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ESTABLISHING THERMAL BALANCE DURING THE COOLING SYSTEM IMPROVEMENT OF AN AIR-COOLED ENGINE

The object of research is the air-cooling system, for F4L912 direct injection diesel engine (mounted on the bench), manufactured by the Motor Enterprise (EMO). It is a naturally aspirated inline 4-cylinder engine. Maximum engine power is 49 kW obtained at maximum speed rotation of 2300 rpm. Air cooling is a critical aspect of engine performance, and studying it experimentally can provide valuable insights into the engine's thermal behaviour and efficiency. One of the most problematic places is the high local temperature of the 4th cylinders sleeves. An innovative improvement of the cooling system is proposed. It is based on increasing the cooling air flow. It consists in the installation of new driving pulleys of the blowing turbine with different diameters. The use of these new pulleys allowed moderating the wall temperature of the liner and the cylinder head of the 4th cylinder and the thermal rebalancing of the engine. Significant improvements have been noted in cylinder wall temperature, exhaust gas temperature, and lubricating oil temperature. Drawing up the heat balance enabled us to quantify the useful work, the heat lost in the cooling water, the heat lost through the exhaust gases, the heat carried away by the lubricating oil and other losses (losses not accounted for). It is clear from the results that the high temperature in the engine has indeed been reduced and the cooling performance of the whole engine has been improved. The results show that the increase in airflow produced an improvement in cooling conditions as well as a reduction in exhaust gas temperatures which will have a significant impact on reducing NOx emissions. In future work, it is planned to improve the cooling system of the Emo F4L912 engine, by studying the effects of the geometry, number, and inclination of the turbine blades on the air flow supplied.

Keywords: *diesel engine, F4L912, air cooling, piston seizure, airflow, hot climate.*

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1. Introduction

Despite the announcements of many countries that they are likely to abandon internal combustion engines in the coming years in favor of electric motors. Various constraints will not allow this as quickly as expected. Indeed, some types of mobility or environments simply do not lend themselves to battery or fuel cell electric propulsion. The scientific and technical progress will keep internal combustion engines or hybrids running for decades, by further improving the efficiency of combustion engines, using conventional fuels and especially biofuels. The cooling system is essential to the functioning of all engines. The entire cooling device must operate in harsh and extreme environments such as deserts and Nordic countries. Liquid cooling systems are known for their high efficiency in heat transfer, making them suitable for high-performance engines, where large amounts of heat need to be dissipated [1–4]. The air-cooling system is a cost-effective and lightweight solution for cooling engines, as it uses ambient air to dissipate engine heat. One approach to improving the low heat transfer coefficient is to increase the surface area in contact with the ambient air. To improve the low heat transfer coefficient the first approach consists in increasing the surface in contact with the ambient air, to which it is possible to associate the use of forced air to increase the air flow on the engine by the use of fans or blowers [5, 6]. The design of the fins and the arrangement of the finned surfaces can have a significant impact on the overall cooling performance of the engine, so optimizing these features is a crucial aspect of designing an effective air-cooled engine [2, 7, 8]. The design and optimization of air-cooled diesel engines is still an important area of research, and advances in this field can lead to improved cooling performance, reduced weight and cost, and enhanced fuel efficiency [9]. The design and optimization of these engines for use in emerging countries in the South are facing new challenges, including the need for reliability and minimal maintenance, as well as the added constraint of operating in a hot climate. These engines must be designed and optimized to meet these demands while still delivering efficient performance [9–12]. Authors of [11] conducted an experimental study on the

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instant flux and surface temperatures of a 4-cylinder air-cooled direct injection diesel engine, and observed the effect of rotational speed on the cylinder head temperature, which showed an increase with increasing rotational speed and exhaust gas temperature. Authors of [12] studied the operation of the EMO F4L912 engine under various loads and climatic conditions, to understand the effect of these factors on the air outlet temperature and engine performance. The study included a modification of the engine cooling system by adding fins at the engine inlet to reduce the thermal load on the fourth cylinder. Authors of [4] presented 3D simulations for optimizing the airflow on the external surface of the cylinder air cooling system of two-stroke engines. The aim was to prevent the separation of airflow from the outer surface of the cylinder, which can lead to high-temperature gradients and thermal destruction of the engine. The study resulted in an optimized design of the fins placed outside the cylinder surface. These studies provide important insights into the thermal behavior and performance of internal combustion engines, and can be used to improve their design and efficiency. Authors of [13] conducted a numerical study to analyze the impact of modifying a cooling air duct to introduce cold air into the cylinder head of an air-cooled engine. The study found that the cooling heat loss in a modern engine can be as high as 49 % at low load, 36 % at medium load, and 27 % at full load, and that this loss can be reduced to improve engine efficiency. These studies provide valuable insights into the thermal behavior of internal combustion engines and can be used to improve their performance and efficiency. In 1934, the results of authors of [14] had shown that only the air speed and the pitch between the fins in an air-cooled engine had an impact on the value of the surface heat coefficient. These results were later confirmed by those of authors of [15–17]. The purpose of this experimental study was to investigate the effects of varying fan air flow rates, load, and operating speed on the performance, wall temperature, exhaust gas temperature, and lubricating oil temperature of the EMO F4L912 diesel engine.

The aim of research is to gain a better understanding of the cooling process and heat transfer in air-cooled diesel engines, with a focus on avoiding piston seizure, which is a recurring issue with this type of engine. The study was conducted in several steps, starting with an inventory of the engine cooling in its initial state, followed by an increase in airflow by adding extra air to the cooling system from an external source, and finally, an increase in airflow by increasing the speed of the cooling turbine. The final step was to establish the thermal balance for the studied cases. The results of this study can be used to improve the cooling performance of air-cooled diesel engines and prevent issues such as piston seizure.

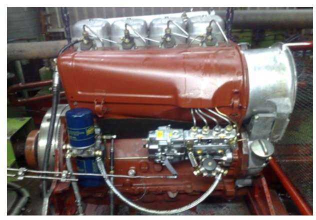
2. Materials and Methods

2.1. Engine. The EMO F4L912 engine is built on the principle of uniformity, making it a versatile and reliable option for powering various types of vehicles and machines. The engine is air-cooled, 4-stroke, 4-in-line, modular design, and direct injection. These features make it well-suited for use in tractors, buses, trucks, and various public works machines. The technical specifications of the engine are detailed in Table 1 and Fig. 1.

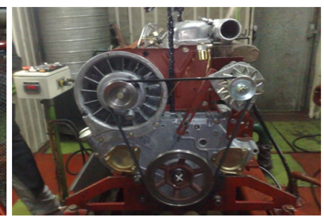
Table 1

Technical characteristics of the EMO F4L912 engine

Engine type	EMOF4L912
Maximum engine power	49 kW
Maximum engine speed	2300 rpm
Cylinder number	4
Cooling system	Air-cooled
Bore	100 mm
Stroke	120 mm
Displacement volume	3768 cm^3
Maximum torque at 1550 rpm	215.8 N·m
Specific consumption	212 g/Wh



a



b

Fig. 1. The engine and its cooling system: *a* – side view; *b* – face view

2.2. Test bench. The Schenck test bench is specialized equipment designed for testing and analyzing diesel engines. The test bench incorporates a hydraulic brake system connected to the engine using a universal joint. The purpose of the brake is to load the engine and simulate real-world conditions during testing engine performance, such as power output, torque, and fuel efficiency. The control panel includes instruments for measuring engine parameters, including temperature measurements at various points such as the air inlet, air outlet, and lubricating oil. Digital indicators on the control panel display the time required to consume a specific amount of diesel fuel. This measurement helps evaluate the engine's fuel consumption rate and efficiency during the testing process. The temperature of the exhaust gases is measured and displayed on a digital indicator. The rotational speed of the engine is shown on a digital display or a tachometer. The test bench incorporates a fuel tank that is positioned at a higher level than the ground. This elevated placement ensures a continuous supply of fuel to the engine during testing.

The test bench utilizes a closed circuit that circulates water. This water circulation system is used to cool the hydraulic brake, which is an integral part of the test bench. Infrared thermometry, also known as infrared temperature measurement, is utilized to measure wall temperatures. The infrared radiation emitted by the engine walls is converted into temperature readings, allowing operators to monitor the temperature distribution and variations across different points on the walls. To measure the temperature of the air and exhaust gases, a device operating on K-type thermocouples is commonly used. These findings indicate a thermal non-homogeneity among the various cylinders, which can be attributed, partly at least, to the complex aerodynamics around the cylinders [12]. The main causes of engine failures remain disturbances, irregularities, or overloads of a thermal or mechanical nature; with the exception of incorrect piston and cylinder sizing (very rare), piston seizure usually occurs when there is a flame passage between the rings, piston, and liner, or when the cooling system is inefficient. The analysis of statistics and previous measurement campaigns has revealed valuable insights into heat transfers within the engine, specifically regarding piston seizure. The clearance between the piston and the cylinder can be reduced intolerably during operation, see even disappear completely following a bad rib of the allied parts, deformation of the cylinder, or thermal overload. In addition, the piston reaches much higher temperatures than the cylinder, which leads to different expansion rates of the piston and cylinder during operation. The piston then undergoes greater thermal expansion than the surrounding cylinder. Furthermore, aluminum-based materials expand twice as much as gray iron, which is the case with our engine. In Fig. 2, under the effect of temperature, engine liners can exhibit differential expansion or contraction along their length. If the liner experiences a temperature rise, it generally tends to expand. However, due to variations in heat distribution and structural constraints, the expansion may not be uniform across the entire length of the liner and at the ends with shrinkage in the middle of up to 0.62 mm. Compared to the piston, which tends to expand in the middle with shrinkage on the ends. The piston skirt was seized around most of its circumference. The seized surfaces are dark in color, rough and heavily eroded. The ring area is only slightly affected by torn-off material that has come up. On air-cooled engines, the source of this type of seizure is overheating due to fouling of the outer cylinder walls, broken fins, or degraded cooling air ventilation [12]. To find out the most frequent type of seizure and the causes that cause this harmful phenomenon, let's start by studying the most exposed parts. According to the results of factory data surveys, the cylinder-piston pair is the organ most exposed to the seizure phenomenon. The statistics made over 3 years, show that the 4th cylinder is the most affected.

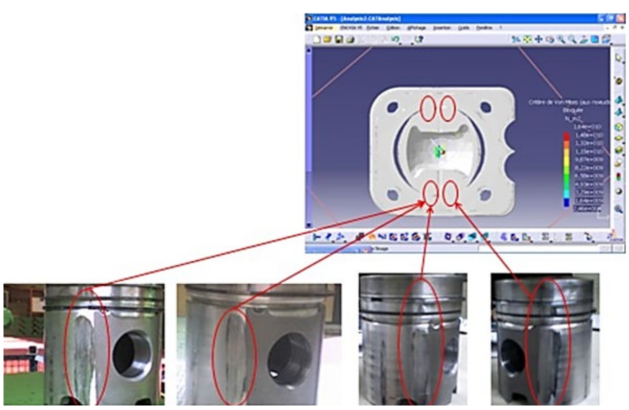


Fig. 2. Seizure piston

Authors of [12] have found two major drawbacks in the cooling of the engine: a seizure and a high temperature in the 4th cylinder due to bad cooling. To overcome these two drawbacks, our approach focuses on the effects of adding external air on the left side of the air guide (Case I, Fig. 3), at the access of the turbine grid (Case II, Fig. 3), on the top of the air guide (Case III, Fig. 4), injecting each time 04 flow rates corresponding to the opening of the control valve of the additional air (25, 50, 75 and 100 %, Table 2).

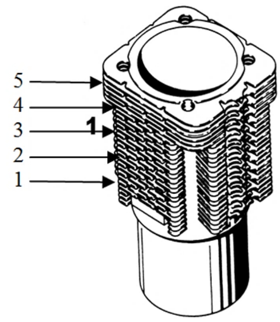


Fig. 3. Positions for measuring the temperature: 1 – 24 mm; 2 – 50 mm; 3 – 75 mm; 4 – 100 mm; 5 – 120 mm

a *b*

Fig. 4. Additional external air added: *a* – air injection on the front (Case I); *b* – air injection on the side (Case II)

Table 2

Valve air flow opening

Code	Valve opening, %	Air flow, m^3/h
G0	0	2390.00
G1	25	2405.12
G2	50	2407.14
G3	75	2408.00
G4	100	2408.72

At the end and for convenience, a set of turbine pulleys with different diameters (Table 3 and Fig. 3) was used, to increase the speed of rotation of the turbine and consequently the flow of air injected to achieve a better cooling in the four cylinders of the engine.

Table 3

Turbine rotation speeds as function of pulley diameters

Pulley	Diameter, mm	Turbine rotation speed at 2300 rpm	Air flow at 2300 rpm, m^3/h	Evolution, %
Original	91	5410	2390	–
Cas IV	88	5594	2471	3.3
Cas V	85	5790	2558	6.5

The air velocity at the exit of the turbine rotating at 2300 rpm was measured by hot wire at 18 m/s. A heat balance was established for all cases studied. Fig. 5 illustrates the temperature measurement positions on the walls of each cylinder, including the fins and cylinder head walls. The measurements are taken from the bottom (position 1) to the top (position 5) of the cylinder [12]. The correlations for predicting heat transfer are obtained from the literature reference [10], it indicates that specific equations or models for calculating heat transfer are being utilized. These correlations are likely derived from

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experimental or theoretical studies conducted by researchers and documented in the referenced literature. The total heat supplied by the fuel Q_T (kW) can be calculated based on the lower calorific value l_{hv} (kJ/kg) and the fuel consumption f_c (kg/h) using the following equation:

$$Q_T=(l_{hv}\cdot f_c)/3600. \quad (1)$$

Based on the equation (2), the heat supply converted to useful work (Q_E) is given by the brake horsepower (p_f) in kilowatts (kW).

$$Q_E=p_f. \quad (2)$$

Q_{cool} evacuated by the engine is calculated from the heat coefficient on the gas side, then the estimate of the temperature inside the jacket and that of the outside wall which is measured at the same time. The estimate of the total exchange surface on the air side is the sum of the surface of the fins and the outer surface of the cylinders.

a *b*

Fig. 5. Additional external air added: *a* – Case 0 (∅=91 mm), Case IV (∅=88 mm), Case V (∅=85 mm); *b* – on top of air guide (Case III)

Knowing the airspeed, let's calculate the cooling airflow, the Reynolds, the Nusselt number and the transfer coefficient. Then let'scalculate the efficiency of the fins, the overall heat transfer coefficient, and finally the quantity of heat evacuated per cylinder which let's multiply by 4 (the number of cylinders).

Based on the equation (4), the calculation of the heat lost through the exhaust gases Q_{exh} takes into account several parameters:

– C_{sh} – total mass (air+fuel) in kilograms per hour (kg/h);
– n_t – the total number of combustion products (kmol/kg), which includes CO_2, H_2O, O_2, and N_2;
– m_{cp}^{tr} – specific heat molar capacity for air;
– m_{cg}^{t2} – specific heat molar capacity for exhaust gases;
– T_{exh} – temperature of the exhaust gases in Kelvin (K);
– l_r – real air quantity;
– T_2 – ambient air temperature in Kelvin (K).

To calculate Q_{Exh}, it is possible to use the following equation:

$$Q_{Exh}=c_{sh}/3600\cdot\begin{bmatrix} n_t\cdot m_{cp}^{tr}\cdot(T_{exh}-273)- \\ -l_r\cdot m_{cg}^{t2}\cdot(T_2-273)\end{bmatrix}. \quad (3)$$

The equation for calculating the unaccounted heat losses (Q_{others}), which is given by the difference between the total heat supplied by the fuel (Q_T) and the sum of the heat converted to useful work (Q_E), the heat taken from the engine by the cooling air (Q_{cool}), and the heat lost through the exhaust gases (Q_{Exh}).

The equation for calculating Q_{others} is as follows:

$$Q_{others}=Q_T-(Q_{Exh}+Q_{cool}+Q_{exh}). \quad (4)$$

3. Results and Discussion

3.1. Actual state. In the case of air-cooled engines, it is common for the fins to have an asymmetric distribution around the cylinder. This design approach aims to compensate for the non-uniform airflow distribution around the cylinder surface. By channeling the airflow effectively, the engine's cooling system can dissipate the generated heat in a better way and maintain the desired temperature levels. The reference sources [3, 8, 17] likely provide more detailed information on the specific design considerations. The analysis of wall temperatures in the four cylinders of the engine, considering the five measurement points, reveals differences in temperature between the points and across the cylinders (as depicted in Fig. 6). It is observed that the temperature increases as moving from position 1 to position 5. This temperature trend can be attributed to the combustion zone being located closer to position 5. The combustion process in an engine generates high temperatures, and these temperatures are highest near the combustion zone. As the combustion progresses and moves towards position 5, the heat transfer to the cylinder walls increases, resulting in higher wall temperatures at that location. The temperature differences between the various points and cylinders can be attributed to several factors, including the non-uniform distribution of heat within the combustion chamber, variations in coolant flow, differences in heat transfer coefficients, and the effects of combustion dynamics and gas flow patterns. It is evident that cylinder No. 4 exhibits the highest temperatures compared to the other cylinders in the engine. This temperature disparity can be attributed to a lack of cooling air, as stated in reference [12].

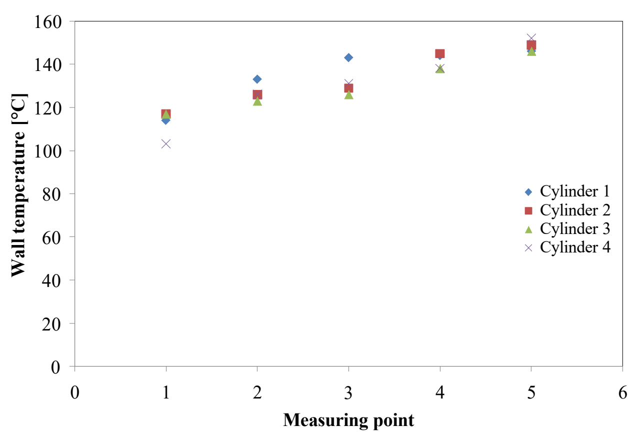

Fig. 6. Wall temperatures on the 5 measurement positions and on the 4 cylinders of the engine at full load before modification (Case 0)

3.2. Modification of the cooling system. The first modification implemented in the engine involved injecting an additional quantity of external air from the compressed air network into the cooling system. This was done at four different flow rates (15 to 19 m^3/h). The engine was operated at its maximum speed of 2300 rpm and under full load. The effects of the increased airflow on the wall temperatures are depicted in Fig. 7. The heat transfer process is enhanced leading to lower wall temperatures. In addition to the air injection at the turbine inlet, it is necessary to understand the importance of the positioning of the air inlet on the thermal fields. For this purpose, three positions were determined: on the left side (Case I), at the turbine inlet (Case II), and on the top (Case III) of the air guide at the level of cylinder No.°4, the most thermally loaded (Fig. 3, 4). The results are transcribed in Fig. 7 where it is possible to notice the effects of the positioning of the air injection on the temperatures obtained on the 4 cylinders. The consequences are different for each case. In the majority of the cases, there is a decrease in the temperatures in all the cylinders. In the original case let's find increasing differences in the temperatures starting from the cylinder 1 the least loaded towards the cylinder 4 the most loaded, and which is delayed compared to the turbine and consequently the least served by the air jet. The case I where the additional air supply is done on the left side of the air guide (Fig. 3), lowers the temperature of the 4 cylinders while maintaining a thermal imbalance. Case II where the additional air is injected at the inlet of the turbine stator (Fig. 3) confirms the downward trend with in addition a thermal balancing between the cylinders due to a good homogenization of the airflow. The last situation, Case III, causes an increase in cylinder 2 to temperatures exceeding the original case. From these results, it is possible to conclude that the best location for the injection of the additional air is the turbine inlet because this situation allows a better mixing and therefore more homogeneity. The temperature drop due to the different configurations and flows can be estimated at 12 % for cylinder 1, 11 % for cylinder 2, 10 % for cylinder 3 and finally 6 % for cylinder 4. The cylinder 4 is the least sensitive to the changes undertaken and this is due to its positioning closest to the turbine and in an eccentric manner.

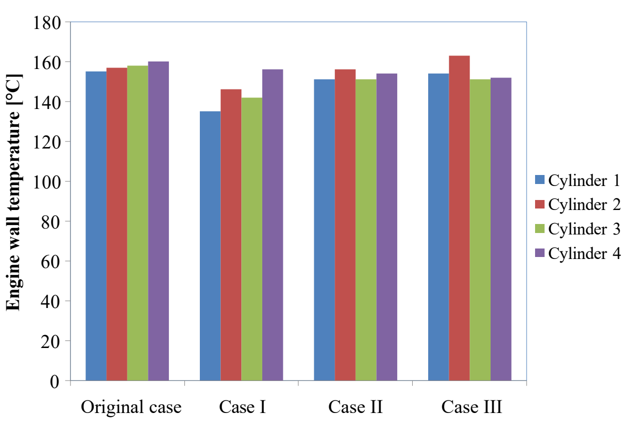

Fig. 7. Wall temperature of the different cylinders

Fig. 8 shows the variations of the wall temperature of the 4th cylinder for all the cases studied (Case 0, I, II, IV and V) according to the load applied to the engine. It clearly appears that the load increases the wall temperature and that the increase of the airflow lowers it. Case IV, which corresponds to the modification of the turbine pulley to a diameter of 88 mm, generating a flow rate of 2470 m^3/h, leads to a reduction of the wall temperature of 11 °C, i. e., a drop of 7 %. The last situation corresponds to the Case V, i. e., a pulley with a diameter of 85 mm for an airflow rate of 2560 m^3/h resulting in a maximum temperature drop of 18 °C, i. e., a 12 % variation on cylinder No. 4. The evolutions on the other cylinders are approximately of the same order of magnitude.

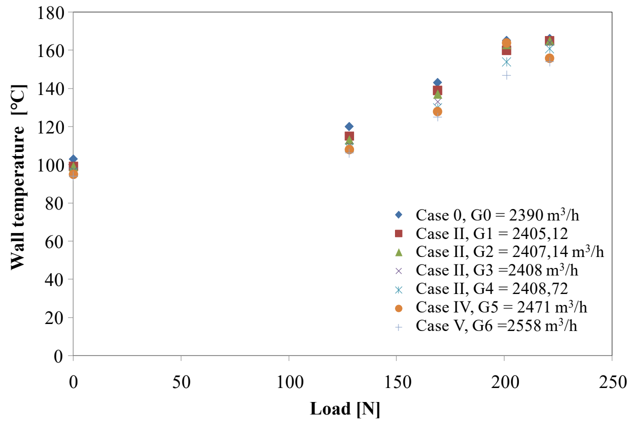

Fig. 8. Wall temperature of the 4th cylinder

Fig. 9 presents the relationship between the exhaust gas temperature and the variation of air flow and engine speed at full load. It shows how changes in these parameters affect the exhaust gas temperature. The effects of the speed of rotation on the temperatures are clearly visible.

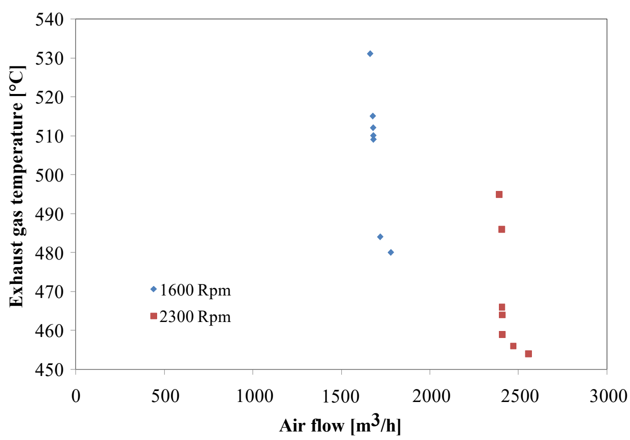

Fig. 9. Exhaust gas temperature

The rotation speed of 1600 rpm generates higher temperatures than at 2300 rpm because of the higher load and the lower rotation speed of the engine which sends de facto less air to the cylinders. At 1600 rpm, an increase in air flow from 1650 to 1750 m^3/h (5.7 %), results in a reduction in exhaust gas temperature from 536 to 484 °C (9.7 %). At 2400 rpm, an increase in air flow from 2400 to 2600 m^3/h (7.6 %), results in a reduction in the exhaust gas temperature from 495 to 453 °C (8.4 %). Let's also notice that the exhaust gas and lubricating oil temperatures decrease with the increase in airflow. This can be explained by the fact that less heat is removed from the exhaust gases by the cooling system (Fig. 10). The reduction of these exhaust temperatures will have a significant impact on the reduction of NOx from the engine. The amount of heat removed through the liner walls becomes greater because of the improved heat transfer coefficient and thus Q_{exh} (Fig. 10). This improvement in heat transfer is reflected in a decrease in exhaust gas temperatures. Reading the results obtained in Fig. 7–9, the beneficial effect of increasing the air flow by varying the diameter of the turbine drive pulley is highlighted. This approach for improving the cooling system and therefore the reliability of the EMO F4L912 engine is very easy to implement at the engine manufacturer level, because it only involves decreasing the diameter of the pulley at the level of their machine shops. So, an improvement without any investment or significant engine modification.

3.3. Thermal analysis of heat balance. Heat balance analysis is a common practice in experiments with internal combustion engines. The goal is to assess the efficiency of the engine's energy conversion process and identify areas where improvements can be made [3, 6, 10, 17]. It is essential to know the percentages transformed and lost at the level of each part. Fig. 10–13 present the results of the heat balance of the EMO F4L912 diesel engine according to the different air flow rates. Fig. 10 presents the percentage of heat lost in the exhaust gas versus the air flow rates in three different cases (3 diameters of the turbine pulley: ∅=91, 88, and 85 mm (Table 3) at full load. It is 35.59 % for Case 0, 34.85 % for Case IV, and 34.32 % for Case V, i. e. a reduction of 3.6 %. The heat lost in the exhaust gases is greater for the first case than for the second, itself greater than the third.

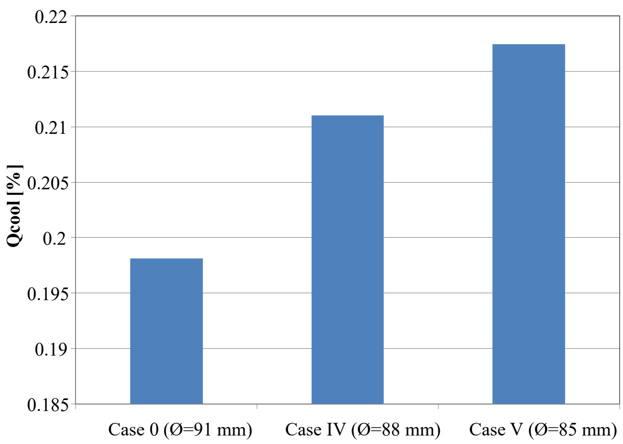

Fig. 10. Heat ratio in the exhaust

The amount of heat dissipated through the liner walls becomes greater as a consequence of the improvement in the heat transfer coefficient and therefore on the burnt gases-wall-cooling air heat exchange (Fig. 10). This improvement in heat transfer is reflected in the reduction of exhaust gas (Fig. 9). A reduction in the amount of heat in the exhaust gases means a reduction in the exhaust gas temperatures (Fig. 9), which may have an impact on the reduction of NOx.

Fig. 11 shows the ratio of heat lost to the cooling medium versus air flow rate in three different cases (∅=91, 88 and 85 mm). The amount of heat in the cooling system varies proportionally to the flow rate. Thus, it is possible to note that the percentage of heat lost in the cooling system is 19.81 % for Case 0, 21.10 % for Case IV, and finally 21.74 % for Case V with an increase of 8.8 % between Case 0 and Case V. The range of heat quantity of cooling is lower for the first case than the second which is itself lower than the third case. As airflow increases, the amount of heat in the cooling system increases as well as the percentage due to improved transfer coefficients. From Fig. 12, it is possible to see that the heat quantity Q_e remains constant regardless of the cooling fluid flow rate. This indicates that Q_{eff} coefficient has no relation to the airflow; it depends only on the speed of rotation which is constant at 2300 rpm. This behavior is the consequence of the optimum operating regime obtained for load conditions, which induces minimal useful work differences.

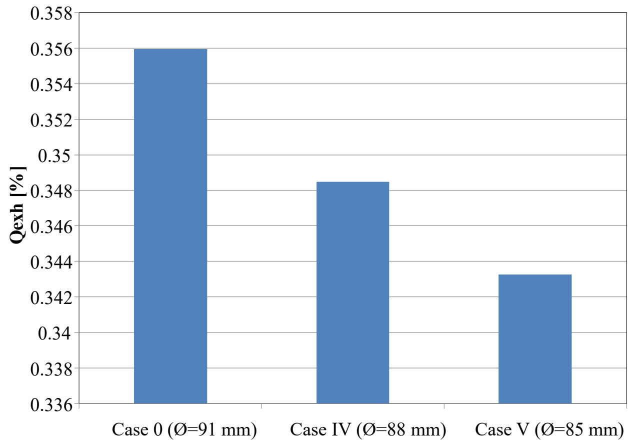

Fig. 11. Heat ratio in the cooling system

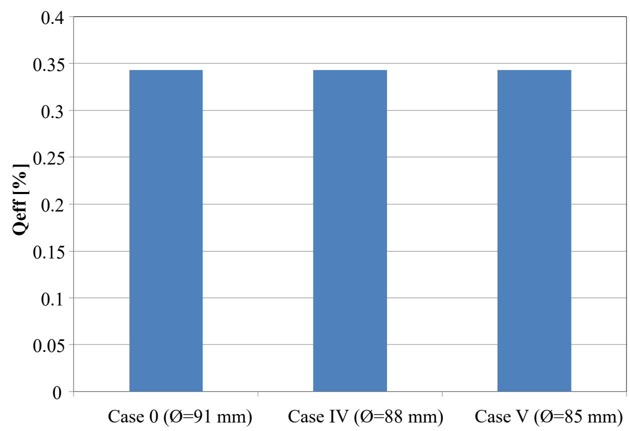

Fig. 12. Power on shaft

Fig. 13 shows the variation of the quantity of other heat versus air flow in three different cases. It can be seen that the Q_{others} quantities vary in inverse proportion to the airflow rate, when it increases; the other heat quantities are essentially losses in the lubrication system. The amount of other heat is highest for the first case followed by the second case and then the third case.

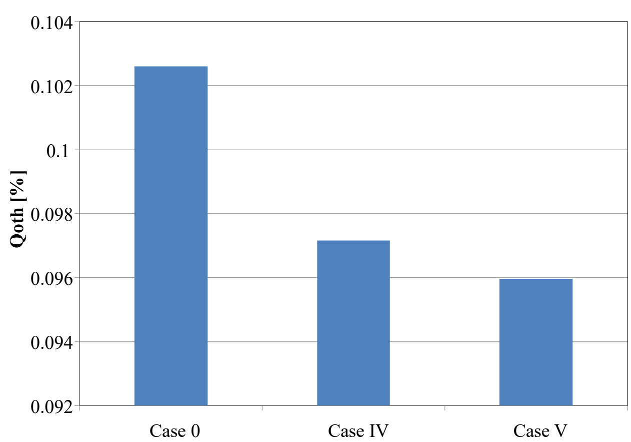

Fig. 13. Other heat

However, this approach has its limits in terms of increasing the diameter of the turbine pulley, which are mechanical limits. Now, that the effects of increasing air flow on the moderation of cylinder wall temperatures (Fig. 7–9) and on the heat balance (Fig. 10–13) are established, it would be appropriate to work on another lever for increasing the cooling air flow by investigating in more depth the effects of the geometry, inclination and number of blades of the turbine.

4. Conclusions

The current work, presents an experimental study on the cooling of the EMO F4L912 air-cooled direct injection diesel engine. The increase in air flow was achieved by adding external air and then changing the diameter of the blower pulley. The air flow has an important influence, the more important it is, and the better the cooling will be. Let's also note that the place of injection of the air or air is important. The injection at the top of the engine is the best cooling compared to the other two cases (increasing about 6.5 %). The most suitable solution was the resizing of the blower pulley to increase its speed of rotation and therefore the flow rate supplied. Comparing the results of this engine (before and after increasing the cooling airflow), it is possible to see that there is a significant temperature variation in the walls of the four cylinders, as well as a decrease in lubricating oil, exhaust gas temperatures (9.7 %) and therefore most likely a decrease in NOx. The modifications made have allowed a reduction in the wall temperature which has reached 18 °C corresponding to 12 % on cylinder No. 4. Added to that, the real risks of seizure that it is possible to notice are now very unlikely. It appears that a bad cooling can have an impact on the good functioning of the engine as well from a performance point of view (power, torque and consumption) as from an ecological point of view (exhaust gas temperature which induces the production of NOx and other unburned).

Conflict of interest

The authors declare that they have no conflict of interest in relation to this research, whether financial, personal, authorship or otherwise, that could affect the research and its results presented in this paper.

Financing

The research was performed without financial support.

Data availability

The manuscript has no associated data.

Use of artificial intelligence

The authors confirm that they did not use artificial intelligence technologies when creating the current work.

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