

# Optimum Design of Heat Exchanger in Diesel Engine Cold EGR for Pollutants Reduction

Nasser Ghassemlou, Armin Rahmatfam, Faramarz Ranjbar

**Abstract**—Using cold EGR method with variable venturi and turbocharger has a very significant effect on reduction of NO<sub>x</sub> and grime simultaneously. EGR cooler is one of the most important parts in the cold EGR circuit. In this paper optimum design of cooler for working in different percentages of EGR and for determining optimum temperature of exhausted gases, growth of efficiency, reduction of weight, dimension, expenditures, sediment and also optimum performance by using gasoil which has significant amounts of brimstone are investigated and optimized.

**Keywords**—Cold EGR, NO<sub>x</sub>, Cooler.

## I. INTRODUCTION

THE most important pollutants of Diesel engines are Nitrogen oxides (NO<sub>x</sub>) and grime particles. One of effective methods for reduction of NO<sub>x</sub> pollutants is exhausted gases restoration (EGR) into combustion chamber; this action could be possible by three following methods: Thermal, Chemical and Dilution. [1]

By cooling EGR gases (Cold EGR), more percentage of exhausted NO<sub>x</sub> pollutants will be decreased. In cold EGR, restored gases are cooled by a heat exchanger and amount of absorbed heat by EGR is more [2], [3]. Also by decreasing the temperature of restored gases durability of EGR valves and other parts which are depended on EGR will be more.

In diesel engines, most of produced nitrogen oxides are from thermal type; increasing combustion temperature will induce rate of NO production. On the other side, carbon suspended particles are productions of imperfect combustion and low temperature. However, the most important factor in reducing pollutants and estimating new environmental standards is to reduce nitrogen oxides and grime simultaneously.

To reduce nitrogen oxides and particles with no brunt to fuel economics, engine systems should be designed and controlled carefully to acquire optimum conditions for temperature, inlet pressure, time and amount of fuel injection, and also quantity and schedule of input EGR. Prediction of all optimization actual variables is a very complicated and important obligation.

Numerous experiences on different EGR systems for heavy diesel engines, in scope of performance effects investigation

on special fuel consumption and pollution, reveal that in comparison with low pressure EGR system (synthesis of hot exhausted gases and input air before turbocharger) or in the other words long circuit EGR system cold, variable and short circuit EGR method (synthesis of high pressure exhausted gases and high pressure input air) with variable venturi, turbocharger and increased injection pressure (common rail), is the most effective method in NO<sub>x</sub> reduction and obtaining minimum amount of specific fuel consumption. Also short circuit EGR on the contrary with long circuit one obstructs sediments of compressor and intercooler (if there is); on the other side injection retardation increases specific fuel consumption [4], [5]. Fig. 1 exhibits these two systems.

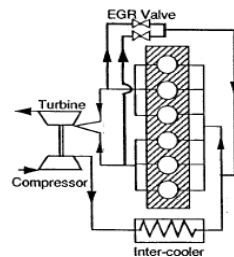


Fig. 1 (a) High pressure EGR System

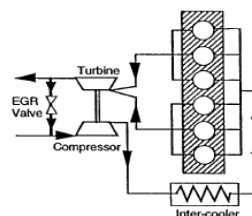


Fig. 1 (b) Low pressure EGR System

Briefly about NO<sub>x</sub> and grime reduction in cold EGR we can mention two important factors; firstly reduction of input load temperature lengthens combustion retardation and conclusively results in more homogeneity between fuel steam and oxygen synthesis; secondly mass of trapped air in cylinder in IVC is increased and this induces  $\lambda$  increment and oxygen concentration. Also cylinder pressure increment causes effective pressure optimization which will optimize fuel consumption.

One of important parts in cold EGR is heat exchanger design and manufacturing which is used to cool the gas. EGR coolers are mostly in shell- tube type [6]-[8]; in this case gas flows through tubes and coolant (engine jacket water) flows in

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shell. This facilitates tubes cleanliness and also control of cooler exhausted gas temperature.

Inspected results reveal that although number of tubes increment will increase shell dimension, which subsequently decreases heat transfer coefficient, will increase heat exchanger heat transfer area and decrease pressure drop [7]. However, this will increase weight and production expenditures.

Using passive heat transfer methods like spiral tubes will result in heat transfer efficiency increment and heat exchanger length decrement [9]. Fig. 2 shows four types of enhanced tubes and a simple tube and also their comparisons with regard to heat transfer rate, pressure drop and cost of manufacturing [8]. About heat transfer increment we should know that spiral tubes have more efficiency in comparison with other extended tubes, also they have appropriate production cost and pressure drop.






		✓+ Best	✓ Good	✓- OK	✗ Not Good
Symbol	Tube Configuration	Heat Transfer Rate	Pressure Loss	Production Cost	
a	<div>Straight</div> 	✗	✓+	✓+	
b	<div>Dimple</div> 	✗	✓-	✓	
c	<div>Ribbon</div> 	✓+	✗	✗	
d	<div>Spring</div> 	✓+	✓-	✗	
e	<div>Spiral</div> 	✓+	✓	✓	

Fig. 2 Comparison of different tubes which are used in heat exchangers [8]



Fig. 3 Specimen of spiral tube [10]

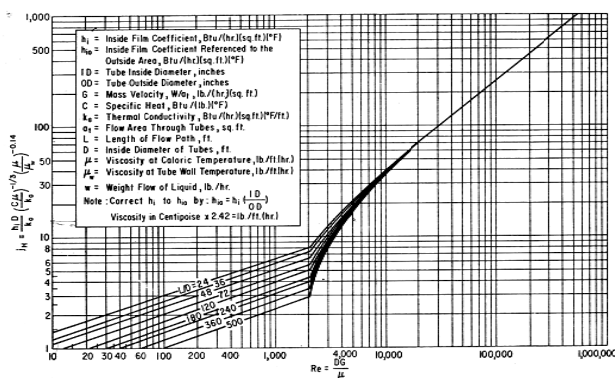


Fig. 4 Heat transfer coefficient in tube by Tate & Sieder [12]

Using spiral tubes is a very effective method to equalize fluid speed in shell, obstruction of local boiling and thermal stresses, and also intensifying perpetuity of cooler [8].

Pressure drop in gas side and subsequently speed decrement of gas in tubes is a restrictive design factor. With regard to different percentages of EGR flow Reynolds number ranges between 2000 and 9000, it is in transient flow field which doesn't have any precise method to calculate heat transfer coefficient. So estimating friction coefficient by using analytic methods won't give any trusted quantity.

Also we cannot calculate actual pressure drop during entrance diffuser. Moreover a stagnation pressure point will be created at the return section. This will happen at the end of diffuser and at the striking section of flow to the wall and striking of boundary layer at the conical section which have a very important effect on the amount of crossing flow, particularly in the side tubes. Therefore in scope of investigation of diffuser entrance diameter effects on the uniform and effective distribution of returning gases into all tubes, performance of cooler, some calculations are done by CFD method.

Significant portion of pressure drop in diffuser is due to flow shear stress at the returning section.

Designed cooler in comparison with typical coolers [6]-[8] has higher efficiency despite of rational pressure drop.

## II. EQUATIONS AND SOLUTION METHODS

Governing equations encompass continuity, momentum, energy and finally correlated pressure drop and heat transfer relations. Because of various design parameters presence such as arrangements, sizes, number of passages, type of tubes and also applying different EGR percentages trial and error method has been used which brings complexity in design; therefore a code has been written at Fortran 90.

Elementary design for different EGR percentages firstly has been done by LMTD method and tubular exchanger manufacturers' association standard (TEMA); and has been probed and proved by the number of transfer units' ( $\epsilon - NTU$ ) method. Also permissive amount of pressure drop has been investigated by related equations in tubes.

At LMTD method heat transfer area is calculated by following equation:

$$A_{to} = \frac{Q}{U_o F \Delta T_{LN}} \quad (1)$$

$\Delta T_{LN}$  is LMTD of Counter flow and is described as following:

$$\Delta T_{LN} = \frac{(T_{h1} - T_{c2}) - (T_{h2} - T_{c1})}{\ln((T_{h2} - T_{c1}) / (T_{h1} - T_{c2}))} \quad (2)$$

Principal part of EGR cooler design is calculating  $U_o$ . This coefficient is depended on heat transfer coefficient of shell and tube sides and also fouling resistance.

$$\frac{1}{U_o} = \frac{1}{1/h_o + A_i R_i / A_o + t_w / k_w A_w + A_i / A_o h_i + R_o} \quad (3)$$

$$A_w = \frac{D_o - D_i}{\ln(D_o/D_i)} \quad (4)$$

In cooler tubes, flow is in transient conditions and there is no precise solution to calculate heat transfer coefficient. One of the best graphs for calculating heat transfer coefficient in tubes is Tate & Sieder which is shown in Fig. 4.

We can arrive to following relations by fitting these curves in this section for different proportions of  $L/D$ .  $L$  is the length of tube and  $D$  is tube internal diameter.

$$J_H = 0.0044 \text{Re} - (2.363 \ln(L/D_e) - 6.46) \quad (5)$$

$$2100 \leq \text{Re} \leq 10000$$

$h_i$  is heat transfer coefficient in tube and is acquired by following equation:

$$J_H = \frac{h_i D_e}{k} \left( \frac{c_p \mu}{k} \right)^{-1/3} \varphi^{-1} \quad (6)$$

where:  $J_H$  is J-Colburn coefficient,  $\varphi$  is correction factor for viscosity depended on temperature.

In the  $(\varepsilon - NTU)$  method effectiveness  $(\varepsilon)$  is described as following:

$$\varepsilon = \frac{Q}{Q_{\max}} \quad (7)$$

$$NTU = \frac{AU_m}{C_{\min}} \quad (8)$$

Basic equation for calculating mean heat transfer coefficient in shell side is explained on basis of Bell-Delaware method [12] as following:

$$h_o = h_{id} J_c J_l J_b J_s J_r \quad (9)$$

where  $h_{id}$  is heat transfer coefficient for completely crossing flow with ideal tube set and  $J$ 's are correction factors for seals and other parts:

$$h_{id} = j_i c_{ps} \left( \frac{\dot{m}_s}{S_m} \right) \left( \frac{k_s}{c_{ps} \mu_s} \right)^{\frac{2}{3}} \left( \frac{\mu_s}{\mu_{s,w}} \right)^{0.14} \quad (10)$$

Factors calculations and complete information about heat transfer coefficient and also pressure drop calculations in shell and tube sides are described in [11] and [12].

For figuring out number of tubes, if shell diameter be less than 230 mm following relations are used. For triangular arrangement equation (11) and for quadrangular and turned quadrangular arrangement equation (12) is used.

$$N_t = \frac{3\lambda}{4} \left( 3 \frac{D_s - D_o}{2P} \right) \left( \frac{D_s - D_o}{2P} + 1 \right) + 1 \quad (11)$$

$$N_t = \frac{3\lambda}{4} \left( 0.886 \frac{D_s - D_o}{P} + 1 \right)^2 \quad (12)$$

For one and two tubes passes  $\lambda$  is 1 and 0.92 respectively. Calculations reveal that number of tubes for triangular arrangement and circular arrangement is equal.

### III. SIMULATION AND RESULTS VALIDATION

With regard to problems such as transient flow in tubes, shortness of cooler, effects of entrance length and what is mentioned at previous sections, using computational fluid dynamics method (CFD) to estimate flow specimens, temperature distribution and pressure in cooler is very advantageous. Computational fluid dynamics software which has been used in this research is Fluent. In simulations pressure basis algorithms are used for continuous solving of converted differential equations which are in numerical form. The used algorithm is SIMPLE. In every section of solving independence of its trend from mesh is assessed.

To validate numerical results, firstly a comparison between experimental data [7] and numerical results of cooler-tube simulation has been done; after comparisons and validation, to calculate heat transfer and pressure drop, simulations for three shapes of entrance for a whole cooler have been run. In Fig. 5 experimental and numerical results through tube length for two sections of center and 0.9 mm near wall are exhibited. In this simulation results has been acquired for inlet gas at 250°C of temperature and 0.25 g/s of flux; also temperature of cooler water has been supposed to be permanent at 90°C and exhausted gas specifications have been supposed to be the mean specifications of combustion productions.

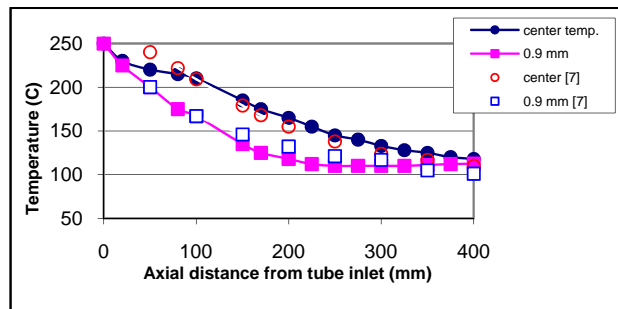


Fig. 5 Gas temperature diversion through a simple tube

In the next section numerical analysis for the whole cooler and three types of diffuser (cylindrical, nuzzle and conical) with flows of 100 g/s, 150 g/s and 200 g/s of flux has been done and compared with experimental results. Results comparisons in form of pressure drop on the basis of flow for a quarter prototypes are showed in Fig. 6. Internal diameter of shell is 78 mm and number of tubes is 70. Chart shows a good coincidence between experimental and numerical results and maximum deviation is about 7%.

Spiral tubes usage prompts growth of heat transfer efficiency in tube side and reduction of size and cooler weight. However, it will induce pressure drop increment in tube side. Some optimizations have been done to reduce this negative effect about an acceptable amount. Spiral tube effective design parameters on flow specifications and heat transfer are tube pitch, spiral angle, spiral height and number of spirals.

As it can be figured out from Fig. 8 spiral wall induces fluid turning near wall; fluid velocity near wall exceeds plain tube velocity and boundary layer is thinner than plain tube. Subsequently heat transfer coefficient near wall is larger and as a result of this heat transfer rate is greater than simple tube. Results have been acquired for a tube with internal diameter of 7.8 mm, pitch of 1.4 mm, length of 200 mm and gas flow of 2.21 kg/h.

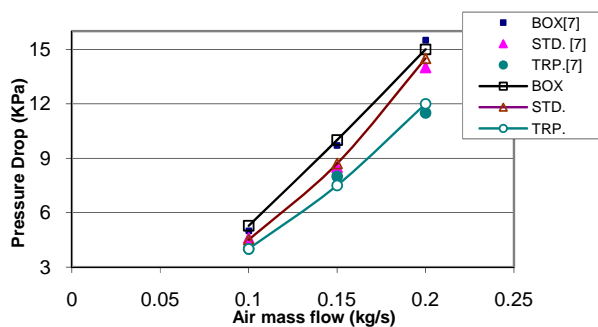


Fig. 6 Pressure drop related to flow rate

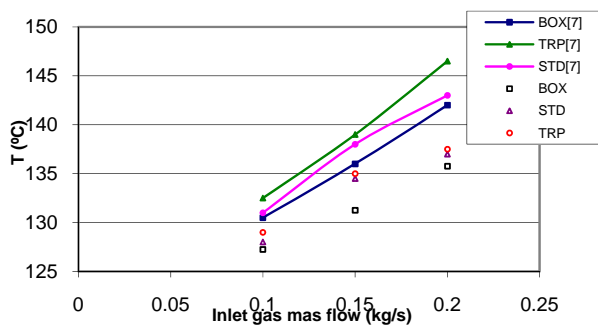


Fig. 7 Outlet temperature of cooler related to flow rate

With regard to negative effects of pressure drop on heat transfer rate in tubes, effects of spiral number increment on pressure drop and heat transfer have been probed. As shown in Fig. 9 increment of spiral numbers from one to two or four with constant hydraulic diameter, induces central flow symmetry and decreases fluid velocity in comparison with one way spiral tube.

Also by increasing heat transfer area significant decrement has not been seen in thermal efficiency in comparison with one way spiral tube.

Obtained results are shown in Figs. 10 and 11.

Reduction of spiral pitch and increment of spiral height cause a growth in heat transfer coefficient.

Increment of sediments at the tube wall decreases total heat transfer coefficient; particularly because of brimstone

sediments and condensation of water vapor a very corrosive and deteriorative environment is created. By increment of fluid velocity in tube sediment creation will be decreased. Fluid swirling in tube causes a reduction of sediments and prompts self-clearance.

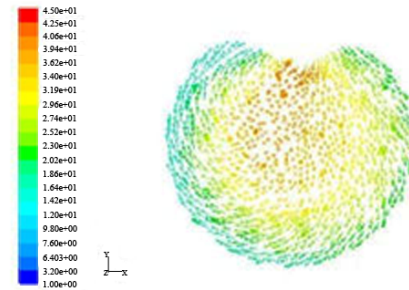


Fig. 8 (a) Velocity counters in one way spiral tube

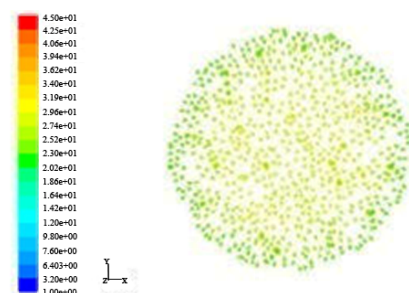


Fig. 9 (b) Velocity counters in plain tube

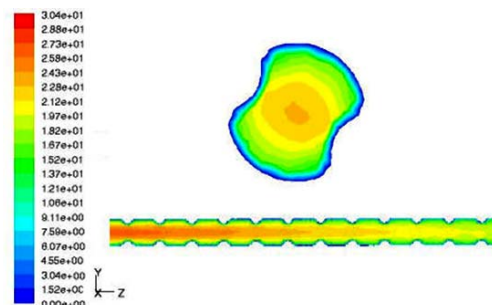


Fig. 10 (a) Velocity contours in 2 way spiral tube

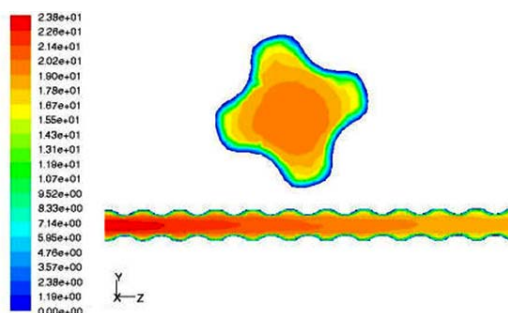


Fig. 11 (b) Velocity contours in 4 way spiral tube

By investigating increment of number, pitch and height of spirals for clear and fouled tubes optimum design condition



has been selected to reach the best situation regard to efficiency despite of pressure drop negative effects. This condition is shown in Fig. 12.

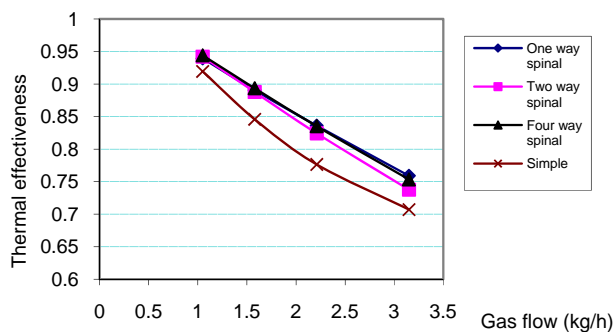


Fig. 12 Spiral number effect on thermal effectiveness

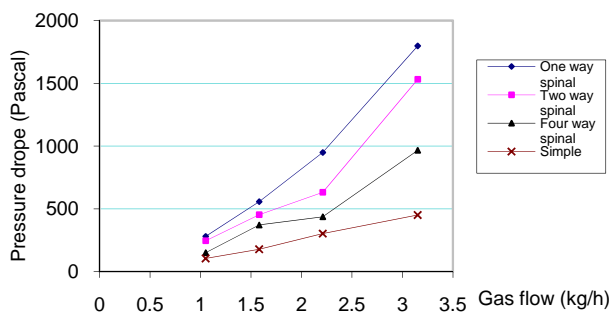


Fig. 13 Spiral number effect on pressure drop

#### IV. EFFECT OF TUBE HEAD DIAMETER TO SHELL DIAMETER ON FLOW

Gas flow equal dispensation in tubes is one of the important subjects in designing of EGR cooler entrance head. Fig. 13 exhibits circular arrangement for 19 tubes of 3/8 inch in a shell with 63 mm diameter. Tubes are numbered from 1 to 19 to facilitate tubes flow percentages denotation in Fig. 14.

At low proportions of entrance diameter to shell diameter ( $D1/D2$ ) flow percent decreases in side tubes and as a result of this more flow is passed into central ones. For example at  $D1/D2=0.4$  flow velocity in central tubes is 35% more than side tubes; this causes pressure reduction in central tubes and increment of cooling water temperature near them, it may cause local boiling at that section which causes cooler vibration and thermal stresses in working time. Also decrement of flow rate and subsequently decrement of turbulence in side tubes causes a reduction of local heat transfer rate. Fluid velocity reduction can prompt sediment increment too. All of these cause decrement of effectiveness and life time of cooler. Fig. 15 shows influence of internal diffuser diameter to shell diameter ratios of 40% and 70%. Temperature increment in central tubes, as a result of high flow rate of gas and consequently temperature raise of cooling fluid near them, is apparently visible.

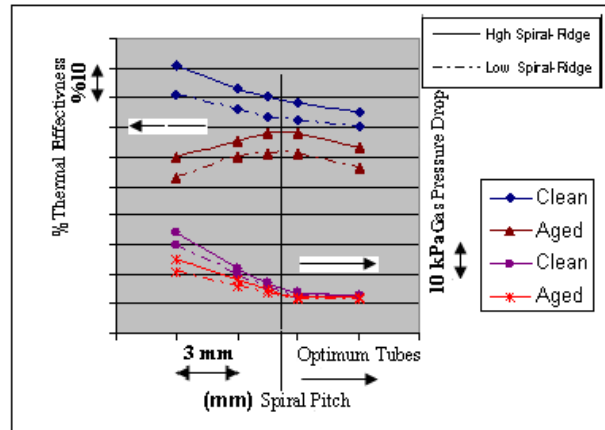


Fig. 14 Influence of pitch and height of spiral on effectiveness and pressure drop for clean and fouled tubes

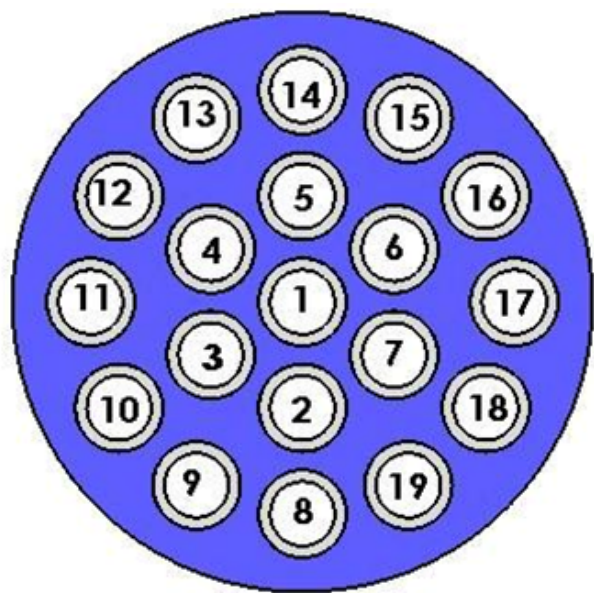


Fig. 15 Circular arrangement with denoting tubes' numbers in scope of flow calculation

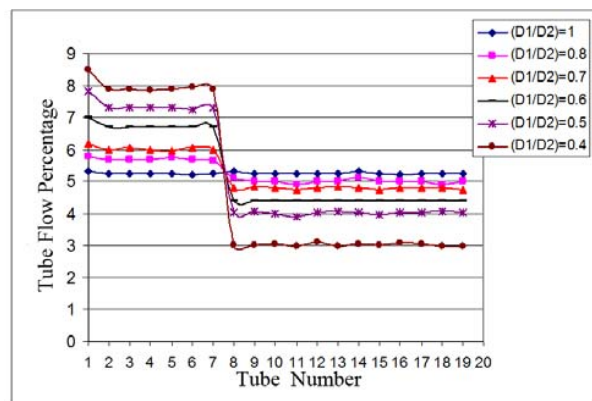


Fig. 16 Influence of entrance head diameter to shell diameter on the tubes' flow

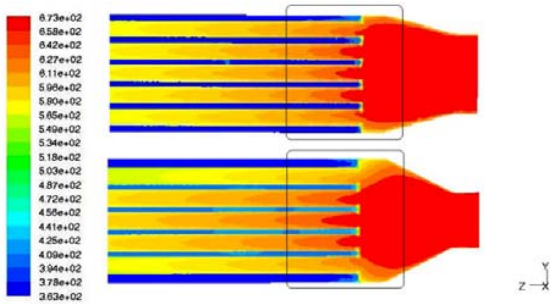


Fig. 17 Influence of diffuser entrance diameter to shell diameter ratio (40% & 70%)

On the other side if  $D_1/D_2$  is less than afore, we will need more length of diffuser to distribute flow appropriately and equally. This will induce larger cooler length and more pressure drop in gas side. Based on fittings' type and EGR tube connectors, regard to acquired results, suggested ratios of  $D_1/D_2$  are 60% to 75%. Of course amounts of  $D_1$  parameter are restricted regard to tubes' diameter.

#### V. COOLER DESIGN FOR A SAMPLE DIESEL ENGINE

Cooler design for working conditions of diesel engine of 4-248 M.F.Tractors, after performance and pollution tests and calculation of maximum quantity of EGR, has been done and results are given in Table I; tests reveal a significant increment of grime in EGR growth more than 20 percent. With regards to gasoil properties, high percentage of brimstone in it and based on numerous tests, optimum EGR quantity for this engine is 10%; but design has been done in a case which works appropriately up to 15%.

High pressure EGR system regard to explained benefits of it is installed on this engine. Exhausted EGR gases in this system have a temperature about 500°C, this is because of their extraction for cooling operations before turbocharger. With respect to the tests, temperature of exhausted gases after turbocharger is 70°C to 100°C less than normal. This has been supposed that thermo-dynamical properties of EGR gases are same as exhausted engine gases mean specifications which are combustion productions of sample gasoil with air. The used fuel (gasoil) chemical formulation is  $C_{12}H_{26}$ .

Cooling temperature range is very important. Some hypothetical and experimental works have been done in scope of considering effects of brimstone particles on the EGR cooling system. Researches exhibit a dependence of various brimstone compositions condensation in every temperature on chemical synthesizes of exhausted gases, but dew point of exhausted gases sulfuric acid with the fuel contains 370ppm sulfur is about 130°C [13]. More decrement of temperature causes sulfur and water vapor condensation at backflow gases and consequently sulfuric acid formation causes erosion in cooler tubes, valves, EGR system connectors and combustion chamber. Also high temperatures of exhausted gases impediment appropriate reduction of  $NO_x$  and grime. With regard to tests, proper temperature of cooler outlet is about 170°C to 210°C.

TABLE I  
DESIGN RESULTS FOR NORMAL AND OPTIMAL COOLER

	Optimum Cooler	Normal Cooler
Input gas flow (kg/s)	0.0153	0.015
Input and output gas temperature (K)	473,723	473,723
Tube internal diameter (mm)	7.2	7
Tube external diameter (mm)	9.5	9.5
Number of tubes	19	19
Tube length (mm)	284	410
Pitch proportion	1.25	1.25
Tube pressure drop (Pascal)	316	185
Water flow (kg/s)	0.1809	0.1894
Input and output water temperature (°C)	90-85	90-85
Shell internal diameter	63	63

#### VI. CONCLUSION

Acquired results for optimum cooler exhibit weight and dimension decrement in comparison with typical coolers; it is due to high surface density, effectiveness increment and also, equal flow distribution in tubes. Negative effects of pressure drop are decreased significantly and cooler has a high perpetuity against thermal stresses.

One way spiral tube usage increases thermal effectiveness of cooler up to 10 percent more than typical coolers and using multi way spiral tubes causes pressure drop reduction, this is because of semi-uniformity of flow in tubes; moreover, as a result of heat transfer area increment, it doesn't have high negative influence on thermal efficiency. Also increment of fluid spinning velocity helps sediment reduction, self-cleaning and heat transfer increment.

Pitch size reduction and spiral depth increment are effective factors at heat transfer and pressure drop growth.

Simulations with regard to pitch size, height, number of spirals and sediment effects have been resulted in ideal and optimal tube design with balance between heat transfer effectiveness and pressure drop in working range of EGR cooler. Entrance diffuser and inlet diameter to shell diameter ratio in scope of uniform flow distribution, thermal effectiveness increment and pressure drop reduction have been investigated. Standard diffuser shape with 60% to 70% diameter ratios gives the best result.

#### NOMENCLATURE

$A_{to}$	: Total heat transfer area ( $m^2$ )
$A_o$	: External heat transfer area of tube ( $m^2$ )
$A_i$	: Internal heat transfer area of tube ( $m^2$ )
$A_w$	: Logarithmic mean heat transfer area ( $m^2$ )
$C_{ps}$	: Specific heat of shell fluid (cooler fluid) (J/kg.K)
$C_p$	: Specific heat of gas (J/kg.K)
$D_e$	: Equivalent hydraulic diameter (m)
$D_i$	: Tube internal diameter (m)
$D_o$	: Tube external diameter (m)
$D_s$	: Shell internal diameter (m)
$F$	: Correction factor of LMTD at multi pass flows
$h_{id}$	: Heat transfer coefficient of shell at ideal flow model ( $W/m^2.K$ )
$h_o$	: Heat transfer coefficient at shell side ( $W/m^2.K$ )
$h_i$	: Heat transfer coefficient at tube side ( $W/m^2.K$ )
$J_I$	: Correction factor for effects of wall seals

- $J_b$  : Correction factor for cross flows  
 $J_s$  : Correction factor for walls distance at inlet and output sections  
 $J_r$  : Correction factor for Reynolds number less than 100  
 $J_c$  : Correction factor for fluid shear effect near the wall  
 $k_t$  : Thermal conductivity of tube side fluid (W/m.K)  
 $k_w$  : Thermal conductivity of wall (W/m.K)  
 $N_t$  : Number of tubes  
 $NTU$  : Number of heat transfer unit  
 $\dot{m}_s$  : Mass flow rate of shell side fluid  
 $Q$  : Actual heat transfer rate (W)  
 $U_o$  : Overall heat transfer coefficient (W/m<sup>2</sup>.K)  
 $R_i$  : Tube internal fouling resistance (m<sup>2</sup>.K/W)  
 $R_o$  : Tube external fouling resistance (m<sup>2</sup>.K/W)  
 $t_w$  : Thickness of tube wall (m)  
 $\mu_s$  : Cooling fluid viscosity (water) (N.s/m<sup>2</sup>)  
 $\mu_t$  : Hot fluid viscosity (gas) (N.s/m<sup>2</sup>)  
 $\varepsilon$  : Exchanger effectiveness  
 $\phi$  : Temperature dependent viscosity correction factor

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