 Investigation on the Fouling and Heat Transfer Characteristics of Novel EGR Cooler (Semi-spiral) for Diesel Engine Fueled with Biodiesel

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Abstract. In this study, heat transfer characteristic and fouling of Novel EGR cooler (Semi-spiral) system were investigated. In this study, waste vegetable oil (Waste Cooking Oil) was used as feedstock for biodiesel production. The experiment was conducted in different loads (25%, 50% and 75%) and different speeds (2100rpm and 2400rpm). Four fuel blends were prepared and used (B0, B5, B10, and B15). The new EGR cooler has 6 tubes in length of 19 cm and overall heat transfer coefficient of 35.63 W/m² K which is acceptable with compared to overall heat transfer coefficient of shell and tube heat exchanger. Also, the total resistance of fouling (Rft) derived was 0.002712 m² K/W. The maximum heat transfer efficiency is 94% and the minimum heat transfer efficiency is 68%. Also, the results showed that when B0 is used, the deposition was 1.37 gr that amounts to 0.23gr/hr. When B15 is used, the deposition was 1.21gr that amounts to 0.20gr/hr.

Keywords: heat transfer, diesel engine, vegetable oil, heat exchanger, pressure drop

1. Introduction

Diesel engines have good potential to reduce greenhouse gases. However, some of the pollutants of diesel engines play an important role in polluting the environment [1]. For example, one of the important and significant emissions of diesel engines is NOx emissions. NOx emissions easily pollute the environment and are a carcinogenic pollutant [2, 3]. NO and NO₂, which are called NOx emissions, combined with hemoglobin in blood and moisture content in liver and creates nitric acid. Due to low concentrations of acid produced with NOx, their effects will be less. But through time they accumulate in the body and cause respiratory diseases. The cause of NOx in diesel engines is the high-temperature combustion [4]. Many strategies have been used to reduce NOx emissions in diesel engines including the use of after-treats like technology and combustion management [5-7]. Another way to reduce NOx is through exhaust gas Recirculation (EGR). EGR is the Recirculating amount of exhaust gas to inlet [8]. Exhaust gas Recirculation causes dilution of oxygen at the inlet, increasing thermal capacity and thus reducing the maximum temperature of combustion. Therefore, by using EGR, NOx emissions can be significantly reduced. EGR has a high temperature; hence there are no significant impacts on reducing the maximum temperature combustion. Therefore, to enhance the performance of the EGR, cooled EGR is used. In which case the heat capacity can be significantly increased [9, 11]. To cool the EGR, heat exchangers (EGR cooler) is used. Until now, various heat exchangers and EGR cooler has been evaluated by researchers. Jang et al. [10] examined two types of shell- tube EGR cooler and a stack EGR cooler. In this study, heat transfer and pressure drop in three types of EGR coolers were experimentally investigated. The results indicated that temperature...
of EGR significantly decreases in the stack type; therefore when stack type EGR cooler is used, the amount of NOx is reduced. Liu et al. [12] investigated the characteristics of flow and heat transfer in EGR cooler with Helical Baffled Cooler with Spiral Corrugated Tubes (EGR-HBCSCT). The results indicated that the tube side heat transfer in the EGR-HBCSCT is 60 to 130 percent more than the tube side heat transfer of EGR-SBCST as shell side resistance in the EGR-HBCSCT is about 2.8 to 5.9 percent less than the shell side resistance of EGR-SBCST. The results showed that the pressure drop in the EGR-HBCSCT was less than EGR-SBCST. Liu et al. investigated the turbulent flow and heat transfer characteristics in the 7 types of cooling pipes of Internal Longitudinal Plate-Rectangle Fins EGR Cooler. The results showed that the friction factor increases with the number of fins, and the friction factor decreases with increase in internal diameter and width of the fin. Liu et al. [13] investigated the flow and heat transfer performances of an EGR cooler with internal longitudinal fins. This study was conducted experimentally and through three-dimension computation. In this study, the effects of fin width and fine height on heat transfer and flow were investigated. The results showed that in both internally finned tubes the heat transfer significantly increases. It was also revealed that the heat transfer in the pipes with blocked structure is better while the pressure drop in the tube without blocked structure is lower. Shabgard et al. [14] investigated the effects of several types of nano-fluid as a coolant on the performance of heat transfer and fluid flow characteristics of EGR cooler. The results showed that using nano-fluid as coolant, the heat transfer increases significantly. Fouling (Deposit) of EGR cooler is of great importance. Deposit formation can be defined as the accumulation of undesirable substances on a surface. In general, Deposit formation refers to the accumulation of unwanted substances that lead to undesired operations on the surface. During the operation of the EGR cooler, the heat transfer surface is deposited. This leads to increase in thermal resistance and pressure drop [15]. Park et al. [16] investigated the effects of deposited PM on the heat transfer in the Fin-Type EGR Cooler. In this study, six different EGR coolers with various pitches and grooves were used. The results showed that the lowest temperature and lowest pressure drop of exhaust gas occurred in the P3.6/36 EGR cooler. Abd-Elhady et al. [17] investigated the effects of velocity gas passing through the EGR cooler on the deposition of PM. In this experiment, the shell–tube EGR cooler was used. The flow rate through the EGR cooler was considered at three levels: 30, 70 and 120 m/s. The results showed that when the flow speed is 120 m/s, the least amount of soot is formed because fewer deposits were formed in the EGR cooler. Hong et al. [18] investigated the size and SOF soot particles deposited in the one type EGR cooler. Effect of EGR at three temperatures of 150, 250 and 350 °C were investigated. The coolant temperature was 80 °C. The results showed that as a whole, by increasing the temperature of EGR, the size of particulate matter (PM) decreased but the amount of deposited formed in the 10 hours of testing increased.

With regards to the review of the above results, it was found that the geometry of the EGR cooler has a significant impact on the flow and heat transfer performance of EGR cooler. It was also revealed that the deposit of EGR cooler pipes has a significant impact on performance of EGR cooler. So far, little research has been done on the effects of the pipe deposition of EGR cooler performance. In this study, a new EGR cooler was designed and built. After fabrication, the EGR cooler was mounted on a single-cylinder diesel engine. Then the performance parameters such as heat transfer efficiency, Nusselt number, and overall heat transfer coefficient and pressure drop along the way were evaluated. Also, the fouling of EGR cooler tubes for different working times and the effects of EGR cooler fouling on heat transfer tubes were investigated.

2. Material and method

2.1. Geometry of EGR cooler

Using the design criteria stated in the previous parts, the EGR cooler was designed and built. Semi spiral-semi smooth tube n was applied to Recirculation of the exhaust gas. The built EGR cooler is shown in Figure 1. The smooth semi spiral-semi smooth tubes used in the EGR cooler is shown in Figure 2.
2.2. Engine test and experimental procedure

2.2.1. Experimental set-up

The schematic of experimental set-up is shown in Figure 3. In this study, air-cooled single-cylinder engine (Lombardini DIESEL 3LD 510) was used and the details of its engine is shown in Table 1.

Table 1. Specifications for the test engine

| Description                  | Specification          |
|------------------------------|------------------------|
| Engine type                  | Lombardini- Diesel 3LD510 |
| Cylinder number              | 1                      |
| Displacement(cm3)            | 510                    |
| Induction type               | Non- turbocharger       |
| Bore * stoke (mm)            | 85*90                  |
| Maximum torque (Nm-RPM)      | 32.8-1800              |
| Compression ratio            | 18                     |

Figure 1. Photograph of EGR cooler

Figure 2. Semi spiral-semi smooth tubes

Figure 3. Engine test set-up with EGR cooler.
The single-cylinder engine used in this study was coupled with the dynamometer (Eddy Current Type DC Dynamometer). In order to measure temperature, 5 Thermocouples (K type) were used. The thermocouple measurement range was -200 to 1200 °C. In order to monitor the temperature measured by the thermocouple, TM 925 thermometer was used.

2.2.2 Fuel properties
In this study, waste vegetable oil (Waste Cooking Oil) was used as feedstock for biodiesel production. After producing biodiesel with the trans-esterification method, some important properties such as kinematic viscosity, flash point, cloud point, pour point, the amount of water and sediment, the amount of free glycerine, and corrosion were measured and the results were compared with the international standard (ASTM D-6751). In Table 2, the properties of biodiesel with the relevant standard acceptable ranges were presented.

| Properties               | Standard     | Exposure Limits | Biodiesel | Unit |
|--------------------------|--------------|-----------------|-----------|------|
| flash point              | ASTM D-92    | ≤130            | 176       | °C   |
| kinematic viscosity      | ASTM D-445   | 1.9-6           | 4.73      | mm²/s|
| Cloud point              | ASTM D-2500  | -               | -1        | °C   |
| pour point               | ASTM D-97    | -               | -4        | °C   |
| corrosion                | ASTM D-130   | 3 ≤             | 1a        | -    |
| free glycerin            | ASTM D-6584  | 0.02≤           | 0.016     | % mass |
| water and sediment       | ASTM D-2709  | 0.05≤           | 0.05      | % Vol|
| Density                  | -            | -               | 0.88      | g/cm³|

2.2.3. Experimental procedure
Given that the best performance of EGR occurs in the part load, so the EGR Cooler Performance Test were done in 25%, 50% and 75% of load. Experiment was conducted in 2100 rpm and 2400 rpm. Four fuel blends were prepared and used. These blends were, B5, B10, and B15 in each of the speeds and blends tested at three levels of EGR (10%, 20% and 30%). A matrix experiment is shown in Table 3.

| Variable                  | Levels         |
|---------------------------|----------------|
| EGR rate (%)              | 10, 20, 30     |
| Load (%)                  | 25, 50, 75     |
| Fuel blends               | B0, B5, B10, B15 |
| Engine speed (rpm)        | 2100, 2400     |
| Coolant inlet temperature (°C) | 85             |

2.2.4 Fouling test
In order to measure the weight of sediment accumulated in the EGR cooler tubes, weighing scales with 0.01gr of precision were used.

3. Result and discussion

3.1 Inlet temperature of EGR cooler
Inlet gas temperatures of EGR cooler at different speeds and loads to different EGR rates for the blends (B0, B5, B10, and B15) is shown in Figures 4-9. According to Figures 4(a-f), the increase of the EGR rate causes the EGR cooler inlet temperature to also increase. On average, at 2100 rpm and 25% load, with 20% increase of EGR, the inlet temperature of EGR cooler is increased by 14.75%.
The increase of the inlet temperature of EGR cooler also occurred for four blends with similar trends. At 2100 rpm and 50% load, with 20% increases of EGR on average, the inlet temperature of EGR cooler increased by 20%. Also considering Figures 4(a-f), when using biodiesel fuel blends, the inlet temperature of EGR cooler is enhanced. For example, at 2100 rpm and 50% load, when using a B10 blend, the inlet temperature of EGR cooler showed 8-14% increase compared to when the B0 blend was used. According to Figure 4-9, it is clear that the highest inlet temperature of EGR cooler is obtained in the following conditions: 2400 rpm, 75% load and B15 blend.

**Figure 4.** (a) Inlet temperature of EGR cooler (25% of load and 2100 rpm of speed), (b) Inlet temperature of EGR cooler (50% of load and 2100 rpm of speed), (c) Inlet temperature of EGR cooler (75% of load and 2100 rpm), (d) Inlet temperature of EGR cooler (25% of load and 2400 rpm of speed), (e) Inlet temperature of EGR cooler (25% of load and 2100 rpm of speed), (f) Inlet temperature of EGR cooler (25% of load and 2100 rpm)
3.2 Temperature difference of EGR cooler
In Figure 5a, and 5b, temperature difference of EGR coolers between inlet and outlet of tube side for two different engine speeds is shown at 25% load. According to these data, with 2100 rpm and 25% load, by increasing the EGR rate, the temperature difference of EGR cooler between inlet and outlet of tube side increased. With the increase of EGR rate to 20%, the temperature difference between inlet and outlet of tube side increased by 66.4%. Also, with 2400 rpm and 25% load, by increasing the EGR rate, the temperature difference between inlet and outlet of tube side of EGR cooler increased. According to results, it is clear that when B0 fuel was used for all EGR rates, the temperature difference of EGR cooler between inlet and outlet of tube side is greater than when B5, B10 and B15 blends were used.

![Figure 5. Temperature changes between input and output: (a) 25% of load and 2100 rpm, (b) Temperature changes between input and output (25% of load and 2400 rpm).](image)

3.3. Heat transfer efficiency of EGR cooler
The heat transfer efficiency of EGR cooler at different speeds and loads to different EGR rates for the fuel blends (B0, B5, B10, B15) are shown in Figures 6 (a-f). It is clear that by increasing the EGR rate, the heat transfer efficiency at 2100 rpm and 25% load did not decrease much. The heat transfer efficiency of EGR cooler with B0 blend compared to other blends (B5, B10 and B15) is more. The heat transfer efficiency of EGR cooler at 2100 rpm and 50% load, where the heat transfer efficiency of EGR cooler with B0 blend compared to other blends (B5, B10 and B15) is more. At 2100 rpm and 75% load, with increase of EGR rate, the heat transfer efficiency of EGR cooler is decreased by 5.75%.

![Figure 6. Heat transfer efficiency of EGR cooler: (a) 50% of load and 2100 rpm, (b) 75% of load and 2100 rpm.](image)
Figure 6. (a) Heat transfer efficiency: 25% of load and 2100 rpm of speed, (b) Heat transfer efficiency: 50% of load and 2100 rpm of speed, (c) Heat transfer efficiency: 75% of load and 2100 rpm, (d) Heat transfer efficiency: 25% of load and 2400 rpm, (e) Heat transfer efficiency: 50% of load and 2400 rpm, (f) Heat transfer efficiency: 75% of load and 2400 rpm.

Heat transfer efficiency of EGR cooler at 2400 rpm and 25% load is shown in Figure 15. A 20% increase of EGR rate decreases the heat transfer efficiency of EGR cooler by 14%. Heat transfer efficiency of EGR cooler at 2400 rpm and 50% load is shown in Figure 16. A 20% increase of EGR rate decreases the heat transfer efficiency of EGR cooler by 10.5%. Heat transfer efficiency of EGR cooler at 2400 rpm and 75% load is shown in Figure 17. A 20% increase of EGR rate decreases the heat transfer efficiency of EGR cooler by 9%.

3.4 Fouling of EGR cooler

To measure the sediment inside the EGR cooler pipes, the standards shown in the table were used. To measure the deposition, two blends (B0, B15) were used. When B0 is used, the deposition was 1.37gr that amounts to 0.23gr/hr. When B15 is used, the deposition was 1.21gr that amounts to 0.20gr/hr. The results show that in B15, the Fouling decreases by 13% when compared to B0. Cross section views of EGR cooler (when B0 is used) after deposition is shown in Figure 7.

Figure 7. Cross section views of EGR cooler (when B0 is used) after deposition.
The causes of sedimentation when biodiesel blend is used are as follows:
- Gasoline often contains 20 to 40 volume percent aromatics which increased emissions such as soot and particulate matter whereas biodiesel is substantially devoid of aromatics.
- Biodiesel is free from sulphur whereas in pure diesel fuels, sulphur is present so the products of the pure diesel combustion are sulphur dioxide and sulphur trioxide.

4. Conclusion
Conclusions of this study are summarized as follows:
- The overall heat transfer coefficient of 35.63 W/m²°K is acceptable compared to overall heat transfer coefficient of shell and tubes heat exchanger because the desirable and standard quantity of heat transfer coefficient for these kinds of coolers which use both water and gas is between 10 and 200 W/m²°K.
- The maximum thermal efficiency obtained was 94 percent and the minimum thermal efficiency obtained was 68 percent. The average of efficiency in speeds was obtained at 2100 and 2400 rpm as 77.2 percent, which is considerable in comparison with other similar studies.
- When B0 is used, the deposition was 1.37gr that amounts to 0.23gr/hr whereas when B15 is used, the deposition was 1.21gr that amounts to 0.20gr/hr. As a result, the use of fuels containing biodiesel in terms of fouling is more appropriate compared to diesel fuel.

References
[1] Shao J, Tao Y and Hansen K K 2016 Electrochemistry Communications 72 36-40
[2] Zheng M, Reader G T and Hawley J G 2004. Energy Conversion and Management 45 883-900
[3] Geng P, Tan Q, Zhang C, Wei L, He X, Cao E and Jiang K 2016 Science of The Total Environment 572 467-475
[4] Dich A, Grimslay K, Koufos D, Sarasin T and Rodas M 2012 Alternative and renewable energy
[5] Gunnarsson, F, Pihl J A, Troops T J, Skoglundh M and Härelind H 2017 Applied Catalysis B: Environmental 202 42-50.
[6] Wang J, Zhao H, Haller G and Li Y 2017 Applied Catalysis B: Environmental 202 346-354.
[7] Feng X, Cao Y, Lan L, Lin C, Li Y, Xu H, Gong M and Chen Y 2016 Chemical Engineering Journal 302 697-706
[8] Liu L, Li Z, Liu S and Shen B 2017 Mechanical Systems and Signal Processing 87 195-213
[9] Abu-Hamdeh N H 2003 Energy Conversion and Management 44 3113-3124
[10] Jang S H, Hwang S J, Park S K, Choi K S and Kim H M 2011 Heat and Mass Transfer 48 1081-1087
[11] Liu L, Ling X and Peng H 2013 Experimental Thermal and Fluid Science 44 275-284
[12] Liu L, Ling X and Peng H 2013 Applied Thermal Engineering 54 145-152
[13] Liu L, Ling X and Peng H 2015 Heat and Mass Transfer 51 1017-1027
[14] Shabgard H, Kheradmand S, Farzaneh H and Bae C 2017 Applied Thermal Engineering 110 244-252
[15] Kakac S and Liu H 2002 Heat exchangers - selection, rating and thermal design CRC Press
[16] Park S, Choi K, Kim H and Lee K 2010 Heat and Mass Transfer 46 1221-1227
[17] Abd-Elhady M S, Zornek T, Malayeri M R, Balestrino S, Szymkowicz P G and Müller-Steinhagen H. 2011 International Journal of Heat and Mass Transfer 54 838-846
[18] Hong K S, Lee K S, Song S, Chun K M, Chung D and Min S 2011 Atmospheric Environment 45 5677-5683
[19] Özçelik Y 2007 Applied Thermal Engineering 27 1849–1856
[20] Adrian B and Allan D K 2003 Heat transfer handbook John Willey and Sons
[21] Shah R K and Sekulic D P 2003 Fundamentals of heat exchanger design John Willey and Sons
[22] Patel V K and Rao R V 2010 *Applied Thermal Engineering* **30** 1417-1425
[23] Babu B V and Munawar S A 2007 *Chemical Engineering Science* **62** 3720-3739
[24] Naphon P, Nuchjapo M and Kurujareon J 2006 *Energy Conversion and Management* **47** 3031-3044
[25] Ponce-Ortega J M and Serna-González, M.; Jiménez-Gutiérrrez, A. 2009 *Applied Thermal Engineering* **29** 203-209.
[26] Bell K J 1980 *Conf. NATO Adv. Study Inst. Heat Exchangers: Thermal-Hydraulic Fund. & Design* 559-580
[27] Bell K, Kakac S, Bergles A and Mayinger F 1981 Delaware method for shell side design, heat exchanger thermal-hydraulic fundamentals and design *Taylor and Francis*
[28] Stachura V J 1988 *Standards of the tubular exchanger manufacturers association*. Tubular Exchanger Manufacturers Association
[29] Kakac S, Liu H and Pramuanjaroenkij A 2012 *Heat exchangers: Selection, rating, and thermal design* *CRC Press*
[30] Caputo A C, Pelagagge P M and Salini P 2008 *Applied Thermal Engineering* **28** 1151-1159
[31] Kakac S and Liu, H 2002 *Heat exchangers: Selection. Rating and Thermal Design* *CRC Press* 88-89