HEAT TRANSFER THROUGH THE OIL FILM AND ITS EFFECT ON CONNECTING ROD BEARING TEMPERATURE

A. A. Alarabi and A. A. Tetewi
a.alarabi@ceh.edu.ly areejmahdy@yahoo.com
College of Engineering Technology, Hoon - Libya

ABSTRACT

The present paper was focused on the heat transfer rate through the main bearing of single and four cylinder engines using fresh (new) and used oil at different rotational speed. The experimental study was carried at I.C. Engine laboratory College of Engineering Technology Hoon, Libya. The results show that for four-cylinder engine, and at a engine rpm. more than 2500 rpm, it was found that the heat absorbed by the oil increases approximately 10% compared with the used oil. The oil condition i.e. (used or new) had a very small affect on bearing temperature. Comparing the new oil with the used oil it was obtained that increasing the engine rpms by 1000 rpm increases the heat transferred by the new oil approximately 40%, while the overall heat transfer coefficient increases by approximately 25% and the bearing temperature increases by approximately 10%. It is concluded that the new oil plays a positive role in heat transfer process and in protecting the bearing from frictional wear.

Key worlds: Heat transfer, connecting rod bearing, oil film, oil temperature.

1. INTRODUCTION AND SURVEY

Most internal combustion engines are fluid cooled using either air (a gaseous fluid) or liquid coolant run through a heat exchanger (radiator) cooled by air. Marine engines and some stationary engines have ready access to a large volume of water at a suitable temperature. The water may be used directly to cool the engine, but often has sediment, which can clog coolant passages or chemicals, such as salt, that can chemically damage the engine. Thus, engine coolant may be run through a heat exchanger that is cooled by the body of water. However, properties of the coolant (water, oil, or air) also affect cooling. Oil cooling is the use of engine oil as a coolant, typically to remove surplus heat from an internal combustion engine. The hot engine transfers heat to the oil which then usually passes through a heat-exchanger, typically a type of radiator known as an oil cooler. The cooled oil flows back into the hot object to cool it continuously. Oil has about 90% the density of water, so a given volume of oil can absorb only about 50% of the energy of the same volume of water.

H. H. Priebsch and J. Krasser (1997), had carried a Simulation work concerning the oil film behaviour in elastic engine bearings considering pressure and temperature dependent oil viscosity. They reported that, the design analyses of crank train bearings of reciprocating engines are strongly influenced by the oil parameters considered in the simulation models.. To improve this purpose a comprehensive thermo-elastohydrodynamic (TEHD) calculation method was developed and implemented into a computer code. It is concluded that nearly two thirds of the frictional heat leave the bearing by convection (oil flow). [1]

Peng Wang1, et. al., (2017), employed the Eulerian multiphase model and the geometry reconstruction scheme to describe the periodic flow and heat transfer processes of engine oil inside the piston cooling gallery. They reported that the heat transfer coefficient increases with an increase in the engine speed and tend to present a similar distribution. In addition the optimal jet velocities of cooling oil under different engine speeds were suggested. The Eulerian multiphase model results were compared with the CLSVOF model, and it was found that the results from the Eulerian multiphase model are more close to the actual state. The error percentage of Eulerian multiphase model was 3.15% while in the CLSVOF model was 14.13% [2].

As it was reported by **J**.M. Conway-Jones and R. Gojon (1991) [3] reported that, the primary route for removal of heat from crankshaft bearings is by oil flow, which in turn depends upon oil supply pressure, oil inlet geometry and bearing clearance. The largest flow will be from a 360° central circumferential groove, and will be primarily controlled by the oil supply pressure and the cube of the bearing clearance. Calculations indicate that the oil flow through a connecting rod bearing may be only 10-20% of the flow through a main bearing with a 360° circumferential groove. This relationship has been calculated by the "oil film history" program NEWSOL **[4]** and allows for the varying gap between the hole and the bearing surface as the crankshaft rotates.

The program follows the history of the oil in the film, and calculates the flow from the sides of the bearing which is influenced by oil supply pressure, drag (rotational) and squeeze (radial) motion [5]. They concluded that the effective oil viscosity in automotive engine crankshaft bearings is controlled by heat flow through the crankshaft. At high speeds the increase in power loss over that estimated from viscous shearing may be attributed to asperity contact through a thin oil film.

Jingjie Cao, et.al. (2019**)**, had established an improved heat transfer model based on an enhanced thermal wall function using the piecewise functions in which only the laminar Prandtl number is considered in the viscous sublayer, and the competition between the laminar Prandtl number and the turbulent Prandtl number is taken into account in the buffer layer and turbulent core regions.

The results show that the characteristics of heat flux can be more suitably reproduced using the improved heat transfer model than Han and Reitz model and Rakopoulos et al. model were presented for comparison. The results indicate that the heat flux characteristics under different combustion modes can be more satisfactorily reproduced using the improved heat transfer model than Han and Reitz model and Rakopoulos et al. model by avoiding the flaws in the thermal wall function [6].

W. Grzesik et.al. (2006) provided a comprehensive characterization of part surface finish produced in dry turning of a hardened AISI 52100 bearing steel using mixed ceramic (MC) and PCBN tools. In this investigation, some important 2D and 3D surface roughness parameters, as well as profile and surface characteristics, such as the amplitude distribution functions, bearing area curves, surface topographies and contour maps obtained for the four surface types selected, were determined and analyzed. They concluded that, the modifications of surface profiles achieved by means of special abrasive machining operations can distinctly improve the bearing properties of previously hard turned surfaces, and exemplarily, they shorten the running-in period.

2. EXPERIMENTAL WORK

The experimental work was carried in internal combustion engines at College of Engineering – Hoon. Two types of Diesel engines were used four cylinder engine and single cylinder engine. Description of the engines shown in table 1. It can be noticed that both engines have the same dimensions. Two types of oils were used. Used oil means an oil operated more than 100 hours. New oil means fresh oil used for the first time. The heat is transferred from the combustion chamber upper the piston to the main bearing through the oil and the piston connecting. The main bearing surface temperature raising due to three reasons the heat transferred by conduction, heat transferred from the oil and due to the friction resulted from crank shaft rotation.

Table 1: Engines specification

Fuel	Crank shaft diameter	Bearing clearance	Bearing thickness	Big end connecting rod thickness	Bearing width	Piston stroke length
Petrol	63.4365 mm	0.0508 mm	3 mm	7 mm	30 mm	127 mm

Firstly, the engine temperatures was measured , then the temperatures of oil and main bearing were measured at intervals varied from one minute to one hour. Due to the movement of the connecting rod it was difficult to find a way to measure the temperature of the connecting rod bearing. Some assumptions was traken in order to find out the value of heat transfer rate. These assumption are:

- One dimension steady flow
- The properties of oil and air are obtained from the standard tables.
- The bearing alloy is Brass consist of (70% Cu. & 30% Zin.)
- The material of big end of connecting rod is steel.
- The air inside the crank case is stable. And its temperature is increased with the increase of engine operating time according the relation,

  $$T_{air} = T_o + K \, . \, t \qquad (1)$$

  Where:

  T_o – The initial air temperature (^oC).

  K – Constant (0.25 ^oC/min).

  T – time (min)
- The properties of the new oil differ from those of the used oil according to the following corrective factors:

  Thermal conductivity (k): $k_{new} = 1.05 (k_{used})$

  Viscosity: $\mu_{new} = 1.05 (\mu_{used})$

  Density: $\rho_{new} = 1.02 (\rho_{used})$

  Prandtl number: $Pr_{new} = Pr_{used}$

The correction factors were taken according to experiment carried in the laboratory so that both the properties of new and used oil were measured and then the average correction factor was obtained.

It is well known that the function of the oil is heat propagated from the walls of combustion chamber and the heat of bearing friction. Thus the oil should be release from the heat in order to sprayed with low temperature to combustion chamber walls. This means that oil should leave the bearing with a temperature as low as possible.

The heat absorbed by oil after certain time of operation can be calculated from the equation:

$$Q = \frac{m.c}{t} \Delta T \qquad (2)$$

- *m* - Mass of oil
- *C – Specific heat*
- $\Delta T – (T_f - T_i)$

  T_i – The initial oil temperature (^oC).

  T_f - The final oil temperature (^oC).
- t – Engine operating time (sec)

At the clearance between the crank shaft and the connected rod the oil will lose an amount of heat Q to the surrounded air in the crank case, through the bearing and the big end of connecting rod. The amount of heat can be calculated by:

$$Q = \frac{(T_{oil} - T_{air})}{R_{th}} \qquad (3)$$

Where:
T_{oil} – oil temperature
T_{air} – Air temperature
R_{th} – Thermal resistance

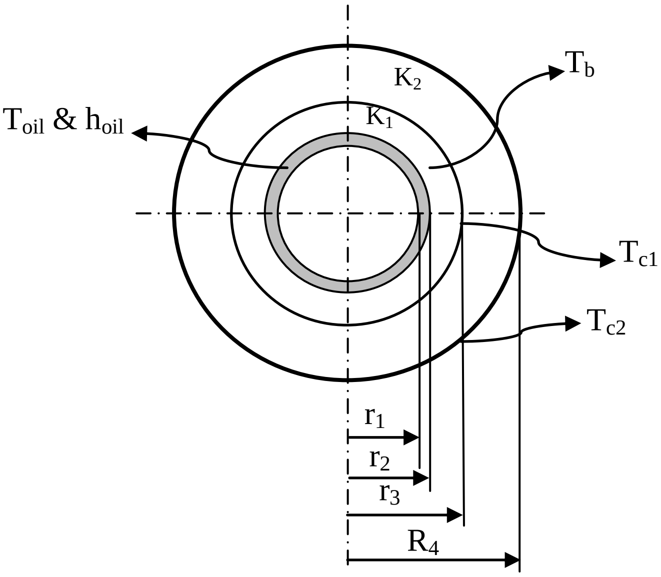

Fig. 1 Heat flow the big end of connecting rod

d_1- Crank shaft diameter
C_1- Oil lubricating film = r_2 - r_1
C_2- Connecting rod bearing thickness = r_3 – r_2
C_3 – Big end connecting rod thickness = r_4 – r_3
$R_{th} = R_1 + R_2 + R_3 + R_4$

$$R_{th} = \frac{1}{2\pi r_2 L h_o} + \frac{\ln\left(\frac{r_3}{r_2}\right)}{2\pi K_1 L} + \frac{\ln\left(\frac{r_4}{r_3}\right)}{2\pi K_2 L} + \frac{1}{2\pi r_4 L h_{air}} \qquad (4)$$

K_1 – Thermal conductivity of bearing material.
K_2 - Thermal conductivity of connecting rod material.
h_o - Heat transfer coefficient of oil.
h_{air} – Heat transfer coefficient of air.

Data from standard tables will be used to find out the relation of thermal conductivity of bearing material and connecting rod material with temperature
Table 1 shows the value of thermal conductivity of the bearing at 20°C and 100 °C

Table1: Thermal conductivity of bearing

Temperature (°C)	20	100
Thermal conductivity (w/m. °K)	111	128

Then.

$$K_2 = 106.75 + 0.2125\,(T) \qquad (5)$$

Table 2 shows the value of thermal conductivity of the connecting rod at different temperatures.

Table2: Thermal conductivity of connecting rod

Temperature (°C)	0	127
Thermal conductivity (w/m. °K)	83.5	69.5

Then,

$$K_2 = 83.5 + 0.111\,(T) \qquad (6)$$

2.1 Heat transfer coefficient of air

The big end of connecting rod rotating about the crank shaft with a constant rotating speed (N) with a radius (r), where:

$$r = \frac{S}{2}$$

S – Is the piston stroke (m)
K_{air} – is the thermal conductivity of air which varied with the temperature as:

$$\nu = 0.0237 + 69.53\times10^{-6}T + 1.67\times10^{-9}T^2 \qquad (11)$$

2.2 Heat transfer coefficient of oil:

Because there is no equation describing the oil condition to find the heat transfer coefficient in such a case where the oil spins in the space between the shaft and the bearing as shown in figure 2. Because there is no equation describing the oil condition to find the heat transfer coefficient in such a situation accurately, where the oil spins in the space between the column and the chair. Accordingly, a relatively similar equation was assumed for this case, since the cylinder or shaft rotates in a stationary fluid and therefore the cylinder can be considered constant and the fluid is rotating around it.

The relative speed of the connecting rod to the air which is denoted by V_2 is equal,

$$V_2 = \frac{2\pi N}{60} * \frac{S}{2} \qquad (7)$$

In small cylinders the Nusselt number can be obtained from the equation:

$$Nu = 0.123\,\mathrm{Re}^{0.651} + \left(\frac{D}{L}\right)^{0.85} \mathrm{Re}^{0.792} \qquad (8)$$

$$\mathrm{Re} = \frac{V_2 D}{\nu}$$

Re – Reynolds number
L – Cylinder length
D – External diameter of cylinder

From the standard tables where the values kinematic viscosity at the corresponding temperatures were recorded the relation of kinematic viscosity with temperature was found to be:

$$\nu = 13.918\times10^{-6} + 86.45\times10^{-9}T + 102.7\times10^{-12}T^2 \quad (9)$$

Therefore the heat transfer coefficient can be calculated from:

$$h_{air} = \frac{Nu \times K_{air}}{D} \qquad (10)$$

Then,

$$Nu = \frac{h \times D}{K} = 0.11(0.5\,\mathrm{Re}^2 + Gr\,\mathrm{Pr})^{0.35} \quad (12)$$

Where: $$\mathrm{Re} = \frac{\omega \times \pi \times D \times \rho}{\mu}$$

ω – The angular velocity
D – Bearing diameter.
Gr – Garashof number

Where:

$$Gr = \frac{g \times \beta \times}{\nu^2}(T_{oil} - T_b)D^3 \qquad (13)$$

T_{oil} – Oil temperature
T_b – Bearing temperature
$\frac{g \times \beta}{\nu^2}$ - Constant factor = 8475 1/m³.K

From oil properties table the variation of density, viscosity, Prandtl number (Pr) and thermal conductivity were established;

$$\rho = 889.12 - 0.5718\,T - 4.4563\times10^{-4}\,T^2 \qquad (14)$$

$$\mu = [e^{(\frac{5251722}{T+273} - 1.2406)}]/100 \qquad (15)$$

$$\Pr = e^{(5.6623 - 0.05092T)}/100 \qquad (16)$$

$$K = 0.147 - 1.116\times10^{-4}\,T - 4.385\times10^{-8}\,T^2 \qquad (17)$$

The properties of new oil can be obtained using the correction factors shown in the assumptions above. From equation 12, the convection heat transfer of oil can be obtained.

$$h_{oil} = \frac{Nu \times K_{air}}{D} \qquad (18)$$

By calculating the total thermal resistance from equation 4 and the heat transfer from equation 3, overall heat transfer coefficient based on to internal surface area will be:

$$Ui = \frac{Q}{A_i(T_{oil} - T_{air})} \qquad (19)$$

The overall heat transfer coefficient the based on to internal surface area will be:

$$Uo = \frac{Q}{A_O(T_{oil} - T_{air})} \qquad (20)$$

Then the temperature of inner surface of the bearing is,

$$Q = h_{oil} A_i (T_{oil} - T_b) \qquad (21)$$

Then,

$$T_b = T_{oil} - \frac{Q}{h_{oil} A_i} \qquad (22)$$

3. RESULTS AND DISCUSSION

Figure 1 shows the variation of heat transfer with time for four cylinder engine using used oil at different engine rotational speed (500 rpm – 3000 rpm). It can be seen that increasing the working time and engine rotational speed cause increasing the value of heat transfer, and the highest value was obtained at 3000 rpm. So increasing the engine rotational speed exposed the engine moving part to more frictional time, i.e. more heat will be gained. Variation of the connecting rod bearing with time at different rotational speed for four cylinder engine using used oil showed in figure 2. It can be noticed that the main bearing temperature increases with increasing the working time and engine rotational speed. This result corresponds to logic because the bearing surface exposed to more frictional time.

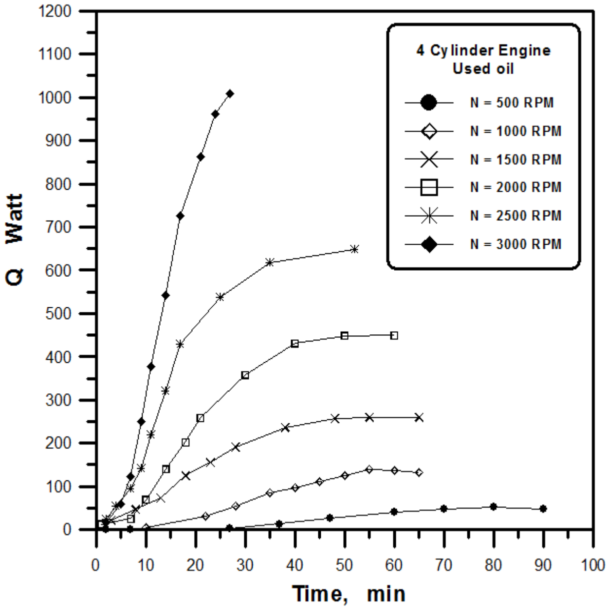

Fig. 1 variation of heat transfer with time for four cylinder engine using used oil at different engine rotational speed

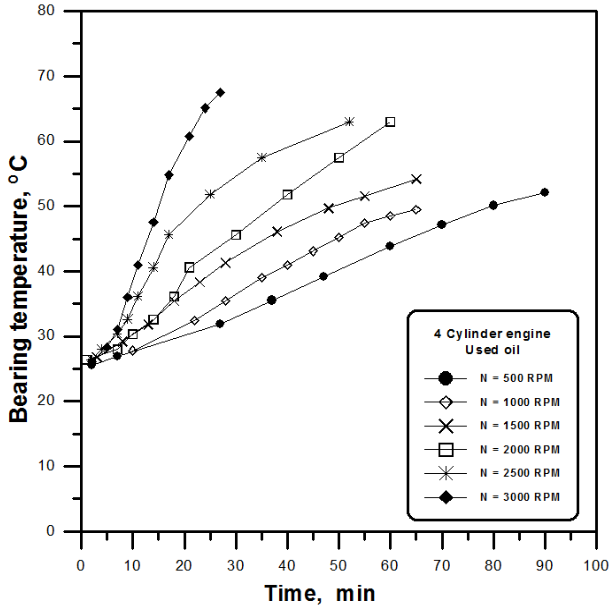

Fig. 2 Variation of the connecting rod bearing temperature with time at different rotational speed (4 - cylinder engine , used oil)

Figure 3 shows the variation of engine oil temperature (used oil) with time at varied rotational speed. It can be seen that increasing both working time and engine rotational speed increase the engine oil temperature. This result is based on the result shown in the previous figure because increasing the temperature of the main bearing means increasing the temperature of the oil in contact with it.

Figure 4 shows the variation of engine oil temperature (new oil) with time at varied rotational speed (1500 rpm and 2500 rpm). By comparison the effect of both used new oil i.e. figures 3 and 4 it can be seen that new oil absorbed more heat which increases its temperature about 5 ^{o}C or more at rotational speed 2500 rpm. while at 1500 rpm the results can be considered almost the same. Also the difference of oil temperature increases with increasing the working time.

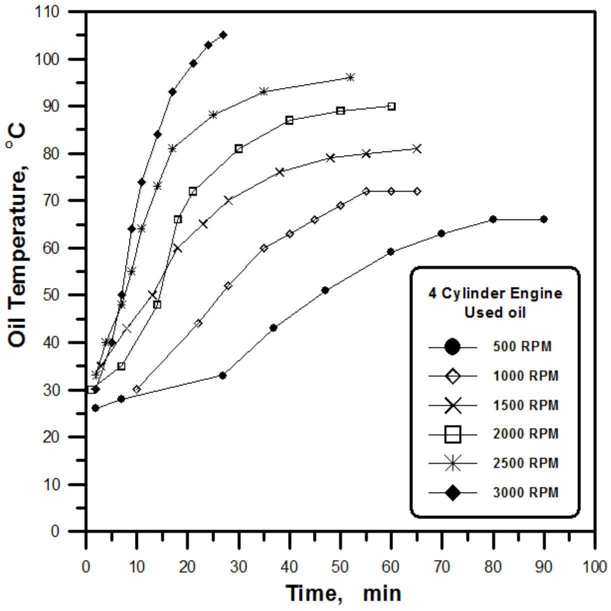

Fig. 3 shows the variation of engine oil temperature (used oil) with time at varied rotational speed

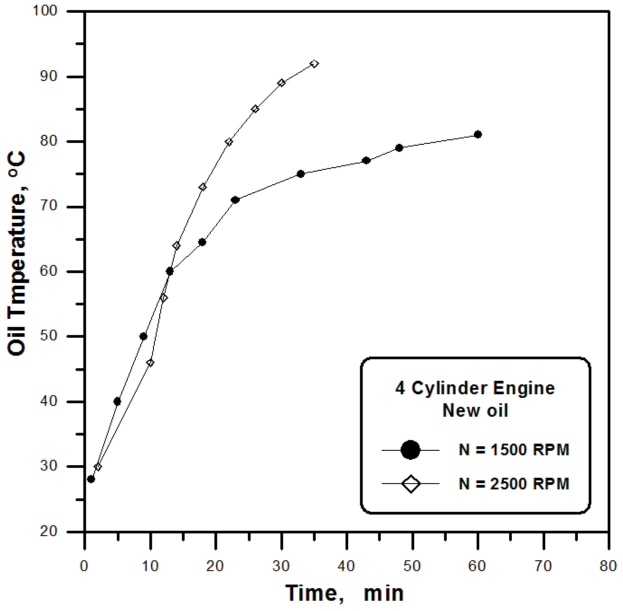

Fig. 4 Variation of engine oil temperature (new oil) with time at varied rotational speed

Figure 5 shows the variation of the connecting rod bearing temperature with time at different rotational speed (1500 rpm and 2500 rpm) for four cylinder engine using new oil. Comparing this result with result shown in figure 2, it can be noticed that the new oil achieved higher temperature than the used oil and main cause of this result is the properties of oil, where the properties of new oil has higher ability to resist friction and absorbing more heat and hence higher temperature was achieved.

Figure 6 shows the variation of heat transfer with time for four cylinder engine using new oil at different engine rotational speed (1500 rpm and 2500 rpm). It can be seen the heat transfer achieved value in the case of using new oil higher than in the case of used oil and this is mainly occurred due to the increase of both oil and bearing temperature. Also it can be seen that the difference between the 1500 rpm and 2500 rpm increases with the increase of working time.

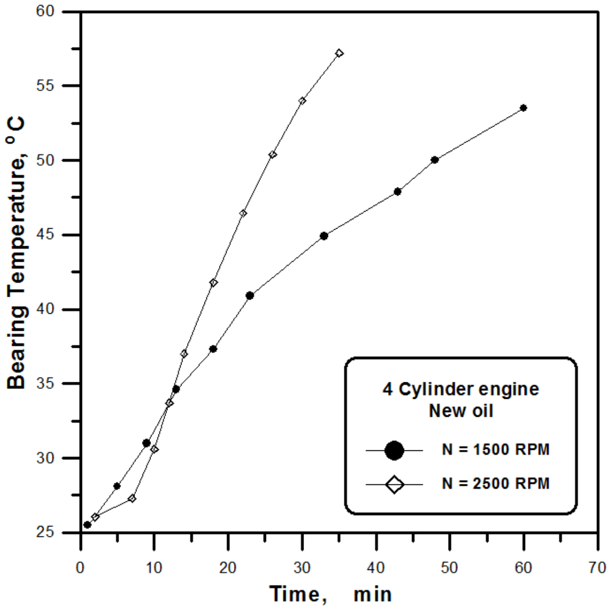

Fig. 5 Variation of the connecting rod bearing temperature with time at different rotational speed

Figure 7 shows the variation of overall heat transfer with time for four cylinder engine using new oil at different engine rotational speed (1500 rpm – 2500 rpm). It can be seen that a noticeable difference was obtained due to changing the engine rotational speed and also the working time. This result improves that the performance of the new oil is higher compared with the used oil.

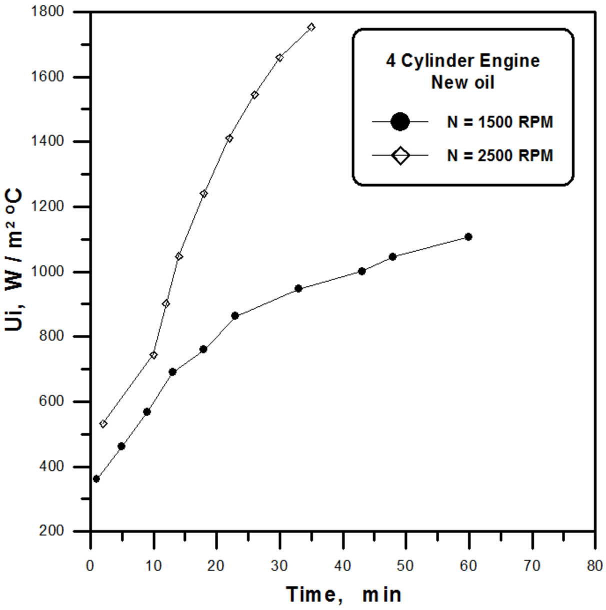

Fig. 7 Variation of overall heat transfer with time for four cylinder engine using new oil at different engine rotational speed

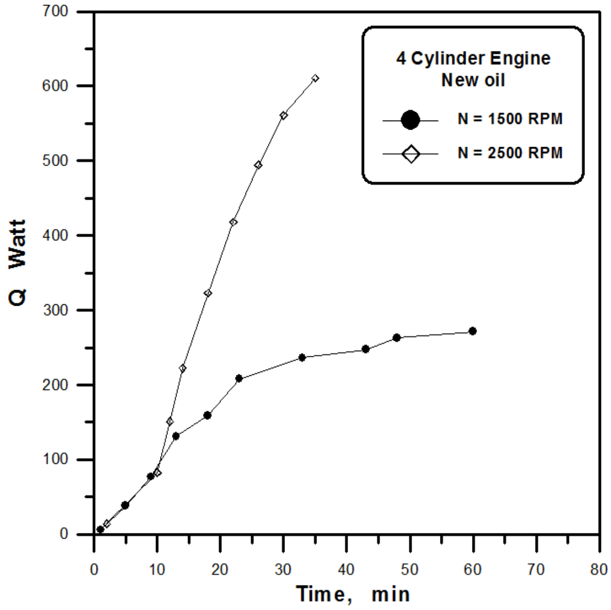

Fig. 6 variation of heat transfer with time for four cylinder engine at different engine rotational speed (new oil)

Figure 8 shows the variation of both used oil temperature and main bearing temperature with time for single cylinder engine with used oil. It can be seen that with increasing engine operating time the temperature difference increases and difference reaches about 15 °C. Increasing in oil temperature compared with bearing temperature means that the oil absorbed more heat which keeping the bearing more save from heating effect and friction.

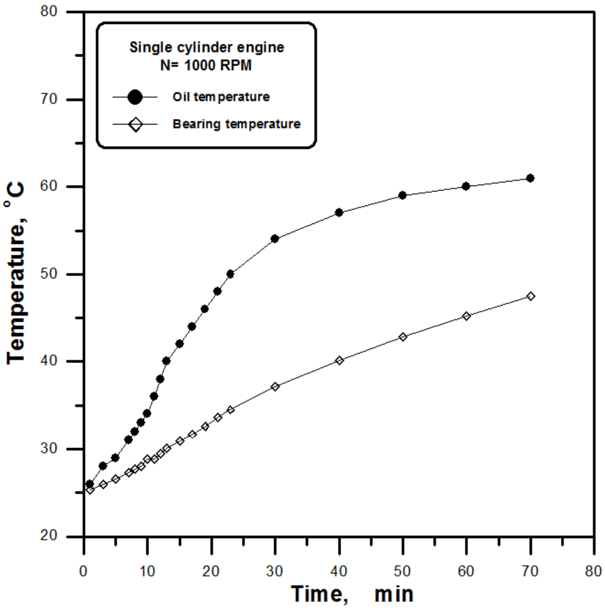

Fig. 8 Variation of used oil temperature and connecting rod bearing temperature with time single cylinder engine (used oil)

Figure 9 shows the comparison of heat transfer between single cylinder engine and four cylinder engine. The investigation was carried at 1000 rpm. Increasing the number of cylinders means increasing the engine power and hence more heat will be released. This result is shown in figure 9, where the difference reaches more than 100 W.

Figure 10 shows the comparison of engine oil temperature (used oil) between single cylinder engine and four cylinder engine. The investigation was carried at 1000 rpm. As a result of the increased capacity of the four-cylinder engine, this will increase the heat generated by the operation and thus increase the oil temperature. This fact is illustrated in Figure 10.

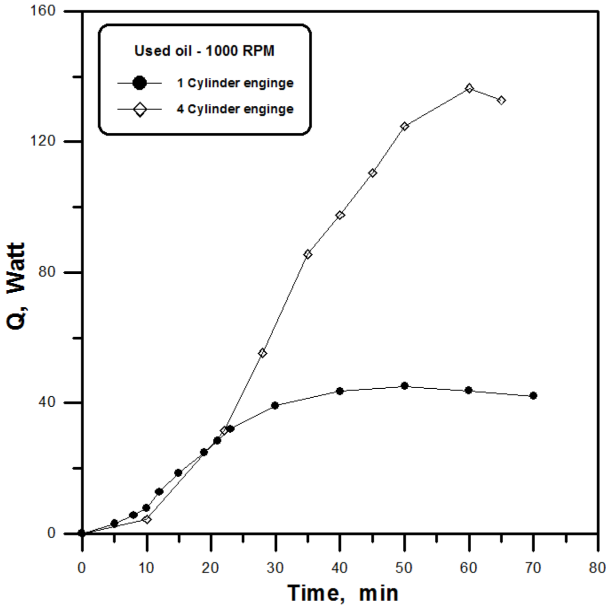

Fig. 9 Comparison of heat transfer between single cylinder engine and four cylinder engine

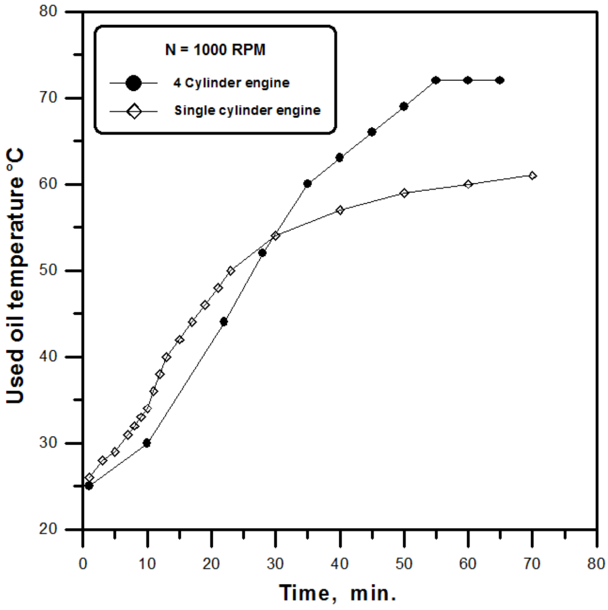

Fig. 10 Comparison of engine oil temperature between single cylinder engine and four cylinder engine (used oil)

Figure 11 shows the comparison of engine main bearing temperature between single cylinder engine and four cylinder engine. The investigation was carried at 1000 rpm. It can be seen that there is no significant difference in the bearing temperature due the proportionality between the amount of heat generated by the engine and the number of cylinders, regardless of the oil condition.

Figure 12 shows the comparison of oil temperature used and new oil with working time. The test was carried for four cylinder engine at 1500 RPM. Generally it can be seen that a small difference was obtained due to oil type, with relative preference for new oil.

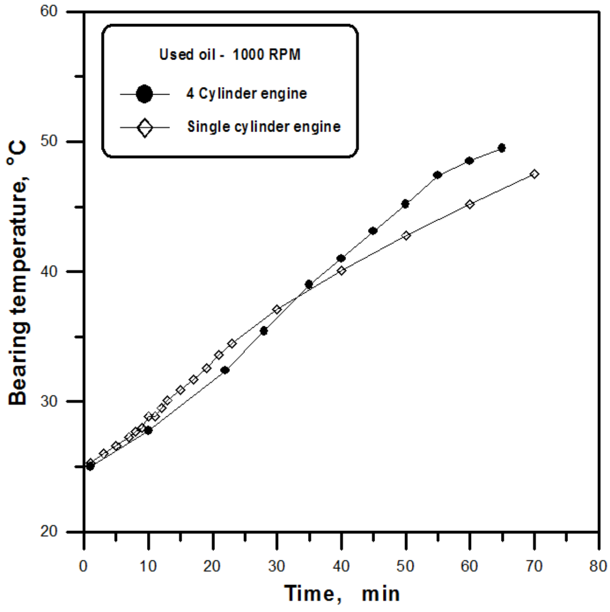

Fig. 11 Comparison of engine connecting rod bearing temperature between single cylinder engine and four cylinder engine

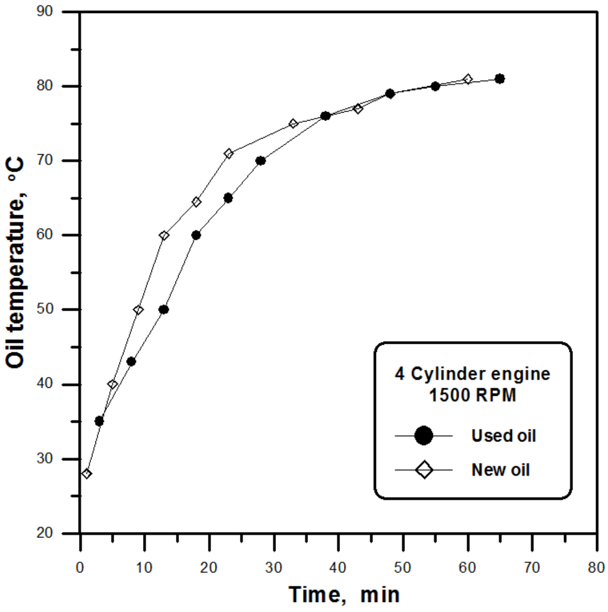

Fig. 12 Comparison of oil temperature (used and new oil) with working time at 1500 rpm.

Based on the results achieved in figure 12 the engine rotational speed was increased in order to achieve more information about the engine oil condition. At 2500 rpm a noticeable difference was obtained due to condition of the oil. This difference was achieved due the change of the oil properties see figure 13.

Figure 14 shows the variation of bearing temperature (used oil and new oil) of four cylinder engine at 2500 rpm. It can be noticed that there is no significant difference in the temperature of the bearing due to change of oil condition. This means that the resistance of bearing alloy playing the major effect in such condition, and the oil effect can be considered negligent

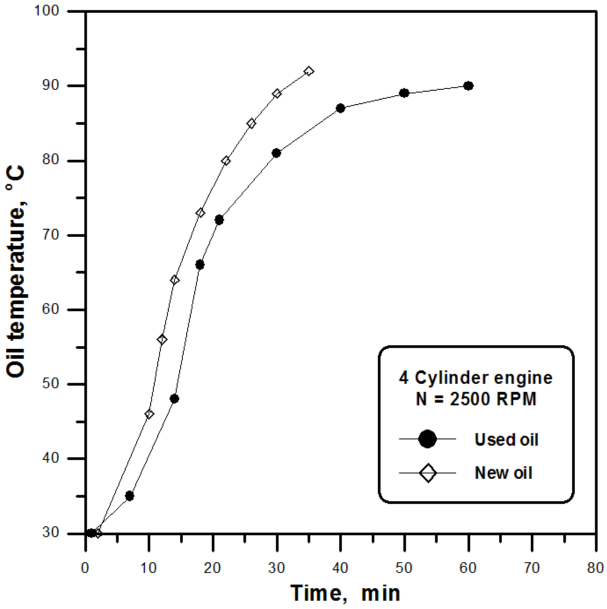

Fig. 13 Comparison of oil temperature (used and new oil) with working time at 2500 rpm.

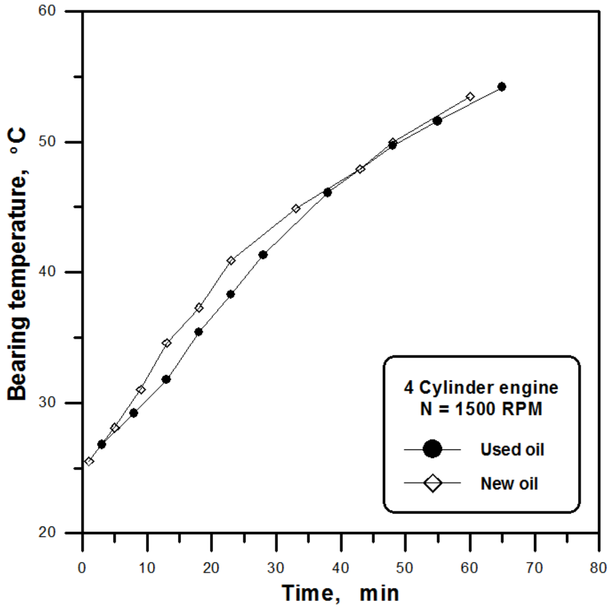

Fig. 14 Variation of bearing temperature (used oil and new oil) of four cylinder engine at 2500 rpm

4. CONCLOSION

- It was obtained that the heat transfer rate, oil temperature, and bearing temperature were increasing the increasing the operating time and engine rpm.

- **The** overall heat transfer coefficient increases with increasing the operating time and engine rpm.

- Heat transfer rate, oil temperature, and bearing temperature increase with the increase of engine cylinder number.

- New and fresh oil plays a positive role in heat transfer process and in protecting the bearing from frictional wear.

5. REFERENCES

[1] H.H. PriebschJ.Krasser, " Simulation of the Oil Film Behaviour in Elastic Engine Bearings Considering Pressure and Temperature Dependent Oil Viscosity", Volume 32, 1997, Pages 651-659, Elsevier proceeding.

[2] Peng Wang, Ruobing Liang, Yongli Wang, Yuebin Yu, Jili Zhang, Mingsheng Liu, " The numerical investigation of heat transfer enhancement of copper-oil and diamond-oil nanofluids inside the piston cooling gallery", Applied Thermal Engineering (2017).

[3] J.M.Conway and JonesR.Gojon," Paper III (iii) Heat Flow in Crankshaft Bearings", Volume 18, 1991, Pages 33-42, Elsevier proceeding.

[4] Jones, G.J. 'Crankshaft Bearings: Oil Film History', Proceedings 9th Leeds-Lyon Symposium on Tribology, Leeds, September 1982, Butterworth 1983.

[5] Conway-Jones, J.M., Martin, F.A., and Gojon, R., 'Refinement of Engine Bearing Design Techniques' T & N Technical Symposium, Wurtzburg and Nashville 1990.

[6] Jingjie Cao, Ming Jia□, Bo Niu, Yachao Chang, Zhen Xu, Hong Liu, " Establishment of an improved heat transfer model based on an enhanced thermal wall function for internal combustion engines operated under different combustion modes", Volume 195, 1 September 2019, Pages 748-759, Elsevier proceeding.

[7] J. P. Holman, "Heat transfer – 10th edition", 2008.

[8] W. Grzesik, J. Rech, and T. Wanat, " Surface finish on hardened bearing steel parts produced by superhard and abrasive tools", (2007), p.p.255–262