Article citation info:
JAMROZIAK K, KWASNIOWSKI S, KOSOBUDZKI M, ZAJAC P. Analysis of heat exchange in the powertrain of a road vehicle with a retarder. Eksploatacja i Niezawodnosc – Maintenance and Reliability 2019; 21 (4): 577–584, http://dx.doi.org/10.17531/ein.2019.4.6.

Krzysztof JAMROZIAK
Stanislaw KWASNIOWSKI
Mariusz KOSOBUDZKI
Pawel ZAJAC

ANALYSIS OF HEAT EXCHANGE IN THE POWERTRAIN OF A ROAD VEHICLE WITH A RETARDER

ANALIZA WYMIANY CIEPŁA W UKŁADZIE NAPĘDOWYM POJAZDU DROGOWEGO Z RETARDEREM

The paper presents a heat exchange model for the cooling system of any complex, physical system. Verification of the correctness of the theoretical model was carried out on the example of a vehicle with a combustion engine and additionally equipped with a hydraulic retarder. The results of laboratory tests, which were carried out on an engine test bench, were also performed for the above mentioned powertrain, so as to compare the results of modelling with the results of the tests. Determining the operating parameters of the components of the cooling system aimed at protecting the entire powertrain against overheating is a key task. Theoretical analysis of heat exchange in the powertrain of a road vehicle was carried out, with particular emphasis on the hydraulic retarder (a device braking the vehicle during a descent on roads with a high gradient of the road, mandatory according to the ADR convention). The subject of the study was a mathematical model of a complex cooling system developed by the authors, described by means of balance equations and differential equations. This model was tested with the use of the Matlab-Simulink suite for given load parameters of the cooling system, which were used in tests on an engine test bench. The values of coefficients describing the thermal state of the powertrain were obtained. Simulations were performed for different variants of technical parameters of the expanded cooling system. In this way, individual units and components of the cooling system were optimized so that it fulfilled its role in the assumed operating conditions and the ecologization of emission of energy sources (fuel) and harmful substances.

Keywords: *heavy-duty vehicle operation, vehicle retarder, brake support, heat exchange.*

Praca zawiera model wymiany ciepła układu chłodzenia dla złożonego, dowolnego układu fizycznego. Weryfikację poprawności modelu teoretycznego przeprowadzono na przykładzie pojazdu z silnikiem spalinowym oraz dodatkowo wyposażonym w retarder hydrauliczny. Wyniki badań laboratoryjnych, które przeprowadzono na hamowni również wykonano dla w/w zespołu napędowego, tak aby porównać wyniki modelowania z wynikami badań. Ustalenie parametrów eksploatacji elementów układu chodzenia, którego celem jest zabezpieczenie całego układu napędowego przed przegrzaniem to kluczowe zadanie. Przeprowadzono teoretyczną analizę wymiany ciepła w układzie napędowym pojazdu drogowego ze szczególnym uwzględnieniem zwalniacza hydraulicznego (urządzenia hamującego pojazd podczas zjazdu na drogach o dużym pochyleniu jezdni, obowiązkowe wg konwencji ADR). Przedmiotem badań był opracowany przez autorów model matematyczny rozbudowanego układu chłodzenia opisany za pomocą równań bilansowych i równań różniczkowych. Model ten testowano z wykorzystaniem pakietu Matlab-Simulink dla zadanych parametrów obciążenia układu chłodzenia, które wykorzystywano w badaniach w stanowiskowych na hamowni silnikowej. Uzyskano wartości współczynników opisujących stan cieplny jednostki napędowej. Symulacje wykonano dla różnych wariantów parametrów technicznych rozbudowanego układu chłodzenia. W ten sposób optymalizowano poszczególne zespoły i podzespoły układu chłodzenia, tak aby spełniał on swoją rolę w zakładanych warunkach eksploatacji i ekologizację emisji źródeł energii (paliwa) i szkodliwych substancji.

Słowa kluczowe: *użytkowanie pojazdów ciężkich, zwalniacz pojazdu, wspomaganie hamowania, wymiana ciepła.*

1. Introduction

Modern trucks for the transport of cargo/humans, apart from the conventional braking system, are equipped with additional braking systems which can take the form of an electrodynamic brake, engine brake or hydraulic brake [5, 9, 10, 12]. When driving on long and steep road sections, the vehicle often uses the service brake, which can lead to increased heat load and serious wear to the brake system. This is a dangerous phenomenon affecting the decrease in the efficiency of the mechanical brake due to the thermal recession of the system [3, 4, 28]. Hydraulic retarders, commonly used in commercial vehicles, are auxiliary devices that can reduce vehicle speed by converting the vehicle's mechanical energy into heat energy absorbed by the retarder's working medium that is oil.

Compared to other auxiliary equipment, hydraulic retarders have many advantages such as low weight, high braking torque, long operating hours, good thermal diffusivity and zero environmental pollution. The design and principle of operation of hydraulic retarders/intarders in the literature is quite well described [14, 23, 24]. Nevertheless, the development of new technologies affects the efficiency of these devices and their characteristics are being investigated by many researchers [18, 26, 35]. It is noted that they are mandatory in Europe for vehicles with a gross vehicle weight (GVW) of over 16 tones. Currently, they are also installed in vehicles with smaller GVW parameters [8]. These units are characteristic heat exchangers, the maintenance of which at the required level of efficiency also depends on the powertrain cooling system's capacity [17, 25, 29, 38]. Operation of the hydraulic retarder is periodic and its cooling system is connected to the cooling system of the traction engine.

The capacity of the cooling system must be selected in such a way as to meet the assumed operating conditions of the truck. When considering the thermal capacity of the mass of the powertrain and the efficiency of the cooling system, the thermal capacity received from the retarder over a period of 12 minutes of its operation shall be estimated. This time has been estimated on the basis of average downward slopes existing on the European roads [24]. This requires designers to develop efficient equipment.

Great emphasis was placed on modelling traction performance characteristics versus technical parameters and geometry of rotating elements of the retarder [6, 13, 30], as well as testing the medium flow field structure [23, 27]. Numerous studies have been devoted to mathematical modelling in the analyses of the effect of rotor speed and fill ratio on retarder output torque [15, 31, 32, 33, 34]. Mathematical models are based on complex differential equations of flows, which are often solved using numerical methods [12, 21, 36]. With the use of numerical models in computer simulations it is possible to make certain verifications of the flow rate or the average speed of the working fluid, as well as the braking torque - which was the subject of papers [1, 2, 20, 22]. In the analyzed literature [11, 19, 35, 37], the majority of problems concern the evaluation of retarders efficiency and their optimization, while the influence of the design heat output of the radiator on the braking efficiency of hydraulic retarder was rather simplified. Taking into account the design thermal capacity of the radiator in the coupled system of hydraulic retarder cooling system, main engine cooling system and engine oil cooling system, a detailed assessment of the thermal capacity of these units is required. Each of the mentioned devices is filled with a cooling medium with different thermo-physical properties. The entire system works correctly if none of the cooling media exceeds the permissible operating temperatures [24, 29].

The authors of this paper focused their attention on estimating the capacity of the main powertrain radiator, assuming that its efficiency can be reduced to about 15% due to local obstructions and contamination of the system. The aim of this paper is to select a main radiator working with multiple sub-systems and a hydraulic retarder. Achievement of the goal required the authors to develop a mathematical model, which was subject to simulation verification for two refrigerating media. The effects of ambient temperature, the volume of liquid in the retarder system, the size of the engine main radiator and the pump capacity in the cooling system on the temperature level of cooling media were analyzed. On the basis of the conducted tests and the obtained results, the direction of optimization of the adopted cooling system was indicated so that it complies with the requirements [7] under the assumed operating conditions. The obtained heat exchange coefficients were verified on a test stand, using the example of the 6C107[1] engine.

The presented calculations are only an example of how this model can be used for practical purposes, the versatility of which lies in the possibility of extending the powertrain cooling system with specific components.

1 The 6C107 engine is a design developed based on 400 Leyland

2. Characteristics of the operating conditions of the vehicle under analysis

For the analytical tests, a propulsion system of a road vehicle powered by a diesel engine type 6C107, with a capacity of 6.53 dm^3 and power of 92.5 kW/2600 rpm, produced in Poland, was adopted [33]. It was assumed that the vehicle is equipped with a Voith R120-4 hydraulic retarder. The data of this retarder were used to estimate the coefficients characterizing the heat exchange in this device [27, 31]. The structure of the cooling system to be tested is illustrated in Fig. 1.

The engine cooling system was protected by a bellows thermostat with the characteristics shown in Fig. 2 [34].

During the simulation tests it was assumed that the vehicle was driving on a horizontal road, emitting energy into the cooling system in the amount of q_{dost} = 29.5 kW. After 50 minutes, the vehicle starts to descend on a gradient of i = 7%. The descent lasts 12 minutes. The vehicle then continues to run on flat ground. The initial time of

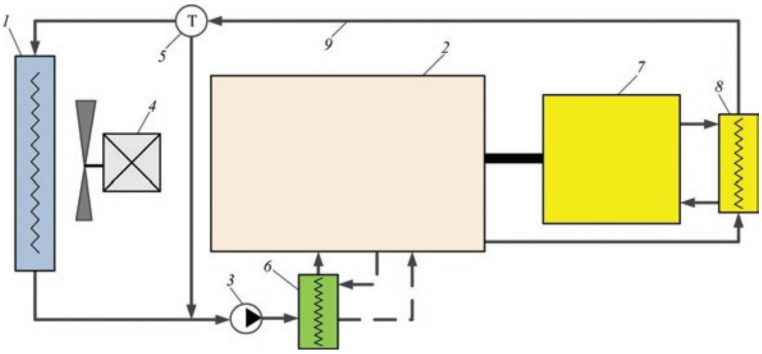
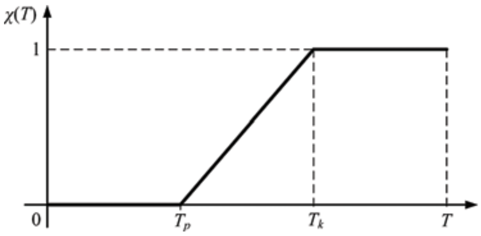


Fig. 1. Structure of the cooling system of a road vehicle powertrain equipped with a retarder: 1 – liquid-air cooler, 2 – engine, 3 – circulation pump of cooling liquid, 4 – fan of the cooler with electric drive, 5 – thermostat, 6 – engine oil cooler, 7 – retarder, 8 – retarder cooler, 9 – ducts



$$\chi(T) = \frac{T - T_p}{T_k - T_p} \quad \text{for} \quad T_p < T < T_k$$

where: 0 is for $T < T_p$ and 1 is for $T > T_k$

Fig. 2. Characteristics of the bellows thermostat for engine protection 6C107, where T_p – temperature at the beginning of thermostat opening, T_k – temperature of full thermostat opening

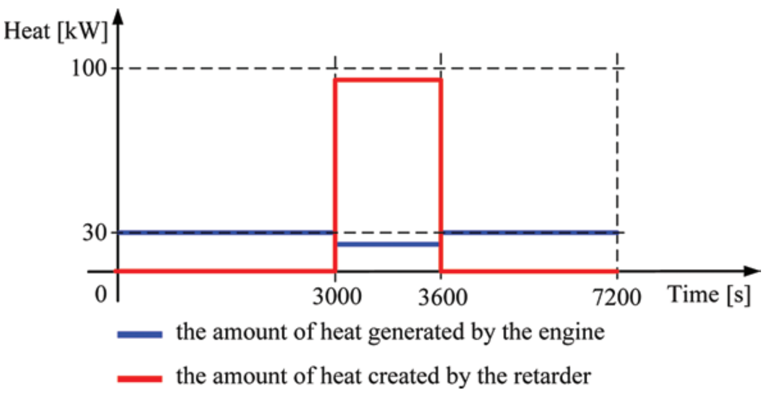


Fig. 3. Trend of thermal loads in the powertrain of the vehicle under analysis

50 minutes of constant speed driving causes the temperatures in the engine cooling system to be close to asymptotic levels under the assumed operating conditions [36, 39]. The trend of thermal loads in the cooling system of the powertrain was assumed as shown in Fig. 3.

3. Powertrain heat exchange model

A diagram of the powertrain model is shown in Figure 4.

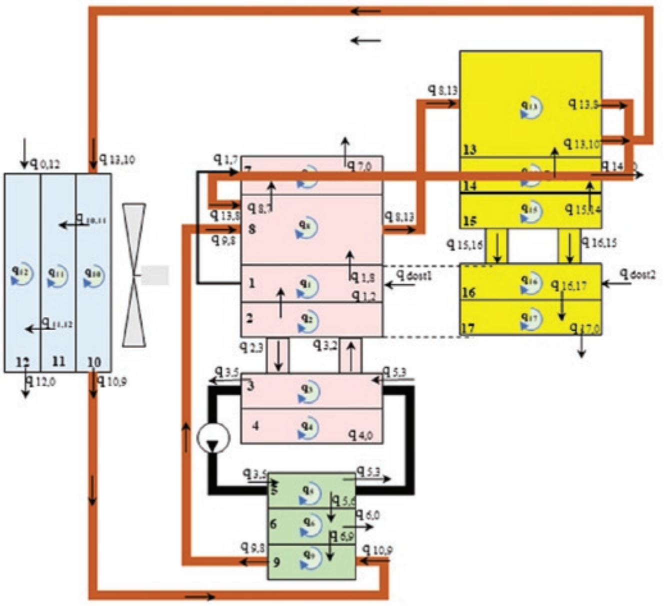


Fig. 4. Block diagram of engine with water-air cooler, engine oil cooler, retarder and thermostat; where: 0 – ambient environment, 1 – hot engine components, 2 – oil in the engine block, 3 – oil in the oil sump, 4 – oil sump, 5 – oil in the oil cooler, 6 – body of the oil cooler, 7 – engine body, 8 – water in the engine block, 9 – water in the oil cooler, 10 – water in the main cooler, 11 – body of the main cooler, 12 – water in the retarder cooler, 13 – water in the retarder cooler, 14 – body of the retarder cooler, 15 – retarder oil in the cooler, 16 – retarder oil in the retarder, 17 – retarder body

The thermal balance of the model is described by the following equations:

$$q_1 = q_{dost1} - q_{1,2} - q_{1,7} - q_{1,8} \quad (1)$$

$$q_2 = q_{1,2} + q_{3,2} - q_{2,3} \quad (2)$$

$$q_3 = q_{3,2} + q_{5,3} - q_{3,2} - q_{3,5} - q_{3,4} \quad (3)$$

$$q_4 = q_{3,4} - q_{4,0} \quad (4)$$

$$q_5 = q_{3,5} - q_{5,3} - q_{5,6} \quad (5)$$

$$q_6 = q_{5,6} - q_{6,0} - q_{6,9} \quad (6)$$

$$q_7 = q_{1,7} + q_{8,7} - q_{7,0} \quad (7)$$

$$q_8 = q_{1,8} + q_{9,8} + q_{13,8} - q_{8,7} - q_{8,13} \quad (8)$$

$$q_9 = q_{6,9} + q_{10,9} - q_{9,8} \quad (9)$$

$$q_{10} = q_{13,10} - q_{10,11} - q_{10,9} \quad (10)$$

$$q_{11} = q_{10,11} - q_{11,12} \quad (11)$$

$$q_{12} = q_{11,12} + q_{0,12} - q_{12,0} \quad (12)$$

$$q_{13} = q_{14,13} + q_{8,13} - q_{13,10} - q_{13,8} \quad (13)$$

$$q_{14} = q_{15,14} - q_{14,13} - q_{14,0} \quad (14)$$

$$q_{15} = q_{16,15} - q_{15,16} - q_{15,14} \quad (15)$$

$$q_{16} = q_{dost2} + q_{15,16} - q_{16,15} - q_{16,17} \quad (16)$$

$$q_{17} = q_{16,17} - q_{17,0} \quad (17)$$

Equations (1) to (17) in the differential notation take the following form:

$$c_1\dot{T}_1 = c_{dost1} - c_{1,2}(T_1 - T_2) - c_{1,7}(T_1 - T_7) - c_{1,8}(T_1 - T_8) \quad (18)$$

$$c_2\dot{T}_2 = c_{1,2}(T_1 - T_2) + c_{3,2}(T_3) - c_{2,3}(T_2) \quad (19)$$

$$c_3\dot{T}_3 = c_{2,3}(T_2) + c_{5,3}(T_5) - c_{3,2}(T_3) - c_{3,5}(T_3) - c_{3,4}(T_3 - T_4) \quad (20)$$

$$c_4\dot{T}_4 = c_{3,4}(T_3 - T_4) - c_{4,0}(T_4 - T_0) \quad (21)$$

$$c_5\dot{T}_5 = c_{3,5}(T_3) - c_{5,3}(T_3) - c_{5,6}(T_5 - T_6) \quad (22)$$

$$c_6\dot{T}_6 = c_{5,6}(T_5 - T_6) - c_{6,0}(T_6 - T_0) - c_{6,9}(T_6 - T_9) \quad (23)$$

$$c_7\dot{T}_7 = c_{8,7}(T_8 - T_7) + c_{1,7}(T_1 - T_7) - c_{7,0}(T_7 - T_0) \quad (24)$$

$$c_8\dot{T}_8 = c_{1,8}(T_1 - T_8) + \chi c_{9,8}(T_9) + (1-\chi)c_{13,8}(T_{13}) - c_{8,7}(T_8 - T_7) - c_{8,13}(T_8) \quad (25)$$

$$c_9\dot{T}_9 = c_{6,9}(T_6 - T_9) + \chi c_{10,9}(T_{10}) - \chi c_{9,8}(T_9) \quad (26)$$

$$c_{10}\dot{T}_{10} = \chi c_{13,10}(T_{13}) - \chi c_{10,9}(T_{10}) - c_{10,11}(T_{10} - T_{11}) \quad (27)$$

$$c_{11}\dot{T}_{11} = c_{10,11}(T_{10} - T_{11}) - c_{11,12}(T_{11} - T_{12}) \quad (28)$$

$$c_{12}\dot{T}_{12} = c_{11,12}(T_{11} - T_{12}) + c_{0,12}(T_0) - c_{12,0}(T_{12}) \quad (29)$$

$$c_{13}\dot{T}_{13} = c_{14,13}(T_{14} - T_{13}) + c_{8,13}(T_8) - \chi c_{13,10}(T_{13}) - (1-\chi)c_{13,8}(T_{13}) \quad (30)$$

$$c_{14}\dot{T}_{14} = c_{15,14}(T_{15} - T_{14}) - c_{14,13}(T_{14} - T_{13}) - c_{14,0}(T_{14} - T_0) \quad (31)$$

$$c_{15}\dot{T}_{15} = c_{16,15}(T_{16}) - c_{15,16}(T_{15}) - c_{15,14}(T_{15} - T_{14}) \quad (32)$$

$$c_{16}\dot{T}_{16} = c_{dost2} + c_{15,16}(T_{15}) - c_{16,15}(T_{16}) - c_{16,17}(T_{16} - T_{17}) \quad (33)$$

$$c_{17}\dot{T}_{17} = c_{16,17}(T_{16} - T_{17}) - c_{17,0}(T_{17} - T_0) \quad (34)$$

The values of c_i and c_{ij} factors were estimated on the basis of technical documentation of individual elements of the powertrain. The c_i

values describe the unit thermal capacities of the distinguished elements and equal the product of the mass of the element [kg] and its specific heat [kJ/kg·K]. The c_{ij} values describe the heat transfer conditions on the surface of the mentioned element and are equal to the product of the heat transfer surface of the element [m^2] and the heat transfer coefficient on the surface [kJ/m^2·K] (see Table 1).

The model of heating the engine itself without a retarder has been verified experimentally on the engine dynamometer in the Laboratory of Combustion Engines of the Wroclaw University of Technology and Science (Fig. 5). The results obtained from the measurement of the selected parameters describing the operation of the cooling system were used for determining the threshold conditions in the simulation model. Matlab-Simulink package was used for simulation tests (Fig. 6).

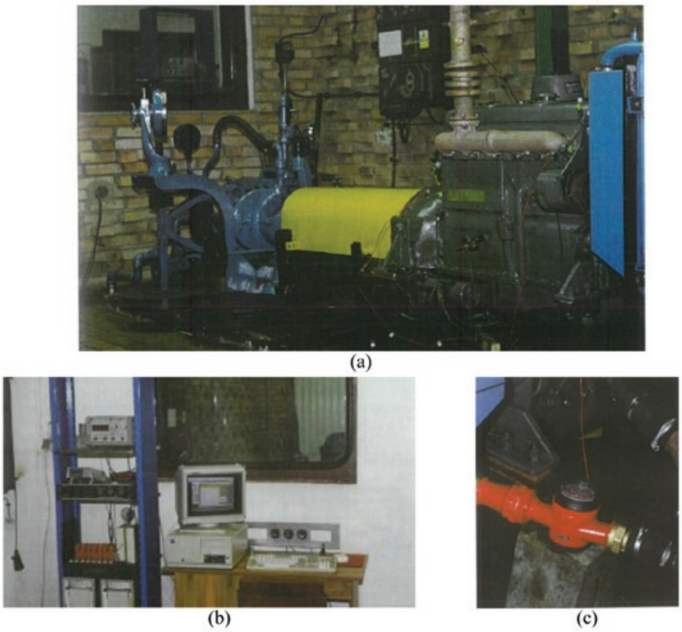


Fig. 5. Pictures of the test stand: (a) view of the 6C107 engine on the engine dynamometer of the Wroclaw University of Technology and Science, (b) computerized test bench on a dynamometer, with a temperature measurement module, (c) view of the engine block coolant flow meter

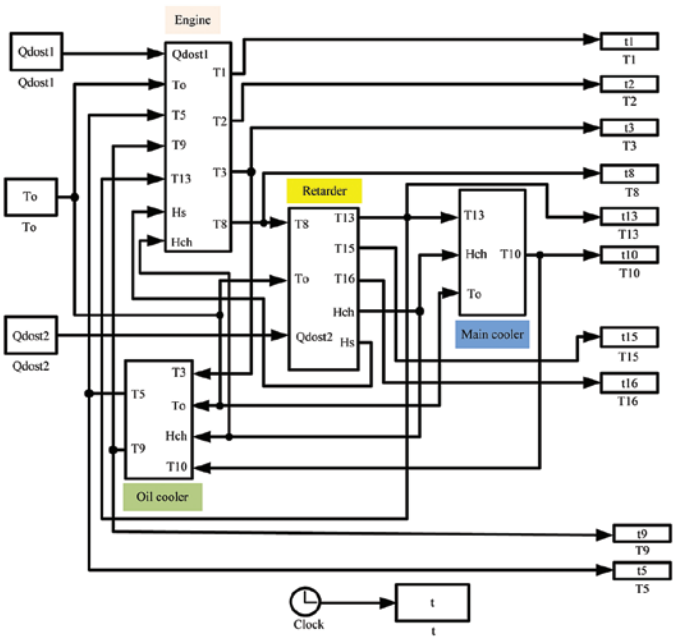


Fig. 6. Matlab-Simulink model

4. Results of simulation tests

Simulation tests were preceded by the development of a test plan. All components of the powertrain, i.e. temperatures from T_1 to T_{16}, distinguished in the temperature model, according to the legend to Fig. 6. A series of tests was carried out taking into account different ambient temperatures, heat exchange surfaces of the radiator, increasing the thermal capacity of the system. The tests used a cooling liquid based on:

a) water-filled engine cooling system,
b) glycol fluid-filled cooling system.

The continuous line indicates temperature fluctuations for the analysed subsystems – T_3, T_8 and T_{16}. The dotted line indicates threshold values in the analysed subsystems.

4.1. Simulation tests of a cooling system at different ambient temperatures

The cooling system tests were carried out at ambient temperatures (T_{ot}) in the range of 253K to 313K every 10K steps. Examples of results are shown in Fig. 7 and Fig. 8. According to them, the system can operate safely in the ambient temperature range up to T_{ot} = 306K (33ºC) (the highest ambient temperatures in Poland reach 313K (40ºC). The improvement of the situation can be achieved by enlarging e.g. the cooler of the engine block coolant.

4.2. Simulation tests of the cooling system for the case of an increase in the heat exchange area of the main cooler

The tests were carried out for three variants of cooler sizes: increased by 10%, by 30% and by 50%, at ambient temperature T_{ot} = 313K (40ºC). The results are presented in Fig. 9.

As can be seen from the presented results, in a glycol-filled system, the use of a cooler with capacity increased by 50% effectively protects all media against exceeding the permissible temperatures, especially the temperature of oil in the retarder. The use of a 50% larger main cooler led to a reduction in oil temperature of about 12K. However, this is not an optimal solution. It is worth investigating the impact that increasing the capacity of the pump in the main cooling system can have on overall capacity.

4.3. Simulation tests of the vehicle cooling system for the case of the cooler increased by 50% and increased coolant flow

The studies were carried out, as in previous cases, for water and glycol at increased flows by 50% and by 100% at ambient temperature T_{ot} = 313K (40ºC). The results are shown in Fig. 10.

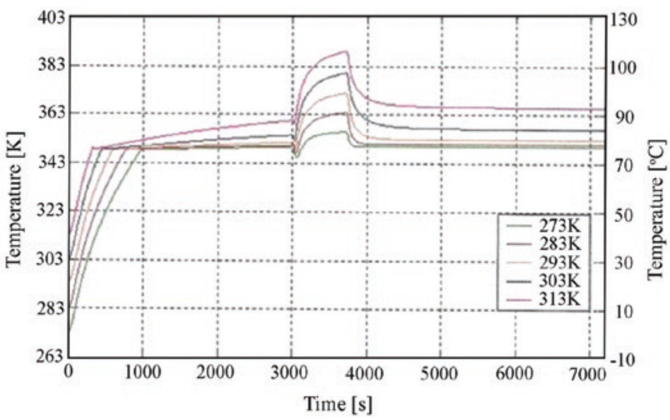


Fig. 7. Trend of cooling liquid temperature changes in engine block at different ambient temperatures (experiment)

Table 1. List of coefficients describing the engine thermal state

No.	A	B	C	D
1	c_i [J/K]	$c_{i,j}$ [W/K]	$c_{i,0}$ [J/K]	other parameters
2	c_1 = 87990	$c_{1,2}$ = 16	$c_{4,0}$ = 1.2	T_0 = 293.15 [K]
3	c_2 = 11750	$c_{1,7}$ = 200	$c_{6,0}$ = 10	q_{dost1} = 29500 [W]
4	c_3 = 1796	$c_{1,8}$ = 998	$c_{7,0}$ = 60	q_{dost2} = 91263 [W]
5	c_4 = 5560	$c_{2,3}$ = 20	$c_{12,0}$ = 13000	T_p = 348 [K]
6	c_5 = 6080	$c_{3,4}$ = 097	$c_{14,0}$ = 12.5	T_k = 358 [K]
7	c_6 = 1800	$c_{3,5}$ = 190	$c_{17,0}$ = 50	-
8	c_7 = 448148	$c_{5,6}$ = 72	-	-
9	c_8 = 20000	$c_{6,9}$ = 100	-	-
10	c_9 = 20950	$c_{8,7}$ = 236	-	-
11	c_{10} = 41900	$c_{9,8}$ = 5866	-	-
12	c_{11} = 19250	$c_{9,10}$ = 5866	-	-
13	c_{12} = 91	$c_{10,11}$ = 955.6	-	-
14	c_{13} = 20960	$c_{11,12}$ = 1240	-	-
15	c_{14} = 4500	$c_{13,18}$ = 5866	-	-
16	c_{15} = 6840	$c_{13,10}$ = 5866	-	-
17	c_{16} = 3040	$c_{14,13}$ = 583	-	-
18	c_{17} = 24750	$c_{15,14}$ = 456	-	-
19	-	$c_{16,15}$ = 5866	-	-
20	-	$c_{16,17}$ = 847	-	-

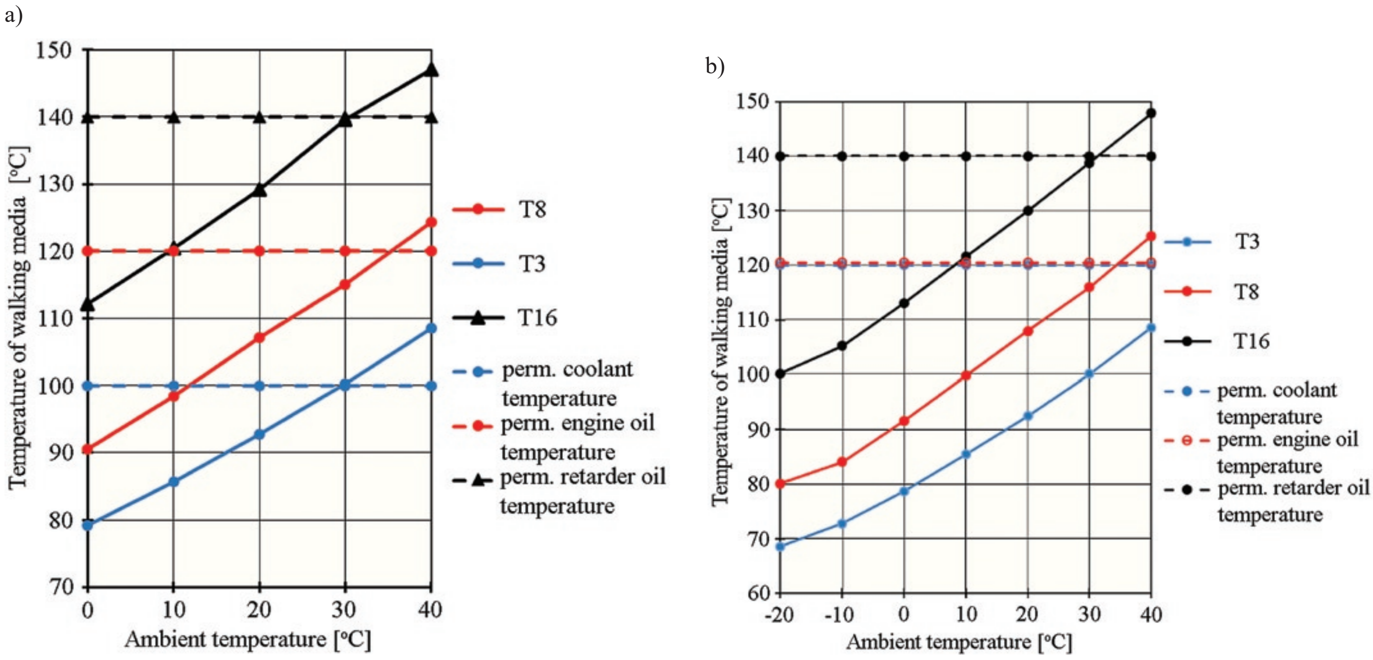


Fig. 8. Temperature of cooling media T_8, T_3, T_{16} versus ambient temperature T_{ot} (experiment): (a) for water, (b) for glycol

Research results show that an effective solution to the problem of temperature reduction in the retarder cooling system can be to increase the pump capacity by 50% or even only by 25% [15, 16].

4.4. Simulation tests of the vehicle's cooling system when an additional coolant tank is used to increase the thermal capacity of the system

The tests were carried out for tanks with a capacity of 20 and 40 dm^3 at an ambient temperature of T_{ot} = 313K (40ºC). The tank was located in the so-called large cooling circuit between the retarder cooler and the main engine cooler. It was assumed that the cooling system is filled with water. The results are shown in Fig. 11.

Equipping the cooling system with an additional coolant tank with a capacity of up to 40 dm^3 causes a slight decrease in system temperature. This solution is ineffective.

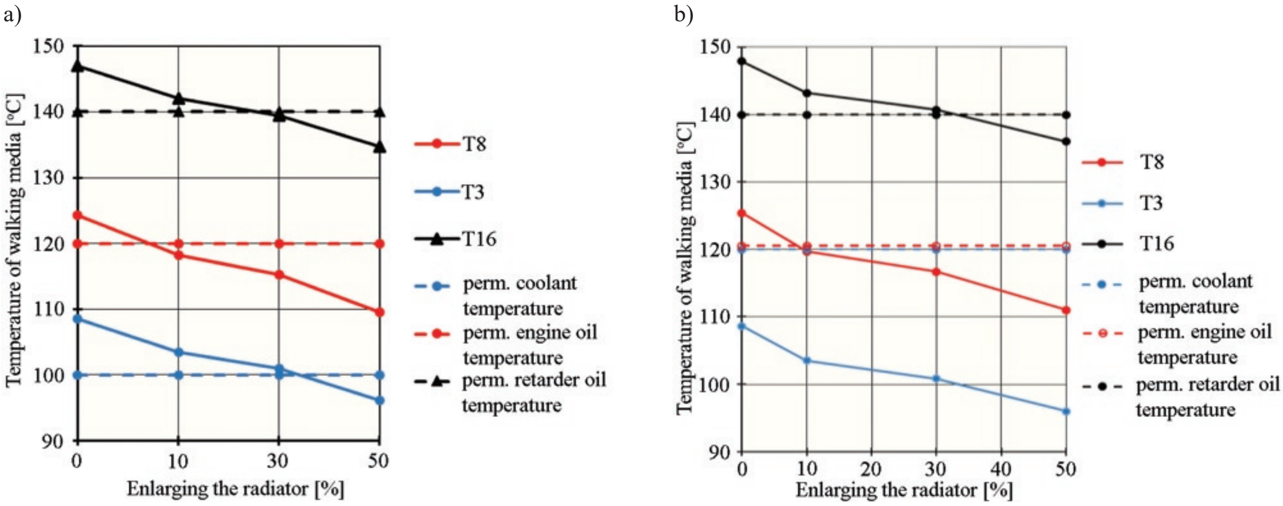


Fig. 9. Temperature of cooling media T_8, T_3, T_{16} versus the size of the radiator T_{ot} (simulation): (a) for water, (b) for glycol

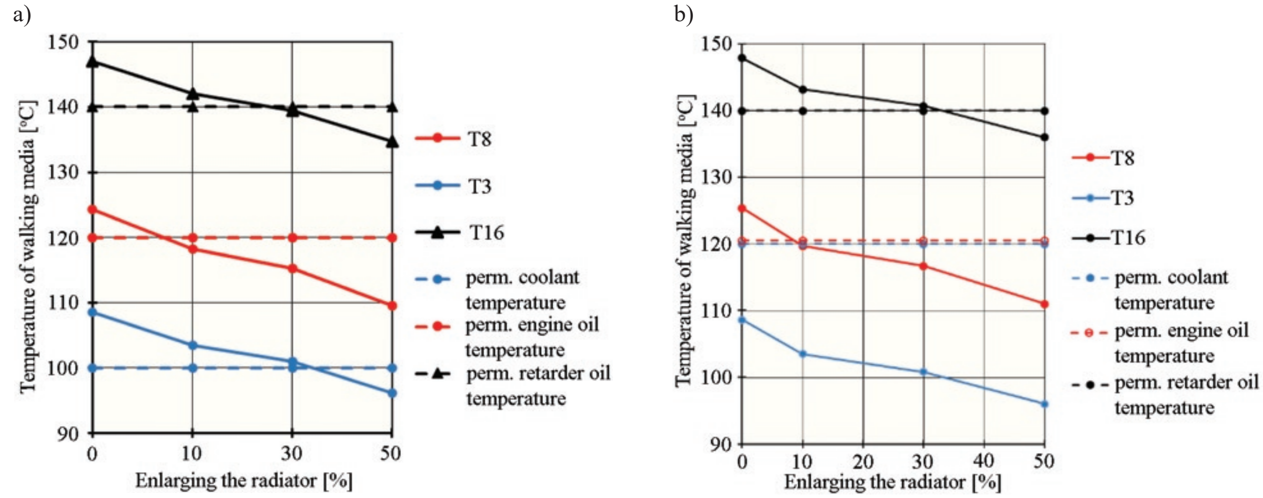


Fig. 10. Temperature of cooling media T_8, T_3, T_{16} versus coolant flow T_{ot} (simulation): (a) for water, (b) for glycol

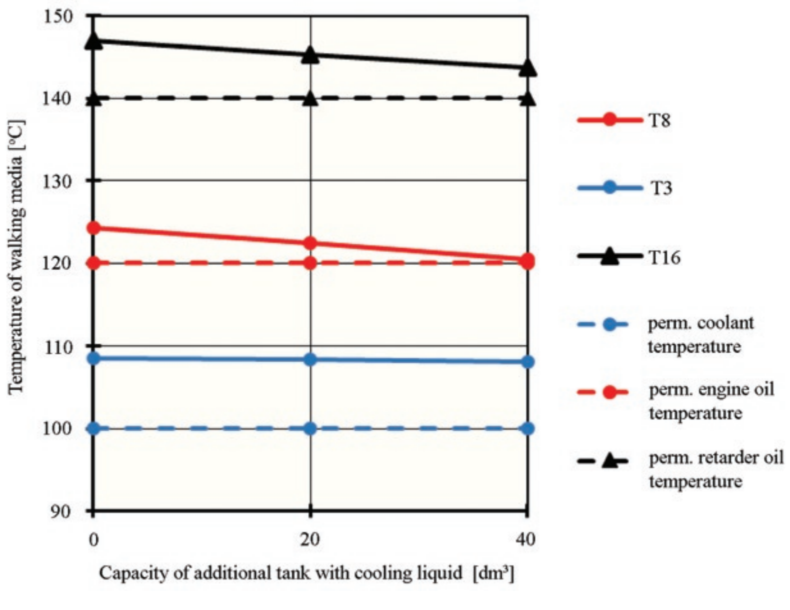


Fig. 11. The dependence of cooling media temperatures on the size of the additional coolant tank (simulation)

5. Conclusion

The aim of the study was to analyze the dynamics of heat exchange in a vehicle equipped with a hydraulic retarder. This device emits large energy flows into the cooling system of the braked vehicle. Therefore, the cooling system should take into account the retarder operation. The requirements in this respect are defined by the relevant EC regulations. On the basis of the analysis of the cooling system structure, a structural and computational model was built. The model of the cooling system is based on a set of 17 balance equations which, according to Newton's principle, were transformed into a set of 17 differential equations describing temperature changes of the distinguished elements versus time. The Matlab-Simulink package was used to solve the equation system. In the study, analyses of the influence of working conditions and constructional conditions on the behaviour of the system in the assumed scenario of its operation were carried out. The results of the work allowed to indicate the direction in which the design of the cooling system of the drive unit should be upgraded so that it could perform its function even in the most difficult working conditions.

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Krzysztof JAMROZIAK
Stanislaw KWASNIOWSKI
Mariusz KOSOBUDZKI
Pawel ZAJAC
Faculty of Mechanical Engineering,
Wroclaw University of Science and Technology
ul. Wyb. Wyspianskiego 27, 50-370 Wroclaw, Poland

E-mails: krzysztof.jamroziak@pwr.edu.pl, stanislaw.kwasniowski@pwr.edu.pl, mariusz.kosobudzki@pwr.edu.pl, pawel.zajac@pwr.edu.pl