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Thermal and Fluid Simulation of a New Diesel Engine Cooling Exhaust Gas Recirculation System to Reduce Exhaust Gas Emissions

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ARTICLE INFO	ABSTRACT

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Recalculating the exhaust gas into engine intake is a method that can considerably reduce the amount of NO_X pollutant. By using the exhaust gas recirculation cooler (EGR Cooler), the efficiency of this method increase. The computational simulation is an exact method to analyse the heat distribution and pressure changes in EGR Cooler. In this study, the heat transfer characteristics of two types of new EGR Cooler were numerically investigated. The results of this simulation revealed When the U- shaped EGR Cooler containing wavy tubes perpendicular to the gas flow direction is used, the heat transfer coefficient improve of 25% compared to wavy tubes parallel to the gas flow direction. This research also revealed that the amount of pressure drop in full load in the two mentioned EGR Cooler is about 11kpa, but in the part load, the amount of pressure drop in both systems is so low. Therefore the use of the new cooling system has many benefits, in term of pressure drop and heat transfer efficiency.

Keywords:
Thermal Simulation and Modeling, NOX Pollutant, EGR Cooler, Exhaust emissions.

1. Introduction

With the world in concerns of fossil fuel depletion, more researches have been done in order to utilize and optimize other alternative of energy sources [1]. Also, Environmental and economic sustainability have drawn the attention of researchers and prompted them to seek alternative methods that offer maximum energy at low cost [2]. Considering some of the advantages of diesel engines such as: low fuel consumption, high combustion efficiency, less amount of produced CO_2 and high durability, the diesel engines are utilized in most of fields and duties [3-6]. Because of the some contaminants emissions such as NO_X and PM (Particulate Matters), the diesel engines are considered as serious threats to human health and environment [3, 7]. So in the recent years restricting laws were enforced globally to decrease these pollutants [8]. Therefore many solutions were presented to reduce diesel engine emissions [9, 10]. One of these solutions is to recalculate some amount of exhausting gas to air intake which is used in most of the modern diesel engines. In the case of EGR

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some amount of exhausting gas is returned to input that aims to decrease the amount of free oxygen in cylinder and increase specific heat capacity and decrease combustion temperature peak which leads to less nitrogen oxides formation [10-13]. Since the temperature of EGR is high, this high temperature cannot reduce the combustion temperature peak in cylinder. So in order to decrease the temperature of EGR, the EGR cooler is used [6, 14, 15]. Accomplished researches reveal that cooling EGR has a great effect on decreasing diesel engine pollutants [3, 16]. In the recent years, many studies were accomplished to optimize the EGR cooler efficiency. Most of the studies include changes in contact surface between coolant and EGR tube. Hang and others used numerical research simulated fluid flow and heat distribution in 3 types of EGR cooler. Their study also revealed that in spiral cover models water goes through a longer pathway, thus it causes more heat transfer between coolant and EGR tube [17].Kim and others During an experimental and computational study found that by increasing the contact surface between the coolant and EGR tube, the amount of heat transfer will increase and also because of contact surface increase and better gas mixture while crossing the tube in spiral tubes, better heat transfer takes place in comparison with flat tubes [6]. Li and others studied about 3 types of EGR cooler and showed that in fin coolers with smaller pitch, heat transfer coefficient between the coolant l and EGR is more than comparison with a coolant with larger fin pitches [18].

The results of CFD analysis on two types of EGR coolant systems with flat tubes and fin tubes revealed that in fin type a better heat transfer takes place in comparison with flat tube. The reason for this event is the contact surface increase in the fin type rather than flat type [19]. Li and others studied about 3 types of EGR cooler and revealed that in fin cooler with smaller pitch, heat transfer coefficient between coolant and EGR is more in comparison with coolant with larger fin pitch and tube-shaped type [20].

A review of the conducted researches showed that heat transfer efficiency plays a major role in the performance of EGR cooler system. Accomplished studies showed that most of the coolers used in EGR cooler systems are designed according to variation in contact surface between the tube containing EGR and coolant. In most of these coolant systems, cover layers are used inside the coolant tank which aims at increasing time of the coolant liquid contact with the tubes containing EGR. Since using such equipment in cooling systems needs exact equipment and cost of production of these cooling systems is so high and also their designing and producing is so time consuming, therefore in the present study a new type of EGR cooler system named EGR is U-shaped was simulated. The U-shaped tubes in the present study have the advantages such as heat transfer improvement, simple and easy design, Low cost of production, and Low thermal variation in the coolant.

According to the mentioned advantages for U-shaped tubes, in the present study two types of U-shaped EGR cooler were simulated for different engine operating conditions. The computational fluid dynamics in finite volume method (FVM) was used. The heat distribution and the pressure inside the tubes were completely simulated. In order to indicate the advantages of the U-shaped EGR cooler, most of the operating parameters of U-shaped EGR cooler were compared with the other types of EGR coolers. Non-commercial ANSYS CFXV15 software was used to simulation. All the computational tasks were performed in a personal computer with 16 GB RAM and Intel (R) Core (TM) i7-4750 CPU @ 3.30 GHz.

2. Methodology
2.1 Modelling
2.1.1 Model description

In the EGR cooler used in the present study to carry EGR includes two models (A and B). A-type contains wavy tubes which are used to form the waves perpendicular to EGR direction. B-type contains wavy tubes which are used to form the waves Parallel to EGR direction. Details of the two models are shown in Figure 1, 2 and 3.

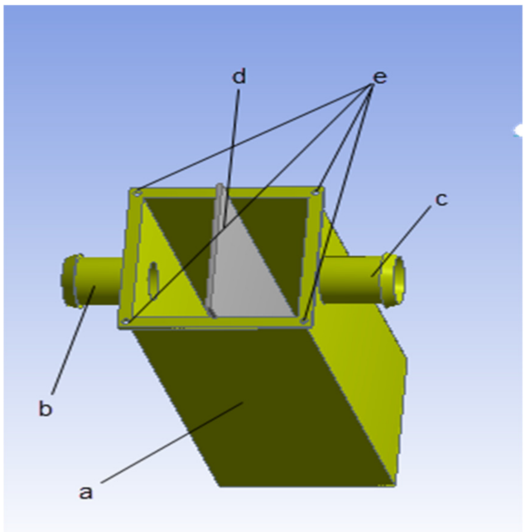


Fig. 1. Configuration of coolant sys. a: shell; b: coolant inlet; c: coolant outlet; d: Sliding door Separator inlet and outlet coolant section; e: Clapboard

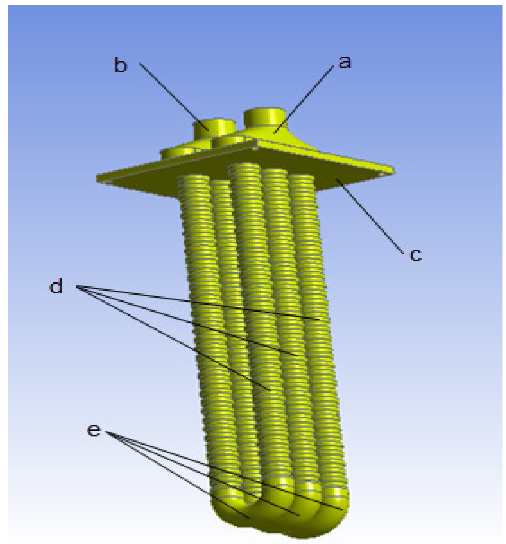


Fig. 2. Configuration of A type. a: inlet EGR chamber; b: outlet EGR chamber; c: Clapboard; d: Wave's tube; e: Smooth U type tube

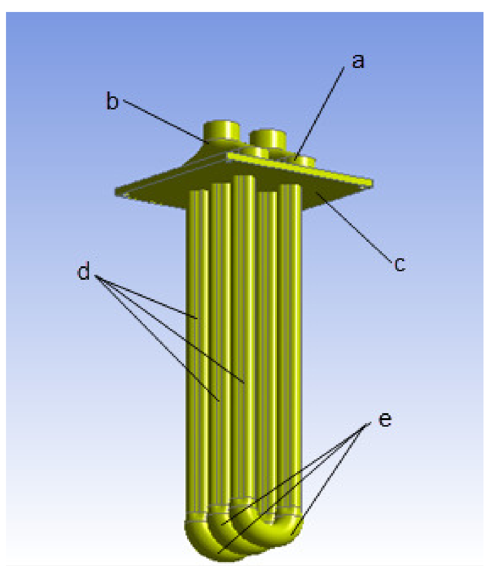


Fig. 3: Configuration of B type. a: inlet EGR chamber; b: outlet EGR chamber; c: Clapboard; d: Wave's tube; e: Smooth U type tube

To stimulate heat transfer, total energy model was used for EGR simulating and for the coolant, thermal energy model was used. The heat transfer models are used to predict temperature distribution in the flow. Since EGR fluid is Compressible, total energy model is used to simulate heat transfer. This type of heat transfer simulates enthalpy and also contains kinetic energy effects, meanwhile the EGR fluid is Compressible, and the best model to heat transfer is total energy. The coolant is incompressible, thus in this part to model heat transfer the thermal energy model was used.

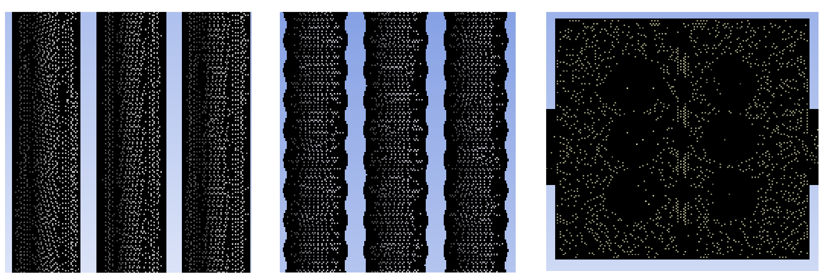
Turbulence simulate is one of the three basic pillars of CFD. An ideal turbulent model should have the least amount of complexity and also predict the nature of correlated physics. In the present study, K-epsilon standard model was used to simulate the turbulence. Since the k-epsilon model is strong, economic and exact, to stimulate flow and heat transfer in industry, it is so preferred in a wide range of turbulent flows. This model is a semi experimental one and extracting its equations is based on empiricism and phenomenological considerations. The K-epsilon standard model is based on heat transfer equations modeling for kinetic energy turbulence (k) and its dissipation rate (ε).

2.1.2 Grid generation

The grid generation of EGR cooler for A-type and B-type was accomplished by analysis MESH software. To mesh different parts of EGR Cooler, tetrahedral and hexahedral mesh were used with the best proximity. Since the number of elements in this grid generation is considered with high accuracy, the sufficient time for grid generation was raised considerably. The Specifications of both A and B-type was shown in table 1. A view of both A and B-type grid generation is shown in figure 4.

Table1
Meshing Specifications of A and B model

Model	Proximity Accuracy	Elements	Nodes
A	1	4778161	1007886
B	1	12644396	2546592


(a) (b) (c)
Fig. 4. Computational grid of models a: A-type; b: B-type; c: coolant

2.1.3 Determining domains and boundary conditions

Whereas the two investigated EGR coolers in the present study include 3 parts (EGR, coolant and gas carrier tubes), to simulate different parts, three domains were used. To simulate EGR domain, the ideal gas was used as EGR. To stimulate coolant, water was used as coolant. Also to stimulate solid parts which include gas carrier tubes and coolant tank, steel was used. One important part in computational simulation is creating accurate and strong boundary conditions which can do simulation with the least error. In the present study, to consider boundary conditions, pressure and temperature were used as input and for outlet boundary conditions, mass flow rate was considered.

The intended structure to create boundary conditions in the present study is so strong that can analyze equations with high accuracy. One advantage of this structure is obtaining heat and pressure in the output of fluid domains. The boundary conditions used in this study are shown in table 2.This considered two engine operating conditions.

Table 2
Boundary conditions

Boundary conditions	Quantities	
Engine operating condition	1630 rpm	2950 rpm
EGR inlet mass flow rate (kg/h)	25.5	64.5
EGR inlet temperature (˚C)	430	500
Coolant inlet mass flow rate(l/h)	800	1400
Coolant inlet temperature (˚C)	95	95

2.2 Mathematical Model

The actual heat transfer processes occurring in an EGR cooler include: (1) the convective heat transfer between the EGR and tube wall; (2) the convective heat transfer between the coolant and tube wall (3) the conduction of heat in the tube wall; and (4) the conduction of heat in the shell and the ground. For the coupling pressure- velocity the governing equations of problem are solved by the finite volume method (FVM), based on the algorithm SIMPLEC [21]. Unsteady mathematical model is formulated in three-dimensional Cartesian coordinate system employing the k–ε turbulence model.

2.2.1 Controlling equations

The controlling equations of the heat transfer model consist of the steadiness equation, the momentum equation, the energy equation and the standard k-ε equation. The general transport equations that describe the principle of conservation of mass, momentum and energy can be expressed in the following conservative form [22, 23].

$$\frac{\partial(u_i)}{\partial x_i} = 0 \tag{1}$$

$$\frac{\partial}{\partial t}(\rho u_i) + \frac{\partial}{\partial x_j}(\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}[(\mu + \mu_T] \tag{2}$$

$$\frac{\partial}{\partial t}(\rho T) + \frac{\partial}{\partial x_j}(\rho T u_j) = \frac{\partial}{\partial x_j}\left[(\frac{\mu}{Pr} + \frac{\mu T}{\sigma})\frac{\partial T}{\partial x_j}\right] \tag{3}$$

The k and ε transport equations of the standard k–ε turbulence model used are given by 4 and 5 equations [23].

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_i}(\rho . k u_i) = \frac{\partial}{\partial x_j}\left(\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right) + G_k + G_b - \rho\varepsilon + S_k \tag{4}$$

$$\frac{\partial}{\partial t}(\rho \varepsilon) + \frac{\partial}{\partial x_i}(\rho . \varepsilon . u_i) = \frac{\partial}{\partial x_j}\left(\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial \varepsilon}{\partial x_j}\right) + \left(C_{1\varepsilon}\frac{\varepsilon}{K}(G_k + C_{3\varepsilon}G_b)\right) - \left(C_{2\varepsilon}\rho\frac{\varepsilon^2}{k} + S_\varepsilon\right) \tag{5}$$

In the above equations:

$$\mu_T = \rho C_\mu \frac{k^2}{\varepsilon} \tag{6}$$

$$G_k = \mu_i \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) \frac{\partial u_i}{\partial x_j} \tag{7}$$

The constants appearing in Equations (2)-(7) take the values given in Table 3.

Table 3
The values of the constants in the k-ε model [21]

C_μ	$C_{1\varepsilon}$	$C_{2\varepsilon}$	α_k	α_ε
0.09	1.44	1.92	1	1.3

According to the physical parameter and velocity of coolant and EGR, The turbulent kinetic energy k, turbulent dissipation rate ε and Prandtl number Pr are calculated by ANSYS CFX automatically [24-30].

3. Results and Discussions

3.1 Temperature Distinction in EGR

In this study, two EGR coolers were simulated. For each one, two engine operating conditions were considered. The heat discrepancies between EGR inlet and outlet are shown in Figure 5. This figure shows that when engine is working at low speed, the heat transfer from EGR to coolant in both EGR cooler (A and B-type) is rather the same. When the engine was operating at high speeds, the rate of heat transfer in A-type rises. When the rate of EGR and coolant increase, the rate of heat transfer in A-type increases. Rising EGR mass rate, the speed of gas flow in EGR carrier tubes will increase. Since the contact surface in B-type is smooth and flat, it doesn't affect better mixture and also doesn't create turbulence in fluid flow, but in A-type because the direction of wave is perpendicular to EGR direction, by rising EGR speed combination will be more while moving in the track and heat transfer takes place better. So, heat transfer in LC2950rpmoperation condition in the A type is %21 more than B type in this operating condition.

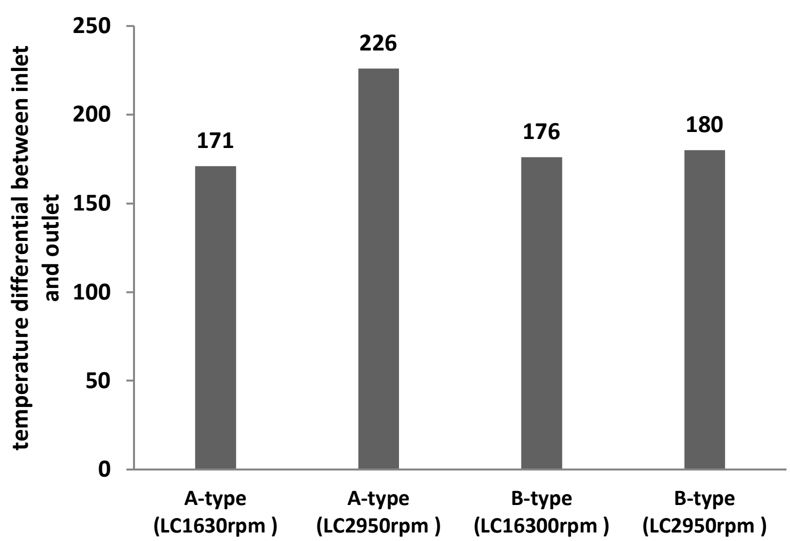


Fig. 5. Heat discrepancies between inlet and outlet of EGR

In Figure 6 temperature distribution counter is shown in both operating conditions. Considering the date of Figure 5, it is clear that in A-type for LC2950 rpm operating conditions temperature in sweeping track of U-shaped tube decrease slowly, but in A-type in the same operating condition in the sweeping track heat transfer will takes place better and the temperature decreases. In the returning track in A-type heat transfer continues rather with high speed, but in the B-type in this track of heat transfer, it decreases. Using this counter it became clear that at low engine speeds and low EGR, the amount of heat transfer in A-type is less than that of B-type. But when the engine operates at a high speed (LC2950rpm) and EGR amount increases, the rate of heat transfer in A-type is much more than that of B-type.

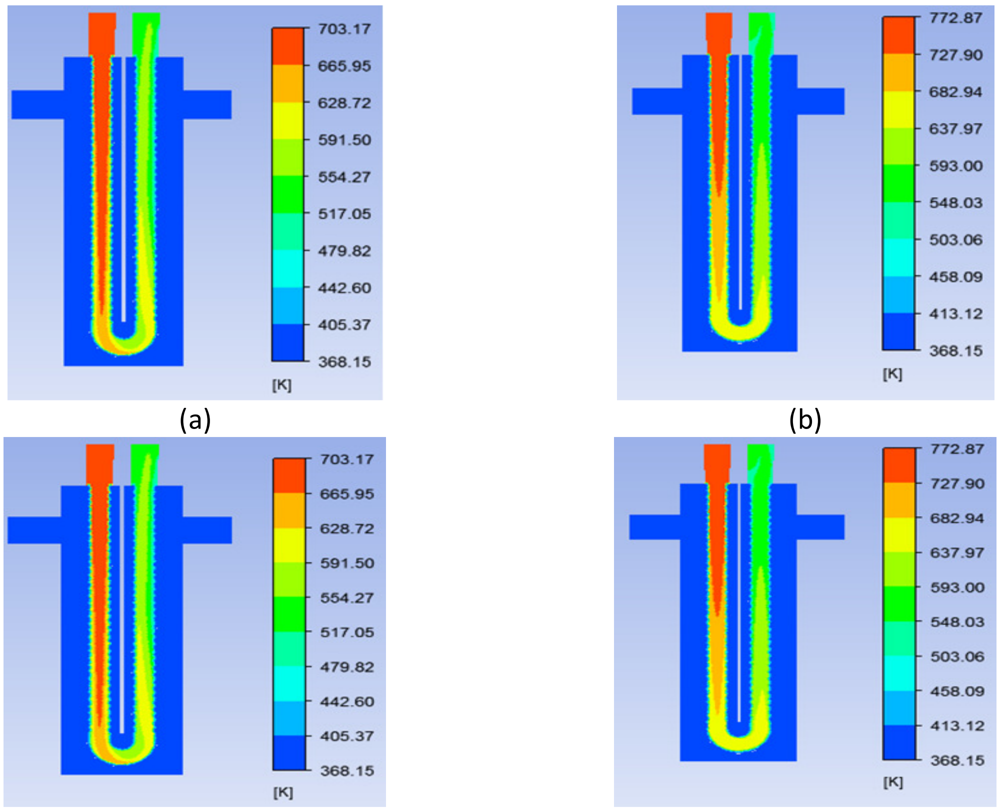


Fig. 6. Counter of Temperature distinction in EGR, (a): A-type (LC1630 rpm); (b): A-type (LC2950 rpm); (c): B-type (LC16300 rpm); (d): B-type (LC2950 rpm).

3.2 Temperature Difference in The Coolant

Considering that the coolant of EGR cooler is obtained from the coolant liquid of engine, the temperature variation in coolant outlet is very important. If the engine coolant temperature increases, it causes bad effects on engine cooling, and if the engine is not completely cooled, it causes hazard to different parts of the engine. In Figure 7 the temperature of coolant outlet in A-type and B-type with two operating conditions of engine is shown. In both models of cooler, the amount of temperature increases in coolant output and is so low, approximately 1˚C. So this increase in temperature does not affect cooling of the engine.

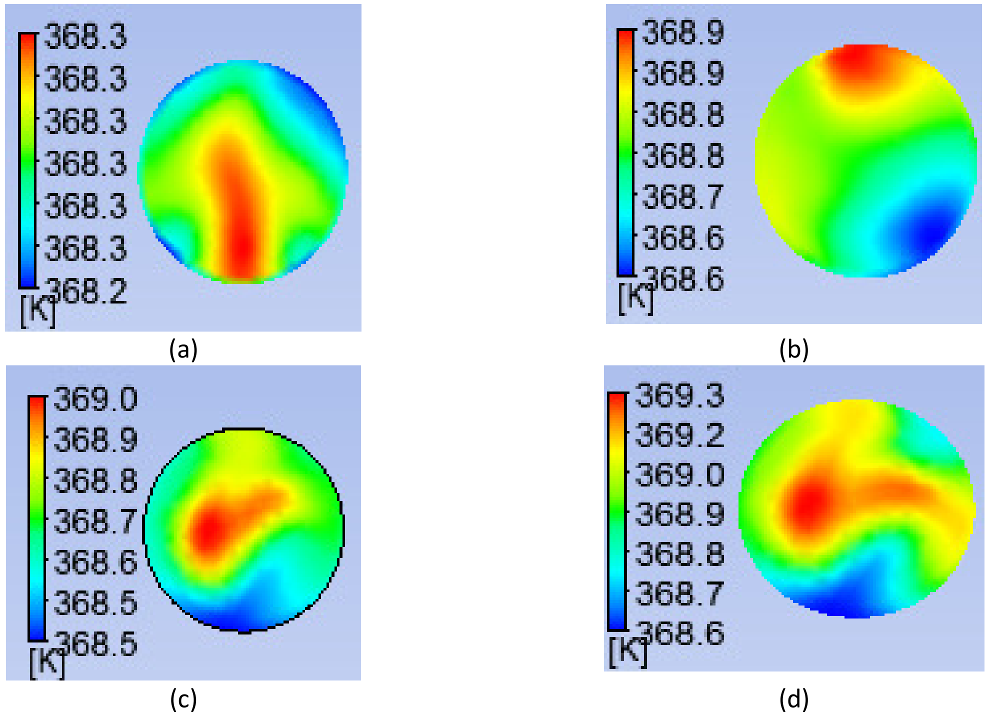


Fig. 7. Temperature counters in the coolant. (a): A type (LC1630 rpm); (b): A type (LC2950 rpm); (c): B type (LC16300 rpm); (d): B type (LC2950 rpm)

3.3 Pressure Difference in The EGR Cooler

Pressure variation is one of the most important issues in designing and operating EGR cooler in a diesel engine. Considering that low pressure of intake air causes a decrease in volumetric efficiency of the engine, while using an EGR Cooler it must be taken into account that pressure drop in outlet of EGR doesn't increase. Figure 8 shows the variation of pressure drop in flow of the EGR for the mass flow rate in two operating conditions. This figure shows that in LC2950 rpm the rate of pressure drop in A-type is more than of the B-type. The designing method of contact surface of the gas with the tubes is so that has a special effect on pressure drop. Since in the A-type, the method of wave creation on the EGR carrier tubes is perpendicular to flow track, the pressure drop in A-type is more than that of B-type. By increasing mass flow rate of EGR in LC2950rpm operating conditions, designing method of cross section tube carrying EGR is more effective on the pressure drop through the Path, and also because of the U-shapes of the models, the pressure fall increases sensibly. The pressure drop is due to EGR direction change in the path. According to Figure 6 it is evident that the high mass rate of EGR in U-shaped tube has a greater influence on the increase in pressure drop, but in low operating condition, the pressure drop in the track is less and does not affect the volumetric efficiency of diesel engine. Pressure variation counter in EGR through the track is shown in Figure 9.

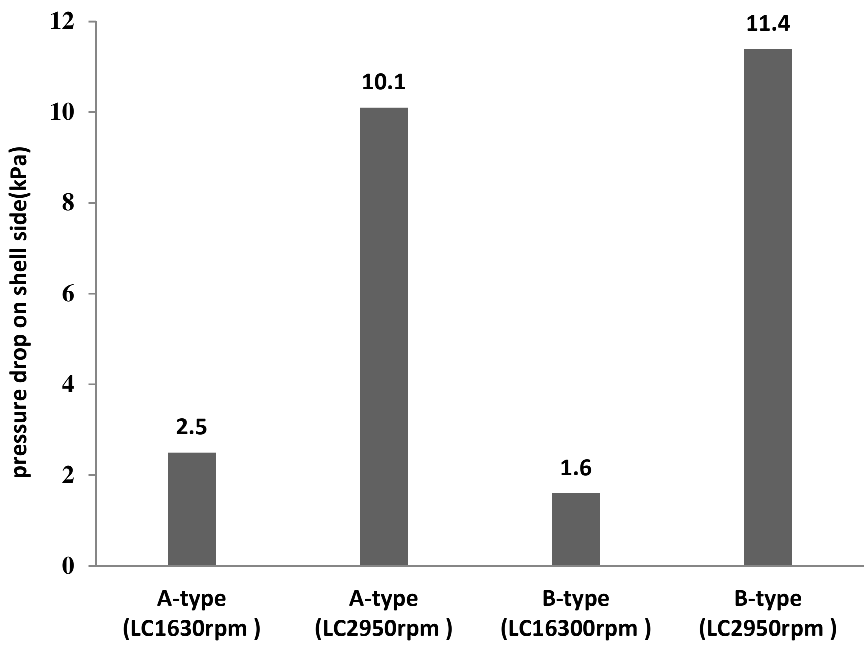


Fig. 8. Pressure drop in the EGR flow path

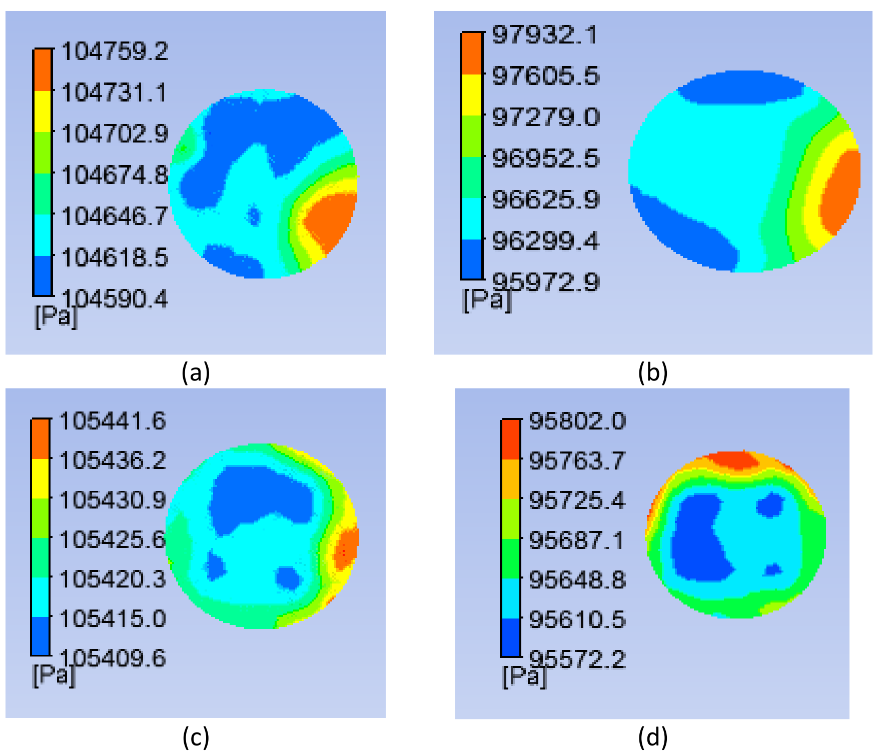


Fig. 9. Pressure counters in the EGR outlet, (a): A-type(LC1630 rpm);(b): A-type(LC2950 rpm); (c):B-type(LC16300 rpm) ; (d):B-type(LC2950 rpm)

3.4 Heat Transfer Efficiency

One way to evaluation the efficiency of EGR cooler is calculating the thermal efficiency, which is calculated by Equation (8).

$$\eta_{Thermal} = \frac{\Delta T_{EGR}}{\Delta T_{max,Fluids}} = \frac{T_{in,EGR} - T_{out,EGR}}{T_{in,EGR} - T_{out,Coolant}} \tag{8}$$

where, $\eta_{Thermal}$ Is Heat transfer efficiency, $T_{in,EGR}$ is inlet temperature of EGR, $T_{out,EGR}$ is outlet temperature of EGR, and $T_{out,\ Coolant}$ is outlet temperature of the coolant.

The rate of heat transfer for A-type and B-type was calculated in two operating conditions. In Figure 10 the calculated heat transfer efficiency is shown. According to Figure 10 it is evident that while mass rate of EGR is low, the amount of thermal efficiency in A-type and B-type is rather the same. By increasing mass rate of EGR, the amount of thermal efficiency in A-type is more than that of B-type. Because of the EGR surface contact with tubes and mixture of gas through the track in A-type is more than the B-type.

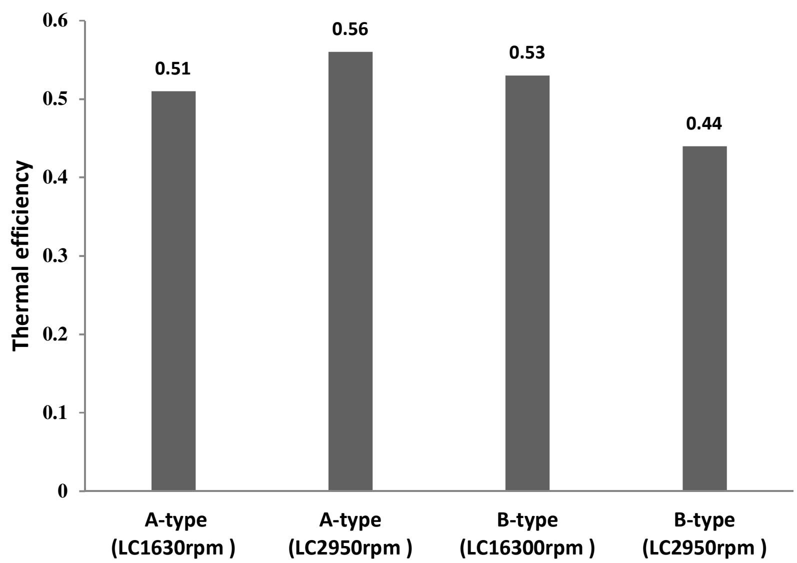


Fig.10. Thermal efficiency of A-type and B-type

4. Conclusions

The present study includes thermal simulation in 2 new types of EGR cooler. In this study temperature distribution, heat transfer and pressure variation in 2 represented EGR cooler was simulated and these parameters were compared. The main conclusions drawn from this are as follows.

1. In high speed operating and also in the high mass flow rate of EGR, heat transfer in A-type is 23% more than that of B-type.
2. In low speed operation pressure drops in both A-type and B-type is low, but in high speed operating the pressure drop increases in both models. In this working condition pressure drop in A-type is 4% less than that of B-type.

3. One of the limitations of the present study is that the study is merely a computational simulation. For verification of the results, experiments should be carried out on 2 types of EGR cooler which is beyond paper.

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