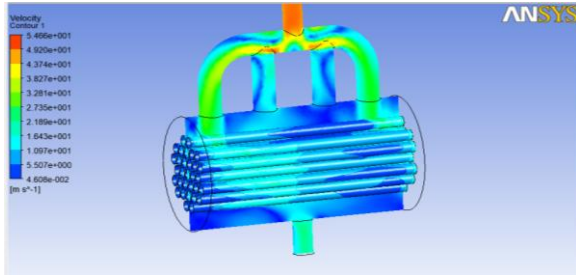


Figure 10: Velocity plot for 2nd design EGR cooler

The velocity contour along with mid-section plane is shown in fig 10. The velocity of EGR coolant is highest at inlet 1st and 4th section of EGR cooler shown by green color contour which reduces considerably as we move towards shell side of EGR cooler and increases towards exit shown by yellow color. Velocity increases and then decreases as we move across the shell.

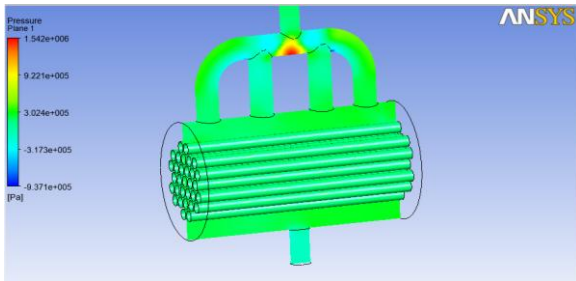


Figure 11: Pressure plot for 2nd design EGR cooler

The pressure contour along with mid-section plane is shown in fig 11. The pressure value of EGR is highest at inlet section of EGR cooler shown by red and yellow color contour which reduces considerably as we move towards shell side of EGR cooler and increases towards exit shown by blue colour. The pressure value remains almost constant towards middle zone of shell. Shown by green color.

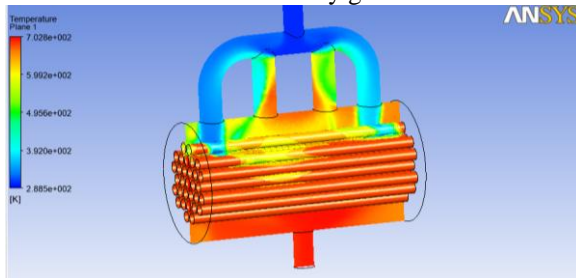


Figure 12: Temp. plot for 2nd design EGR cooler

The temperature contour along with mid-section plane is shown in fig 12 above. The portion of tubes containing EGR gas which is in immediate contact with coolant shown by dark blue colour and light blue colour has the lowest temperature while the portion of pipe away from coolant inlet shows higher temperature shown by dark red color.

Table 3: Temp table for single entry EGR cooler gas

GAS IN TEMP(K)	692.92	862.85	961.0	1059.2
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GAS OUT TEMP(K)	357	482.85	551.0	689.26
COOLANT IN TEMP(K)	341	341	341	341
COOLANT OUT TEMP(K)	349.26	352.58	353.4	350.65

Output data of EGR cooler with multiple entry coolant temperature at inlet and outlet, gas temperature at inlet and outlet is provided in table 4

Table 4: Temperature table for single entry EGR cooler coolant

OPERATING TEMP(K)	692.92	862.85	961.05	1059.26
GAS IN PRESSURE(Pa)	129938	129938	129938	129938
GAS OUT PRESSURE(Pa)	129445	129445	129445	129445
COOLANT IN PRESSURE(Pa)	105645	105645	105645	105645
COOLANT OUT PRESSURE(Pa)	105390	105390	105390	105390

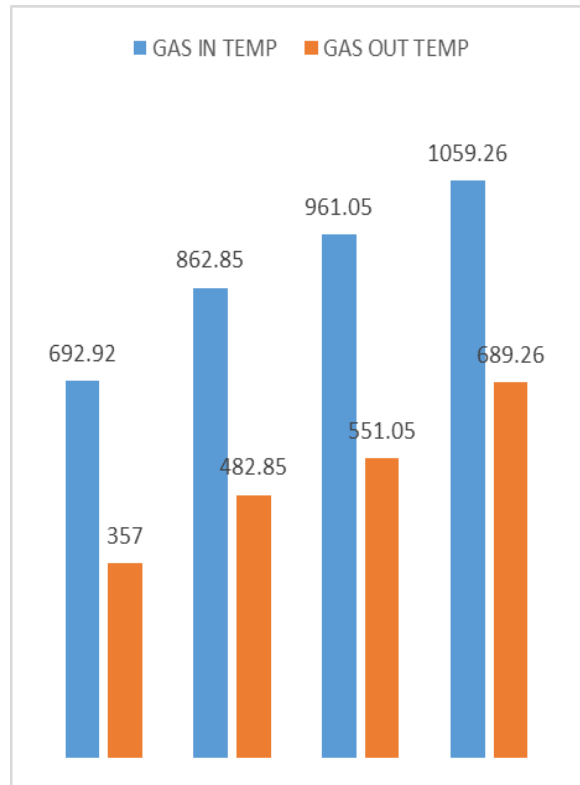


Figure 13: Gas inlet and outlet temperature graph of single entry EGR cooler

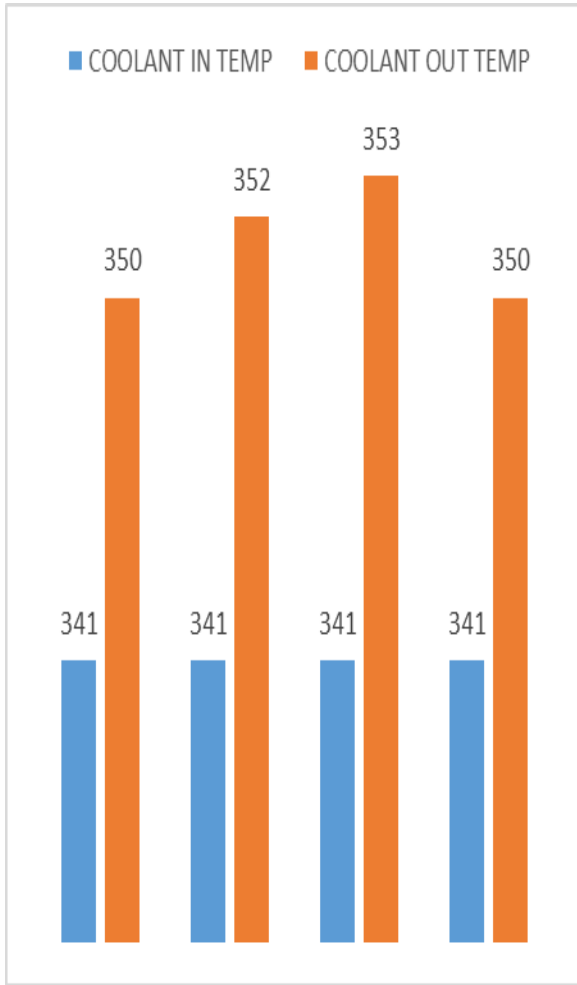


Figure 14: Coolant inlet and outlet temperature graph of single entry EGR cooler

Table 5: Pressure table for multiple entry EGR cooler gas

GAS IN TEMP(K)	692	862.85	961.05	1059.26
GAS OUT TEMP(K)	348	465	539.21	677.52
COOLANT IN TEMP(K)	300	300	300	300
COOLANT OUT TEMP(K)	354	356	354	351

Subsequent analysis has been carried on 2nd design having multiple inlet entry for coolant using ANSYS CFX under the same boundary conditions. The results are discussed in next section

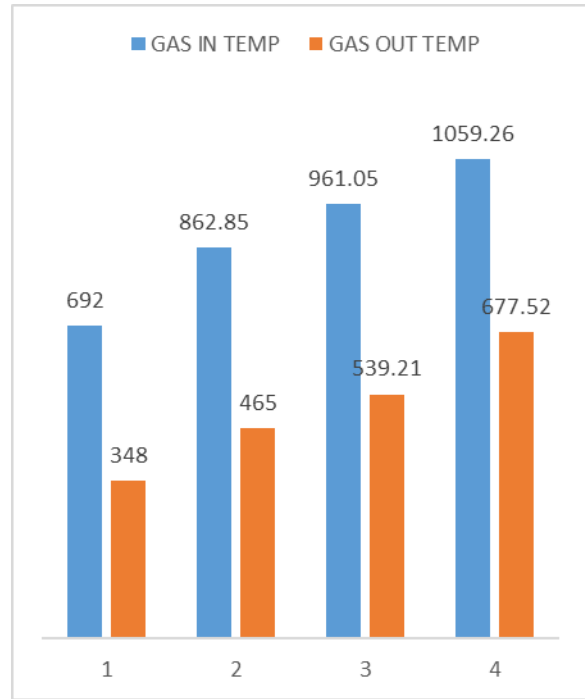


Figure 15: Gas inlet and outlet temperature graph for multiple entry EGR cooler

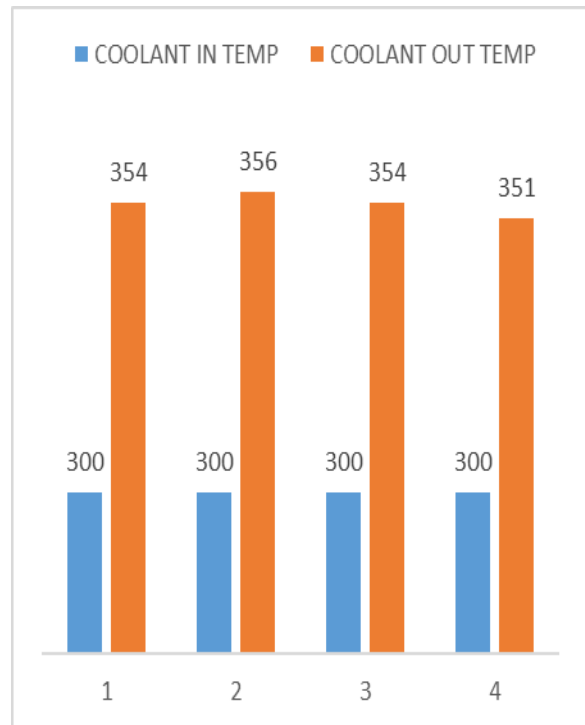


Figure 16: Coolant inlet and outlet temperature graph for multiple entry EGR cooler

As it can be noticed from comparison chart above the outlet EGR gas temperature is lower for 2nd design (multiple entry) for all operating temperature as compared to 1st design (single entry) therefore 2nd design is better as compared to 1st design in terms of EGR cooling.

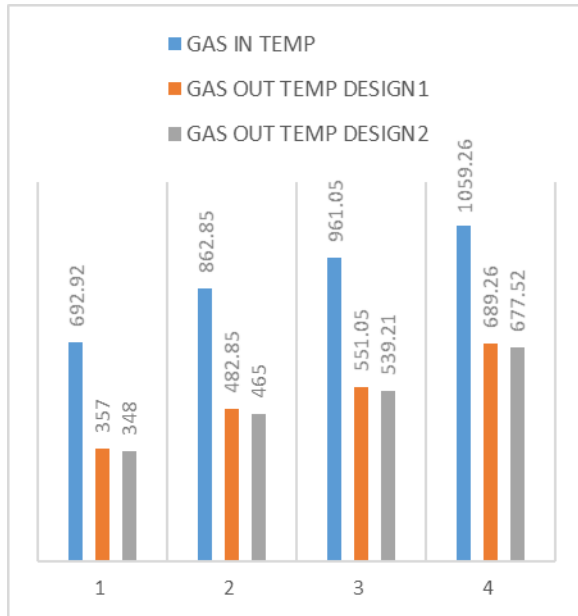


Figure 17: Coolant inlet and outlet temperature graph for of multiple entry EGR cooler

VI. CONCLUSION

The EGR (exhaust gas recirculation) technique can greatly reduce the NO_x emission of diesel engines, especially when an EGR cooler is employed. Numerical simulations are applied to study the flow field and temperature distributions inside the EGR cooler. Two designs of EGR cooler are analyzed using ANSYS CFX under same operating conditions. First design is based on single entry of coolant while second design is based on multiple entry of coolant in EGR cooler. It has been found that 2nd design of EGR cooler with multiple inlet for coolant performs better in terms of cooling effectiveness as compared to 1st design with single inlet for all operating temperatures. This can be attributed to better heat transfer achieved in 2nd design due to more turbulence and turbulent flow heat transfer.

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