computer simulation, mathematical modeling, cooling systems, energy balance

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MODELING OF THE HIGHER PRESSURE COOLING SYSTEM FOR TRANSPORT VEHICLES ENGINES

Summary. This paper presents a model of the engine cooling system for high coolant temperatures developed through AmeSim software. It presents the results of temperature course simulation, pressure course, and liquid cooling pump efficiency. It shows that it is possible to maintain the assumed constant pressure in the system and obtain it at the elevated liquid temperature leading to an increase in overall engine efficiency.

MODELOWANIE CIŚNIENIOWEGO UKŁADU CHŁODZENIA SILNIKÓW POJAZDÓW TRANSPORTOWYCH

Streszczenie. W artykule przedstawiono model układu chłodzenia silnika o podwyższonej temperaturze płynu chłodzącego opracowanego w oprogramowaniu AmeSim. Zaprezentowano wyniki badań symulacyjnych przebiegu temperatury, ciśnienia oraz wydajności pompy cieczy chłodzącej. Wykazano, że istnieje możliwość utrzymania założonego stałego ciśnienia w układzie i uzyskania przy tym podwyższonej temperatury cieczy, prowadzącej do wzrostu sprawności ogólnej silnika.

1. INTRODUCTION

Internal combustion engines known for their low efficiency are still widely used to drive vehicles. Although the introduction of hybrid vehicles is still being developed, technical difficulties cause their application is a matter of the future since other research is still focused on developing internal combustion engines with increased efficiency and reduced toxic exhaust components [2, 4, 10].

The most popular and widely used means of cooling internal combustion engines is through liquid cooling which ensures greater uniformity of temperature around the combustion chamber than direct air cooling to spite the properties of water being limited to the maximum temperature of the coolant [1, 3, 6, 7].

The efficiency of liquid cooling systems can be increased by the use of electronic control work of the assembly equipment, as well less intense engine cooling thus reducing heat loss. In the case of systems in which coolant containing water is applied, raising the boiling point of the coolant requires an increase of pressure in the cooling system which requires a reasonable accommodation system and strengthen its structure. Preliminary scientific research of such a system indicates the possibility of increasing overall efficiency and reducing toxic components in exhaust gases at low engine loads when the engine exhaust temperature of the classic system is too low for efficient catalytic action [5].

The aim of this work was to develop a model of the pressure of an internal combustion engine cooling system as well as checking the possibility of maintaining the established pressure in the system and attaining it at a high coolant temperature, as well as the frequency and scope of the parameters changes controlling pressure and temperature. The range of the work included mathematical system modeling and determining its optimal parameters in commercial AmeSim followed by verifying the simulation results on the test stand.

2. HEAT BALANCE AND EFFICIENCY OF THE INTERNAL COMBUSTION ENGINE

Energy supplied to the piston engine through fuel and air is only partly converted to useful engine work [6, 9]. An analysis of the heat balance of various engine solutions show that effective work can be directly converted to about 25%-45% of the supplied energy (Fig. 1). The remaining resources of energy are dissipated directly or indirectly into the surrounding atmosphere, which is caused by thermodynamic efficiency and the need to reduce the temperature of the materials of the engine components [6 - 8]. About 22%-35% of the energy supplied to the fuel is drained by the same cooling system or, as a result of radiation and conduction, through the engine block, head, oil pan and other engine components. A significant part of about 25%-35% of the heat flows out along with the hot exhaust fumes. Part of the energy resulting from mechanical loses (about 8-12%) further converted to heat friction from the engine mechanism which also must be discharged outside the system [8]. This result in a large number of different streams of heat which flows inside and outside the engine and therefore is often presented in the form of simplified or detailed Sankey's charts [5, 6]. An example of the measured heat flow for the engine 4CT90 as a function of its speed is shown in Fig. 1 [10].

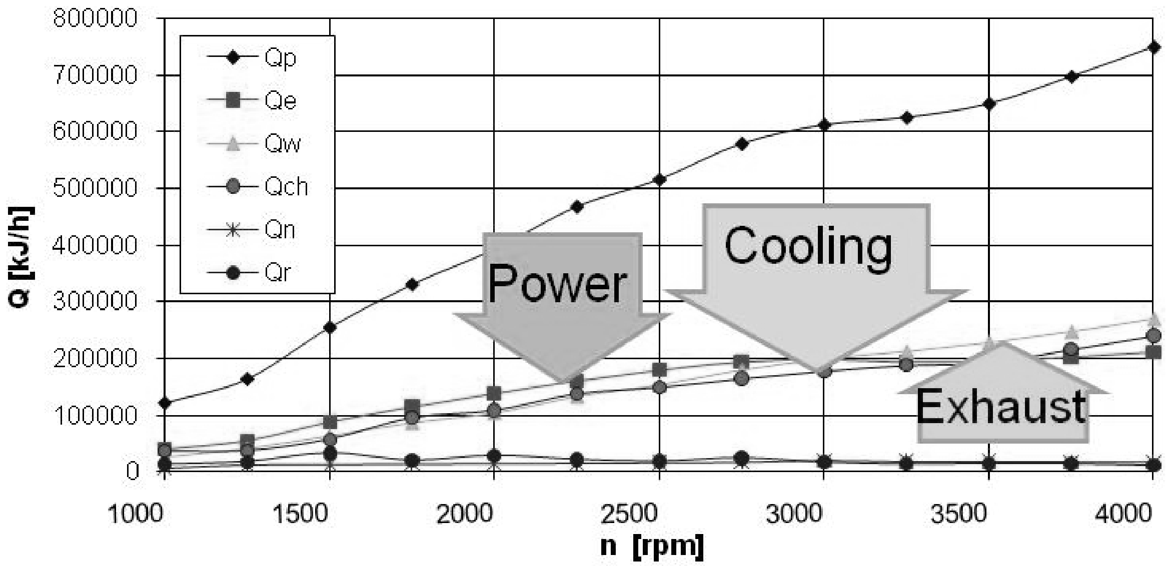


Fig. 1. Comparison of heat engine 4CT90 streams (designation: Q_p – heat from the combustion of fuel spilled, Q_e – heat converted into useful work, Q_{ch} – dissipate heat to the cooling system, Q_w – convection heat from the exhaust gases, Q_r – the rest of the balance, and the heat discharged into the environment from the hot walls of the engine and other losses)

Rys. 1. Zestawienie strumieni ciepła silnika 4CT90 (oznaczenia: Q_p – ciepło wydzielone podczas spalania paliwa, Q_e – ciepło zamienione na pracę użyteczną, Q_{ch} – ciepło odprowadzone przez układ chłodzenia, Q_w – ciepło unoszone ze spalinami, Q_r – reszta bilansu, a w tym ciepło odprowadzane do otoczenia od gorących ścianek silnika oraz inne straty)

Since even 35% of heat can be lost by the same cooling system, studies have been opened on how to improve the existing cooling system by introducing effective thermal management in the vehicles to increase total engine efficiency. One of the ways to do this is to use an electronic work group, as well as less intense cooling of the engine resulting in an increase in coolant temperature and thus reducing heat loss [2, 4].

Checking the possibilities of maintaining the established overpressure in the system and obtaining it at a higher coolant temperature implemented by AMESim software enables the solution of many engineering problems at an early stage of design as well as introduces any eventual changes without the necessity of building expensive prototypes [14, 15].

3. MATHEMATICAL MODEL OF THE ENGINE COOLING SYSTEM WITH AN INCREASED TEMPERATURE OF COOLANT DEVELOPED IN AMESIM SOFTWARE

In AMESIM software was developed the model stand scheme expressed with the help of flowcharts and was performed calculations and simulations showing the courses of pressure, temperature, and coolant pump flow rate for the assumed parameters of pressure (Fig. 2) [11 - 13]. The model of the cooling system was made on the basis of test stand solutions designed and built using the original components of diesel engine 4CT90. The primary source of heat for the test stand are three heating elements with different electrical power to the adjacent wall of each cylinder [9].

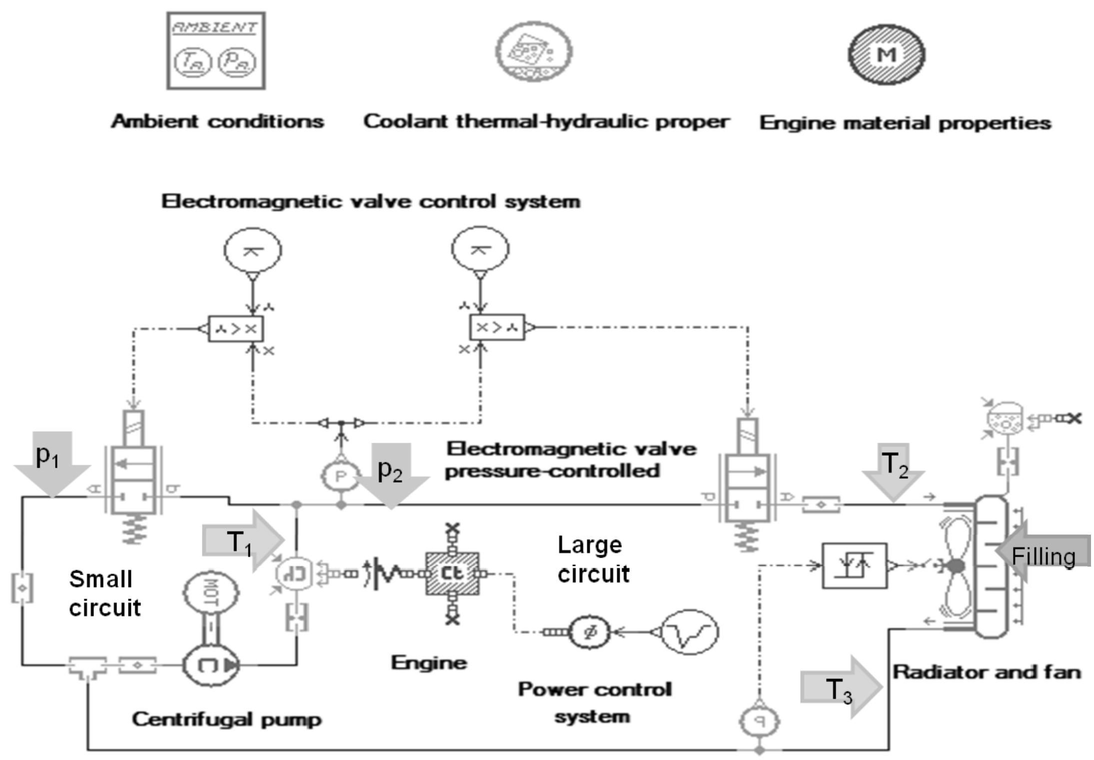


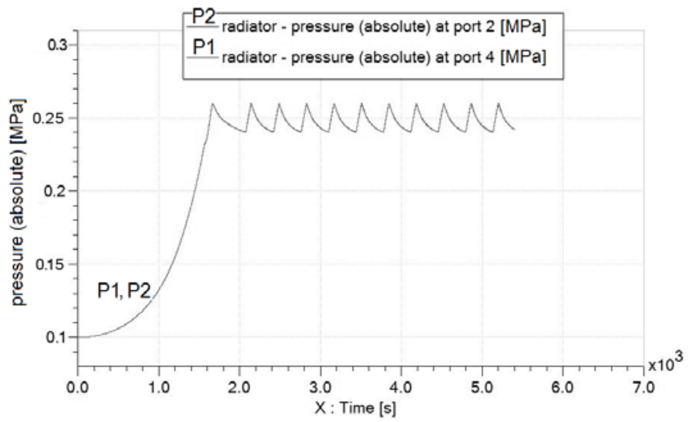
Fig. 2. Model stand scheme of cooling system developed in AMESIM software
Rys. 2. Schemat stanowiska modelowego układu chłodzenia opracowany w oprogramowaniu AMESIM

In the presented model are elements which represent the engine block along with the system responsible for removing heat from the walls and transferring it to the coolant. The control system, through the power of heating, enables regulation of heat as it is discharged to the cooling system. Using the electrically driven centrifugal pump with adjustable speeds, makes it possible for the intensity of cooling not to be dependent on the assumed engine speed. The flow of liquid is the pressure between the small and large circulation and was controlled by solenoid valves. In order to perform the calculation of the cooling circuit parameters (temperature and pressure of liquid, coolant pump flow, operational characteristics of pumps and valves), it was necessary to introduce a large amount of data including, above all, the liquid properties, material properties of the engine, environment parameters, the volume of liquid in a small and a large circulation, the mass of the engine, etc. Detailed information was introduced according to the requirements of the program [2, 4, 15].

4. RESULTS OF THE CALCULATION AND SIMULATION IN AMESIM SOFTWARE

Calculations and simulations were performed for pressures of 0,15 MPa and 0,2 MPa at 96% and 81% of the filling with coolant at a total volume of 11 dm^3. Fig. 3 - 6 shows pressure and temperatures courses for the above values.

a)



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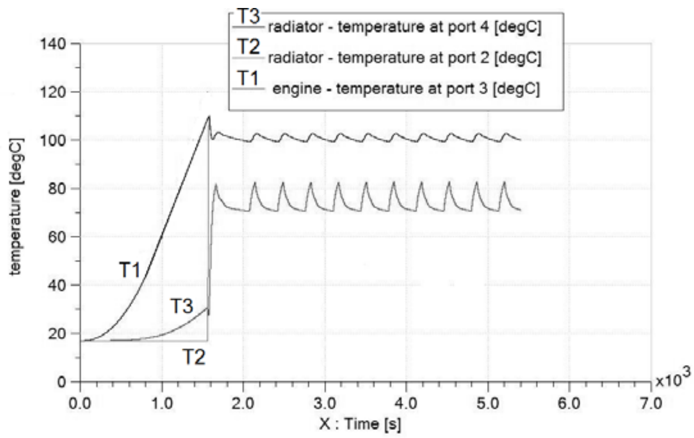

Fig. 3. Course characteristics at the pressure of 0,15 MPa and 96% of the filling with coolant: a) pressure: P1 – small circuit, P2 – large circuit, b) temperature: T1 – out of the cylinder block, T2 – entrance to the radiator, T3 – out of the radiator

Rys. 3. Charakterystyki przebiegów przy ciśnieniu 0,15 MPa i 96% wypełnienia układu w ciecz chłodzącą: a) ciśnienia: P1 – w małym obiegu, P2 – w dużym obiegu, b) temperatury: T1 – wyjście z bloku cylindrów, T2 – wejście do chłodnicy, T3 – wyjście z chłodnicy

The pressure courses, both for 0,15 MPa and 0,2 MPa are very similar for all system fillings. Therefore, the paper presents one of the above mentioned overpressure courses in the system. However, the temperature runs were obtained at these pressures showing in each case.

a)

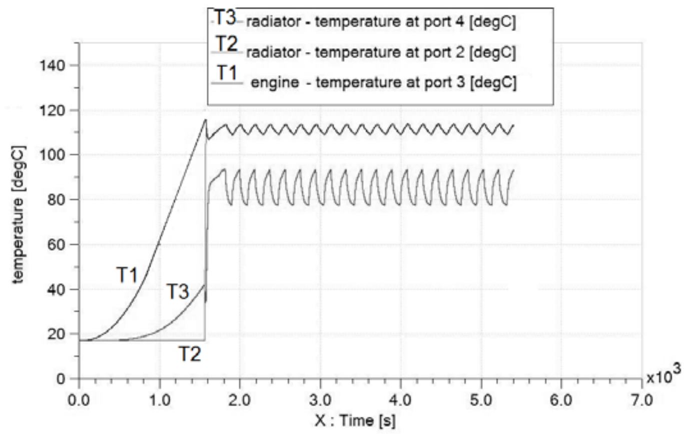

Fig. 4. Courses characteristics at the pressure of 0,15 MPa and 81% of the filling with coolant: a) temperature: T1 – out of the cylinder block, T2 – entrance to the radiator, T3 – out of the radiator

Rys. 4. Charakterystyki przebiegów przy ciśnieniu 0,15 MPa i 81% wypełnienia układu w ciecz chłodzącą: a) temperatury: T1 – wyjście z bloku cylindrów, T2 – wejście do chłodnicy, T3 – wyjście z chłodnicy

During the simulations for pressures of 0,15 MPa and 96% of the filling with coolant, the pressure is maintained within the limits of 0,14 ÷ 0,16 MPa. At the moment the assumed pressure is reached (about 23 minutes) immediately afterward the goal was to maintain the pressure at 0,15 MPa. The pressure course is characterized by uniformity of pressure as well as the possibility to maintain pressure on the possible stabilities. Temperature waveforms at the exit of the engine block and the entrance to the radiator after the assumed pressure are at 100°C for 96% of the filling of the cooling system with coolant. The temperature at the exit of the radiator, however, is between 70°C - 82°C. At 81% of the filling with coolant, the courses of the temperatures are similar but varied in that the temperature at the exit of the engine and the entrance to the radiator grew to 110°C whereas the output ranged from 78°C - 90°C. Such bandwidth, both for the first case and the second, results from switching between large and small circuits.

Further calculations and simulations performed at the pressure of 0,2 MPa and at 91% filling of the system with coolant. For about 27 minutes, a mild increase of pressure took place in the limits of 0,2 MPa maintained within 0,19 ÷ 0,25 MPa. The course is also characterized by pressure uniformity and greater frequency of temperature change with greater density.

Increasing the pressure in the system also confirms that it is possible to maintain the pressure as constant as possible. In this case, the temperature at the exit of the cylinder block and the entrance to the radiator after the assumed pressure are at 110°C for 96% filling of the cooling system with coolant. However, the temperature out of the radiator is included from 78°C - 90°C. As far as the temperature courses are concerned, at 81% of the filling with coolant, the temperature at the exit of the cylinder block and at the entrance to the radiator nearly rose to 122°C, whereas at the exit of the radiator the temperature was between 82°C-100°C.

Since the coolant pump flow in all cases is very similar, changes to the expenditure have been omitted. To the moment of warm-up and attaining the assumed pressure, output is almost constant at about 26 l/min. In order to maintain the pressure as constant as possible, the system is switched between large and small, from hence results the output of the liquid pump which ranges from 21 l/min to 26 l/min.

5. EXPERIMENTAL IDENTIFICATION OF THE MODEL

Experimental research was carried out as a comparative model of diesel engine 4CT90 at an pressure of 0,2 MPa and 96% liquid filling system. The characteristics of course temperature and pressure of coolants before and after the radiator, the temperatures before entering and after leaving the engine as well as the same engine, in addition to the measurements of the coolant pump flow at different speeds were all determined. Warm up on a small circuit occupied about 24 minutes when the pressure in the system had obtained a certain value system switched to a large circuit and then the pressure decreased by about 0,05 MPa, and its value is included within the limits of 0,15 ÷ 0,2 MPa (Fig. 7). The temperature, however, decreased by about 25°C, but it was possible to achieve a maximum temperature of 125°C. Switching frequency increased, with a large circuit switched on for about 2-3 seconds. Before switching on the circulation of the large circuit, a fan was switched on for about 20 to 25 seconds.

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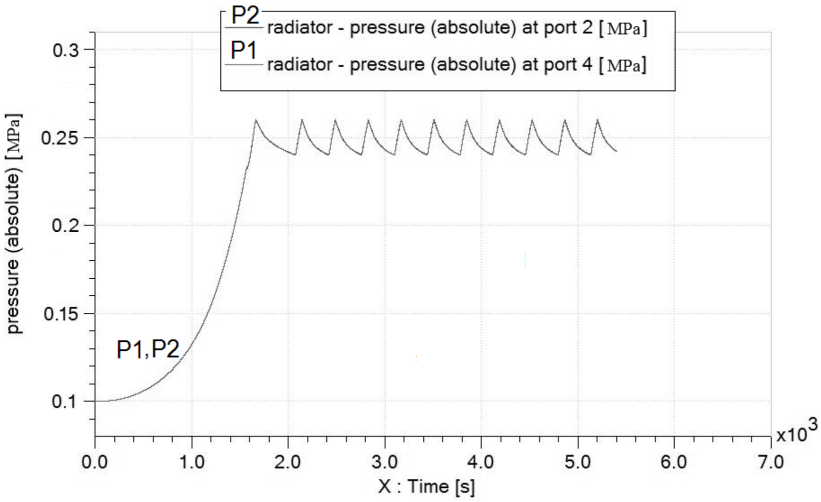

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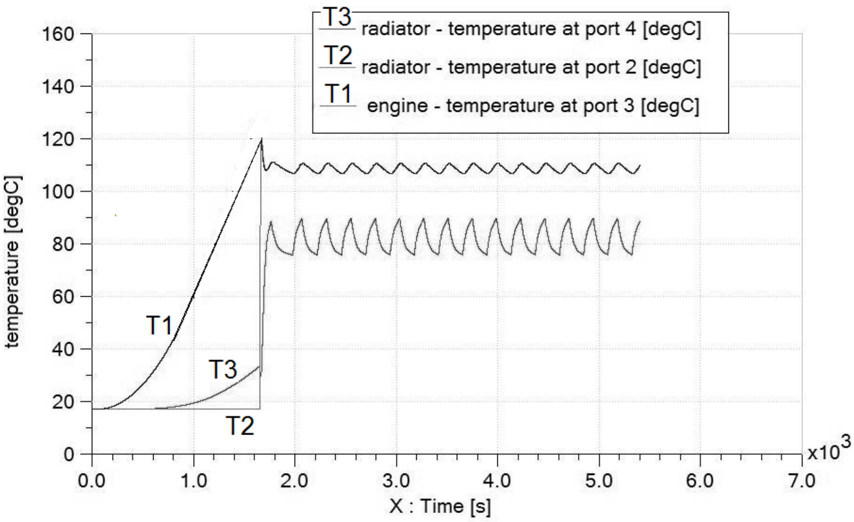


Fig. 5. Course characteristics at the pressure of 0,2 MPa and 96% of the filling with coolant : a) pressure: P1 – small circuit, P2 – large circuit, a) temperature: T1 – out of the cylinder block, T2 – entrance to the radiator, T3 – out of the radiator

Rys. 5. Charakterystyki przebiegów przy ciśnieniu 0,2 MPa i 96% wypełnienia układu w ciecz: a) ciśnienia: P1 – w małym obiegu, P2 – w dużym obiegu, b) temperatury: T2 – wejście do chłodnicy, T3 – wyjście z chłodnicy, T1 – wyjście z bloku cylindrów

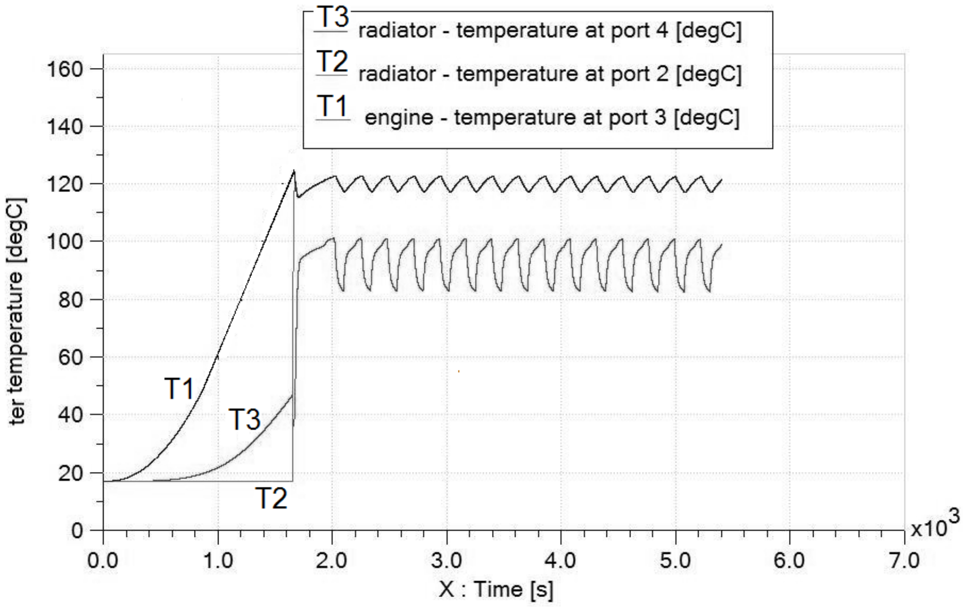


Fig. 6. Courses characteristics at the pressure of 0,2 MPa and 81% of the filling with coolant: temperature: T1 – out of the cylinder block, T2 – entrance to the radiator, T3 – out of the radiator

Rys. 6. Charakterystyki przebiegów przy nadciśnieniu 0,2 MPa i 81% wypełnienia układu w ciecz: temperatury: T1 – wyjście z bloku cylindrów, T2 – wejście do chłodnicy, T3 – wyjście z chłodnicy

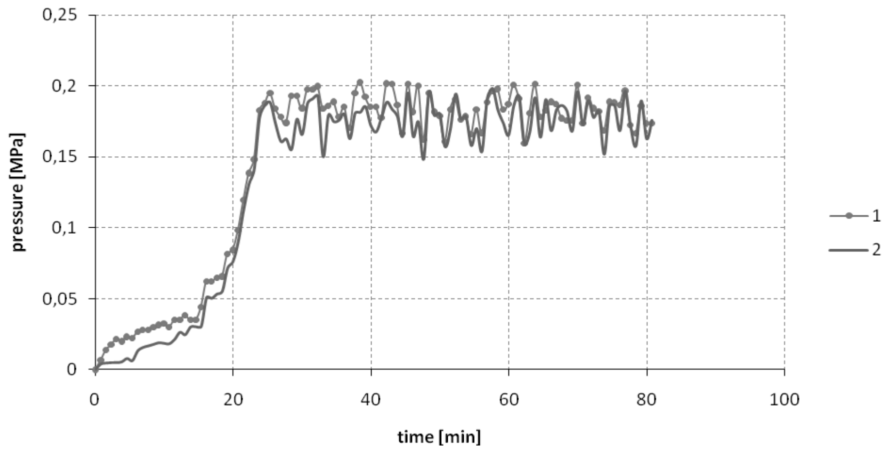
6. EXPERIMENTAL IDENTIFICATION OF THE MODEL

Experimental research was carried out as a comparative model of diesel engine 4CT90 at an pressure of 0,2 MPa and 96% liquid filling system. The characteristics of course temperature and pressure of coolants before and after the radiator, the temperatures before entering and after leaving the engine as well as the same engine, in addition to the measurements of the coolant pump flow at different speeds were all determined. Warm up on a small circuit occupied about 24 minutes when the pressure in the system had obtained a certain value system switched to a large circuit and then the pressure decreased by about 0,05 MPa, and its value is included within the limits of 0,15 ÷ 0,2 MPa (Fig. 7). The temperature, however, decreased by about 25°C, but it was possible to achieve a maximum temperature of 125°C. Switching frequency increased, with a large circuit switched on for about 2-3 seconds. Before switching on the circulation of the large circuit, a fan was switched on for about 20 to 25 seconds.

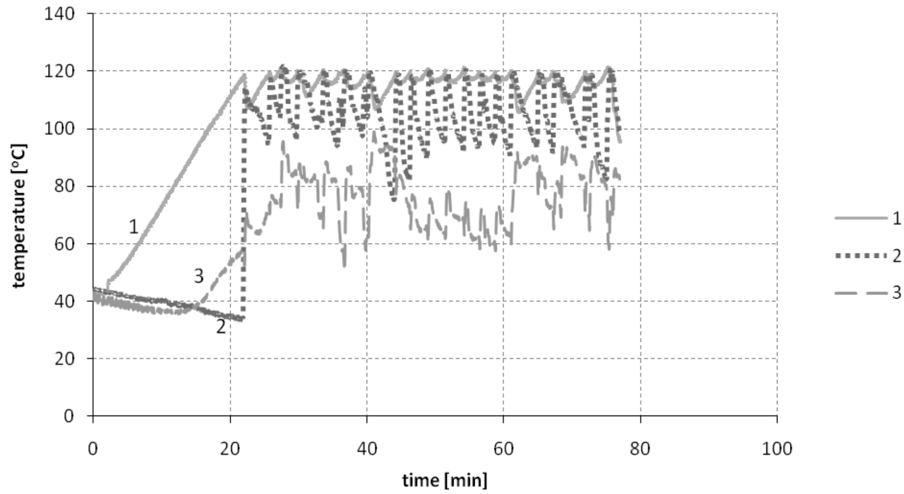
After analyzing all courses, it can be concluded that filling coolant has an impact mainly on the coolant temperature at the outlet of the condenser, and temperature values for different pressure values are very similar. However, with increasing levels of overpressure and reduction of the volume of the fluid, obtaining passes are more even and a smooth change of parameters is the function of time, as well as easier control and obtaining easier runs.

Coolant flow rate was not constant, because at temperatures close to boiling point, vapor bubbles form in the water pump which caused a decrease in the efficiency of liquid extraction shown on the flow meter and the rapid increase in pressure in the system. The result was unevenness clearly visible in the flow of coolant. Therefore, the pump speed is chosen so as to provide optimal expenditure in each case.

Analyzing graphs of the simulation and experimental research, it is noted that the results are very similar, which indicates that the properly constructed simulation model mimics the action of the corresponding experimental engine cooling system. The characteristics show some differences, which may also derive from the fact that the mathematical model used an electronic control of all processes, whereas in studies of actual control takes place by hand.



a)



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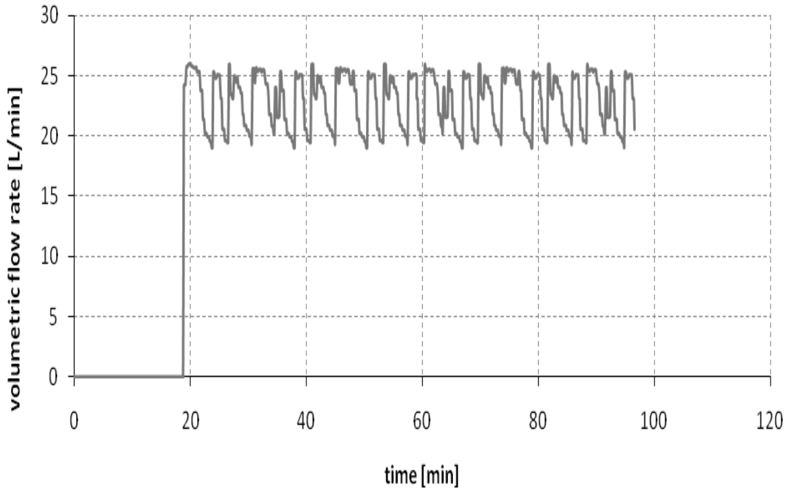

c)

Fig. 7. Courses characteristics at the pressure of 0,2 MPa and 91% of the filling with coolant: a) pressure: 1 – small circulation, 2 – large circulation, b) temperature: 1 – out of the cylinder block, 2 – entrance to the radiator, 3 – out of the radiator, c) coolant pump flow

Rys. 7. Charakterystyki przebiegów przy nadciśnieniu 0,2 MPa i 91% wypełnieniu układu w ciecz: a) ciśnienia w małym i dużym obiegu, b) temperatury: 1 – wyjście z bloku cylindrów, 2 – wejście do chłodnicy, 3 – wyjście z chłodnicy, c) wydatku pompy cieczy chłodzącej

7. CONCLUSIONS

1. The result of the simulation studies found that it is possible to maintain at a nearly constant course temperature in the block and head as well as at entry and exit from the engine in this research. This means that the system can be controlled in such a way to make it possible for maintaining the pressure and temperature at a given level and the acceptable limits.
2. The experimental studies carried out showed the proper working of the system and the satisfactory outcome to the experimental simulation research. For an extended amount of time it was possible to maintain the pressure in the system at a given level.
3. After analyzing all of the courses it can be concluded that filling the cooling system with coolant does not affect the temperature for different values of the assumed pressure cooling system test. However, with increasing levels of overpressure and reduction of fluid volume, obtained courses are more uniform and there is an increase of temperature by a few degrees.
4. Switching between system circuits, control fans, or the intensity of the heating took place automatically. This has also provided a satisfactory coolant temperature. Controls should increase repetition of the parametric changes of cooling liquid.
5. After comparing the mathematical model with experimental results obtained from experience, one may consider that despite slight differences, the simulation model is built correctly. Controlling the cooling of the engine due to the pressure limits obtained similar values of temperature and frequency of these temperature changes. For this reason, the model makes it possible to optimize the parametric characteristics of the object, its method of control, and exploitation. It can also be used during research on the introduction of the changes of modernization, which may be preceded by a simulation using the developed mathematical model, which should reduce the cost of building the next version of the test stand.

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