vehicles

Article

Influence of Coolant Additives and Core Geometry of Fin-Tube Automobile Engine Radiators on the Enhancement of Cooling Process Efficiency

Marek Lipnický, Zuzana Brodnianská * and Pavel Beňo

Faculty of Technology, Technical University in Zvolen, Studentska 26, 960 01 Zvolen, Slovakia; marek.lipnicky@tuzvo.sk (M.L.); pavel.beno@tuzvo.sk (P.B.)
* Correspondence: zuzana.brodnianska@tuzvo.sk

Abstract: The paper deals with the research on the influence of the shapes of tubes and fins of automobile engine radiators and ethylene glycol coolants of type G12 on the cooling process. This involves cross-flow without mixing of coolant and air. The circular tubes with straight fins are compared with flat tubes with corrugated fins at identical external dimensions of the radiators. The new coolant is compared with the used coolant (10 years of usage) and further with a mixture of the used coolant and the additive (coolant enhancer). The goal is to reduce the heat dissipation time during the cooling process. Forced air convection is generated by three fan variants with diameters ϕ400 mm, ϕ345 mm, and a pair of fans ϕ345 mm and ϕ290 mm. The radiator core with flattened tubes and corrugated fins achieved lower outlet temperatures of 0.35 °C, 1.56 °C, and 2.43 °C compared to circular tubes and straight fins when using the ϕ400 mm diameter fan, the fan pair, and the ϕ345 mm diameter fan, respectively. The addition of the coolant enhancer to the used and new G12 coolant, depending on the fan variant, caused the outlet temperature to decrease in the range of 0.64 °C to 1.47 °C and 0.55 °C to 1.65 °C, respectively. The fan cover is also important for efficient cooling. Refilling of the coolant enhancer in the used coolant ensured that the heat transfer properties were recovered.

Keywords: coolant; engine radiator; automobile; fin; tube

check for updates

Citation: Lipnický, M.; Brodnianská, Z.; Beňo, P. Influence of Coolant Additives and Core Geometry of Fin-Tube Automobile Engine Radiators on the Enhancement of Cooling Process Efficiency. *Vehicles* **2024**, *6*, 2011–2030. https://doi.org/10.3390/vehicles6040099

Academic Editors: Davide Di Battista, Fabio Fatigati and Marco Di Bartolomeo

Received: 22 October 2024
Revised: 14 November 2024
Accepted: 22 November 2024
Published: 27 November 2024

1. Introduction

The increase in the performance parameters of internal combustion engines is associated with their continuous development and the necessity for more detailed research on automotive radiators and coolants. Also, stricter emission limits are the reason for the continuous improvement of engine cooling. At least one third of the thermal energy from the combustion process passes as waste heat through the cooling circuit. Not least, fuel saving and hence environmental protection also play an important role. If heat dissipation is achieved more efficiently, smaller radiators can be designed and manufactured, which is also an important parameter for automobile manufacturers.

Automobile radiators consist of a set of tubes through which the coolant flows and a set of fins. The fins can be of different shapes (rectangular, lamellar, perforated, corrugated) and help to transfer heat more efficiently on the air side. The shape and geometric arrangement of the tubes and fins also affect the overall heat performance of the radiator and the cooling efficiency [1]. The staggered arrangement of circular tubes is more efficient compared to tubes arranged in a line. When the tubes are arranged in a staggered arrangement, the cooling air impinges on the first row of tubes and also on the second row of tubes that are not in the cover, as is the case in the in-line arrangement. The Bougeard [2] found that the staggered tube arrangement increases the heat transfer coefficients on the second row of tubes. Plate fin-and-tube heat exchangers with oval tubes achieve higher thermal efficiency and lower pressure losses compared to circular tubes due to their improved aerodynamic shape. In the case of automobile radiators, they operate in a cross-flow arrangement where air flows perpendicular to the oval tubes through which the coolant flows [3].

The effect of fin height, thickness, and spacing on heat transfer has been investigated by Aytunc Erek et al. [4]. As the height of the fins increases, the heat transfer area is greater and therefore the overall heat transfer is higher. As the tube thickness is decreased, heat transfer is increased. The use of elliptical tubes leads to an increase in the cross section of air flow. Recirculating vortices formed behind the tube attenuate negative effect on heat transfer when the fin tube is placed at the downstream region. Thermal-hydraulic characteristics are also dependent on the spacing and number of rows of tubes [5]. For the number of tube rows 1 or 2, the heat transfer performance increased with decrease of fin pitch. However, with a higher number of rows of tubes, the effect of fin pitch on heat transfer performance was negligible.

Vertically arranged tubes with corrugated ribs were used in the research by the Balanna and Kishore [6]. Although the heat transfer surface area of corrugated fins is larger than that of conventional straight fins, they generate a higher pressure drop. In the experimental measurements, a constant coolant flow rate was maintained as the automobile speed was varied from 30 to 70 km/h. The impingement air flow rate was varied between 4.3 and 8 kg/s at an air temperature of 30 °C. The coolant flow rate through the tubes reached 1.3 kg/s at an inlet temperature of 90 °C. Mixtures of ethylene glycol and water in a 50:50 ratios, and admixtures of nanofluids were used as coolants. Colburn factor including heat transfer and pressure drop decreased on the air side with increasing mass flow rate. Increasing the speed of the automobile by 1.33%, heat transfer coefficient of air enhanced by 91.63%. Heat transfer and pressure losses on a plate heat exchanger with flat tubes and corrugated fins fixed on the tubes by means of a brazed joint were investigated in the paper [7]. The coolant flow rate was constant at 1.82 m^3/h, varying from Re = 350 to 1270. As the airflow increased, the heat transfer coefficient increased while the fanning factor decreased.

A further possibility for intensifying the heat transfer by forced convection is the use of tubes with a dimpled surface [8]. Coolant flows into the dimple and turbulent flow is promoted. The effect of fins on automobile radiator performance at higher atmospheric temperatures of 30.2 °C and 39.5 °C was discussed in the paper [9]. The use of fins resulted in a 25% decrease in the outlet temperature from the engine radiator. At higher atmospheric temperatures, a finned aluminium radiator is more preferable due to the higher heat dissipation at a relatively small radiator size.

Enhanced heat transfer can be achieved by optimising the radiator geometry and operating conditions. It is not only the geometry of the fins and tubes, but also the arrangement of the coolant flow, the modification of the material of the tubes and fins, and the modification of the type of coolant in different ratios [10]. The compact shape of the engine radiator leads to reduced airflow drag, reduced overall automobile weight, and lower fuel consumption. Heat transfer enhancement can also be achieved using nanofluids, thereby affecting a number of physical parameters. The cooling intensity of the tubes is also influenced by the inclination of the impinging air on the tubes. With increasing oblique angles, stronger vortices and more intense heat transfer are developed compared to cross flow [11].

A further widely expanded area of research is the coolants circulating in the cooling system. Their chemical and physical properties, dilution with water, or the use of nanofluids in different concentrations has been the subject of research for a long time. The use of nanofluids as a cooling medium has its advantages because of the slight increase in conductivity and the significant increase in viscosity and therefore heat transfer, but there are also some negatives in their use. There is a gradual deposition of particles in the cooling system. Patel et al. [12] found that if the volume fraction of CuO increases, heat transfer performance is improved but it is observed that the pumping power is also increased.

Efficient coolants and additives are used to increase the dissipation of excess heat in automobile engine cooling [13]. A commonly used automotive coolant is usually a mixture of water and ethylene glycol, diluted in various ratios to form an antifreeze that raises the boiling point and lowers the freezing point [14]. Antifreeze fluids based on ethylene glycol and propylene glycol are among the most widely used in the world [15]. The specific heat capacity of the cooling mixture decreases as the volume of ethylene glycol in the water increases. The freezing point temperature value decreases as the volume of ethylene glycol increases. It has been found that 40% ethylene glycol and 60% water is the most optimal coolant concentration [16,17]. However, a 50:50 ratios are often used. Increasing the volume of ethylene glycol in the cooling mixture leads to a decrease in thermal conductivity [18].

While current organic coolants achieve long-term efficiency, they do not provide the same corrosion protection compared to inorganic coolants. For this reason, coolant manufacturers fortify coolants with various additives. The use of specific additives depends on the engine block material (cast iron, aluminium), radiator components (metal, plastic). Commonly used additives are borate, nitrite, phosphate, silicate, sodium benzoate.

The presented paper is focused on the experimental investigation of the influence of coolant additives and radiator core geometry (tubes and fins) on cooling process efficiency. These aspects also play an important role in the cooling of automobile engines. The ethylene glycol coolants used in the research are commonly available and used to cool automobile engines. The goal is to compare new and used coolant (10 years of use) and then compare with a mixture of used coolant and an additional additive (coolant enhancer). We hypothesize that by adding an additive (coolant enhancer) to the used coolant, the cooling process can be made more efficient. Another goal is to investigate the influence of the geometry of the radiator tubes and fins on the efficiency of the cooling process. In addition, air cooling is provided by three different fan variants mounted on the radiator. The goal is to compare the effect of a single fan with different diameters (ϕ400 mm, ϕ345 mm) and an assembly of two fans with different diameters (ϕ345 mm and ϕ290 mm) on the efficiency of the cooling process.

2. Materials and Methods

The experimental investigation of the influence of the radiator core geometry and the effect of coolant additives (coolant enhancer) on the efficiency of the cooling process in the cooling circuit of the automobile engine was carried out on the laboratory experimental setup (Figure 1). The experimental cooling circuit consists of a Skoda engine with a mounted water pump and thermostatic valve. The thermostatic valve is connected to the engine radiator with a fan or set of fans, the heater, and the expansion tank by means of connecting pipes. The experimental cooling circuit is designed as in real operation, but without fuel combustion in the engine. The required coolant temperature is generated by the coil heater. The coolant circulates in the short or long cooling circuit depending on the actual temperature. The coolant volume in the coolant circuit is identical to the real volume in the automobile (nearly 6 litres).

Filling of the cooling circuit with coolant is carried out via the expansion tank (9). The water pump (4), driven via a V-belt by the electric motor, ensures the circulation of coolant firstly in a short cooling circuit at the temperature below 80 °C. The short cooling circuit consists of a water pump (4) which pumps the coolant through the cylinder block (5), cylinder head (6) to the thermostatic valve housing (8). From there the coolant flows into the heater (12) where it is heated by the coil (11) with a total output of 1500 W. The heater is connected to the water pump suction by a rubber hose. The coolant circulates in the short cooling circuit until the temperature increases to the operating temperature of 80 °C. Subsequently, the thermostatic valve (7) mounted in the thermostat housing (8) is actuated and the hot coolant flows into the radiator inlet pipe (1). If the hot coolant does not flow sufficiently fast in the radiator tubes, the rest of the coolant flows in the reverse pipe (c) to the expansion tank (9). The cooled coolant flows to the radiator outlet pipe (2) and back to the water pump inlet (4), where it circulates again in the short cooling circuit, where it is reheated to operating temperature. The cooling fans are activated when the coolant is not cooled by the engine radiator itself (3). The fan intensifies the heat dissipation from the radiator core to the ambient environment (tubes and fins). For sensing the coolant temperature in the inlet and outlet pipes of the radiator and in the thermostat, resistance temperature sensors (T_{in}, T_{out}, T_{therm}) are used. The coolant flow in the circuit is controlled by the flowmeter (14) and the temperature at the outlet of the heater by the resistance temperature sensor (T_h). All resistance temperature sensors were NTC type ZA 9040-FS (Murata Manufacturing, Kyoto, Japan) with a measuring range from −50 °C to 125 °C and an accuracy of ±0.01 °C. The sensors were fitted through the holes in the cooler inlet (1) and outlet (2) pipes, the thermostat housing (8) and the heater body (12). Flow meter (14) FVA 915 VTH (Ahlborn Mess- und Regelungstechnik GmbH, Holzkirchen, Germany) with a measuring range from 2 L/min to 40 L/min and an accuracy of ±1% was used to measure the coolant flow in the cooling circuit. All the sensors were connected to an Almemo 2590-4S data logger (Ahlborn Mess- und Regelungstechnik GmbH, Holzkirchen, Germany) (13).

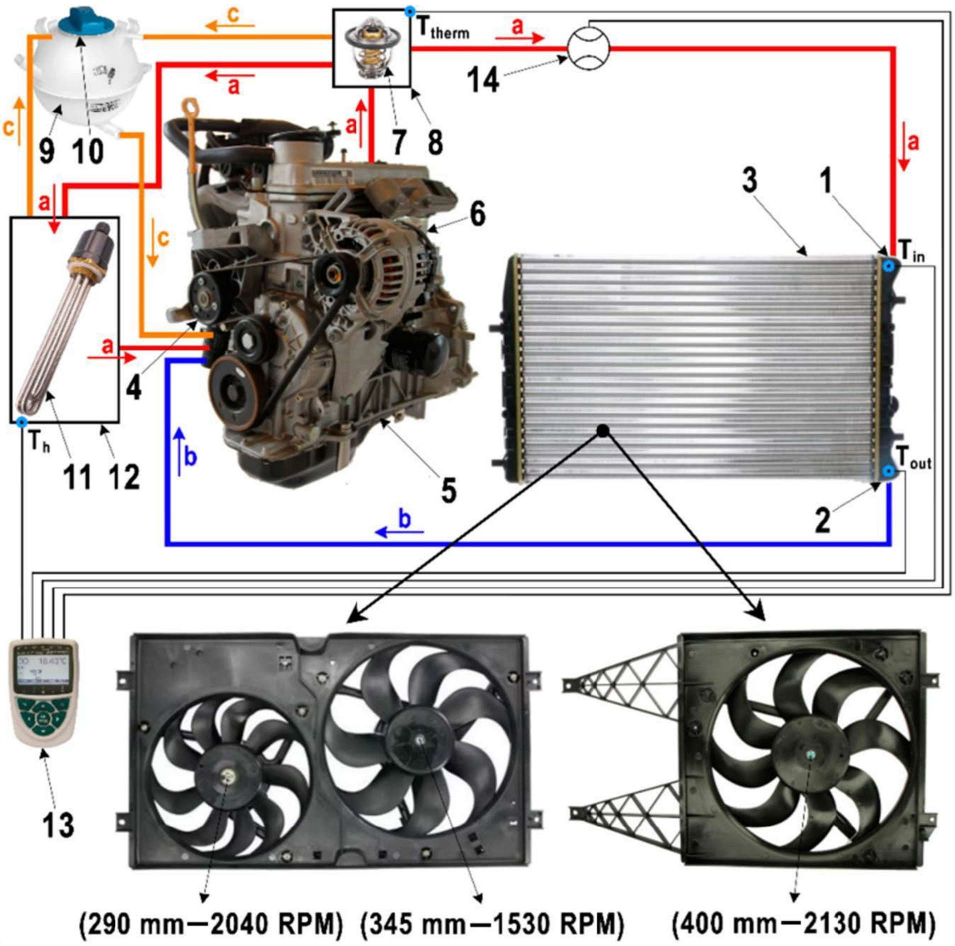


Figure 1. Experimental automobile engine cooling circuit setup. 1—radiator inlet pipe, 2—radiator outlet pipe, 3—engine radiator (liquid-air), 4—water pump, 5—cylinder block, 6—cylinder head, 7—thermostatic valve, 8—thermostat housing, 9—expansion tank, 10—overpressure plug, 11—heating coil (1500 W), 12—heater, 13—data logger, 14—flowmeter, a—hot coolant (above 80 °C), b—cooled coolant (below 80 °C), c—reverse pipe, coolant operating temperature (80 °C), T_{in}—coolant temperature at the radiator inlet, T_{out}—coolant temperature at the radiator outlet, T_h—temperature at the heater outlet, T_{therm}—temperature in the thermostat.

The main role of the coolant is to dissipate the excess heat produced by the engine's activity. If there is insufficient antifreeze in the coolant, the coolant solidifies at low temperatures and subsequently expands in the cooling channels, leading to mechanical damage to engine components. The most commonly used coolants in internal combustion engine cooling systems are G11, G12, G13 and their various modifications (e.g., G12+, G12++). These cooling mixtures are usually supplemented with additives that provide enhanced heat transfer properties and a consequent increase in cooling efficiency. In our case, two types of radiators with identical dimensions but different geometry of the radiator core are used for the research. The examined radiator heat exchange surfaces consist of circular tubes and straight fins (Figure 2a) and flat tubes and corrugated fins (Figure 2b). The shape and number of tubes and fins directly affect the time required to heat and cool the coolant. The Skoda engine (Skoda auto, a.s., Mladá Boleslav, Czech Republic) is equipped with an OHV valve train (camshaft mounted in the engine block). For this type of engine, the manufacturer recommends the use of a freeze-resistant compound corresponding to specification VWTL774D—type G12. For this reason, the original new G12 coolant (0 km mileage) and the original used G12 coolant (120,000 km mileage, in use for 10 years) were used in the research. The coolant consists of antifreeze and water in the ratio of 50:50 (up to −35 °C). The physical properties for the ethylene glycol coolant at temperature 80 °C are given in Table 1. The coolant additive added to the base coolant consists of the compounds: 2-amino-2-methylpropanol $(CH_3)_2C(NH_2)CH_2OH$, 2-(diethylamino)ethanol $(C_2H_5)_2NCH_2CH_2OH$, Potassium benzoate C_6H_5COOK, Benzotriazole $C_6H_5N_3$, Poly(ethylene glycol)bis(carboxymethyl) ether $HOOCCH_2(OCH_2CH_2)_nOCH_2COOH$. The coolant additive is light yellow in colour, and it is mixable with glycol concentrates to which it is added in liquid form. It is chemically stabilized under standard conditions and has the following physical properties: flash point: 115 °C, autoflammability 300 °C, relative density 1.11, vapour pressure 23 hPa (20 °C), and pH 9.3.

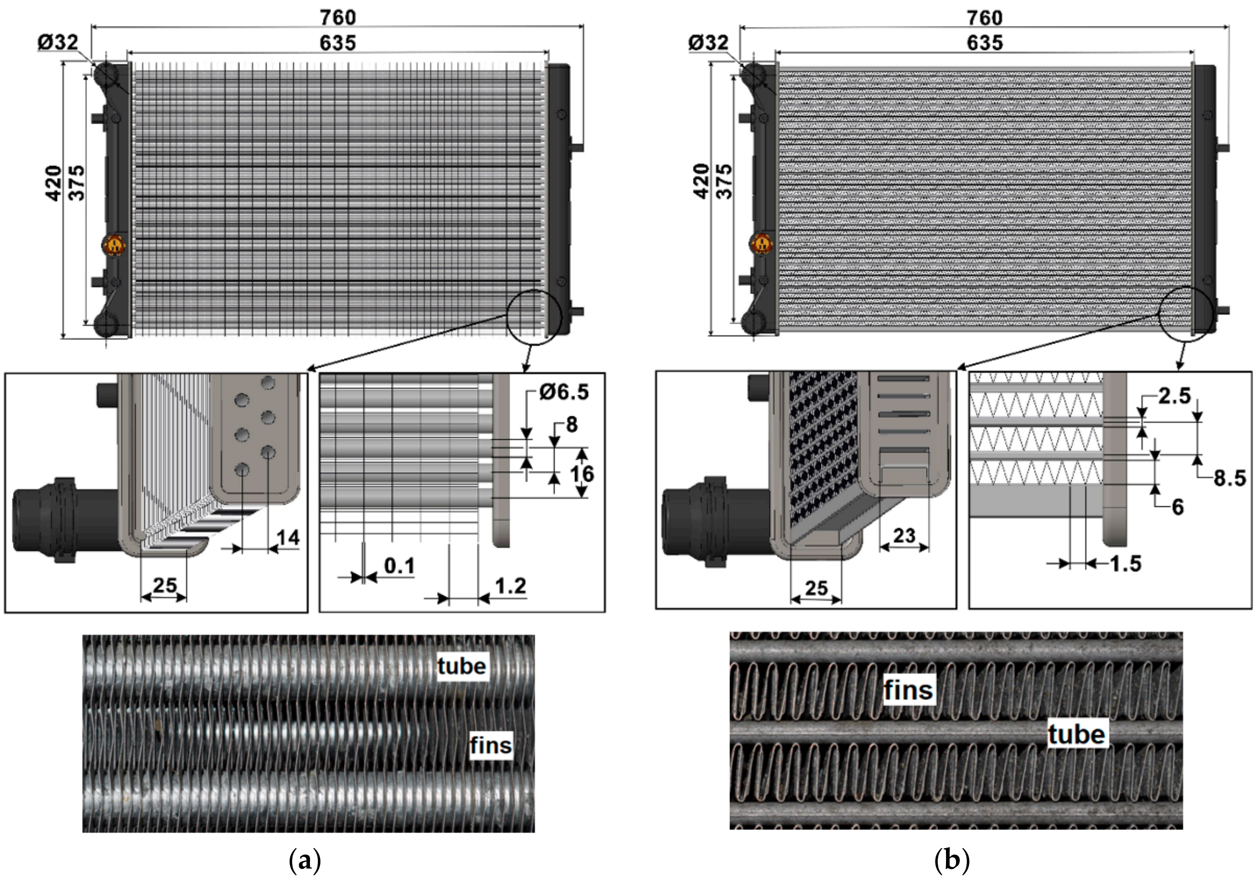


Figure 2. Dimensions of the investigated radiators: (**a**) circular tubes and straight fins; (**b**) flat tubes and corrugated fins.

Table 1. Physical properties for the ethylene glycol coolant at temperature 80 °C.

Thermal Conductivity (W/(m·K))	Density (kg/m^3)	Specific Heat (J/(kg.K))	Kinematic Viscosity (m^2/s)	Dynamic Viscosity (kg/(m.s))	Thermal Expansion Coefficient (K^{-1})	Thermal Diffusivity (m^2/s)
0.4657	1023.2	3422.5	8.4281×10^{-7}	8.6236×10^{-4}	2.8317×10^{-3}	1.3299×10^{-7}

The external cooling air bypass of the tube arrays (circular and flattened) was solved numerically on the software Creo Simulate 7.0.1.0. The selected section for the simulations corresponds to the tube bundle in the inlet of the radiator when the temperature of the outer surface of the tubes reached 353.15 K. Cooling air enters from the left side (v_{in} = 3 m/s, T_{in} = 293.15 K) as a velocity inlet boundary condition. The indicated velocity corresponds to the real airflow velocity when using a pair of fans in the experimental research. The boundary condition pressure outlet belongs to the right side (p_{out} = 101,325 Pa). The top and bottom sides belong to the Wall (adiabatic; q = 0) boundary condition and the tubes belong to the Wall boundary condition (T_w = 353.15 K).

The solution was fully converged when the residual of Temperature, Velocity, Pressure, Turbulent Kinetic Energy, and Turbulent Energy Dissipation Rate parameters reached the criterion 10^{-6}. The flow was solved by the k-epsilon model using the ideal gas law density. The mesh consisted of triangular and quadrilateral elements. The mesh independence test was carried out for the circular tube array until the variation in average velocity (Table 2).

Table 2. Mesh independence test for the circular tube array.

Grid	Total Nodes	Average Velocity (m/s)
Coarse	812,910	3.62
Medium	1,164,540	2.94
Fine	1,573,182	2.69
Ultra-Fine	2,425,092	2.64

The average velocity of the cooling air passing through the investigated area was obtained experimentally using the FVAD 15-H digital vane anemometer (Ahlborn Mess- und Regelungstechnik GmbH, Holzkirchen, Germany) with a measurement range from 0.3 to 40 m/s, resolution 0.01 m/s. The differences between the average velocity obtained experimentally and by simulation were in the range of 1.5% to 6.4% depending on the number of nodes of the mesh.

The geometry and basic dimensions of the investigated radiators are shown in Figure 2. The total height, length, and width of the radiators are identical; however, the geometric shape of the tubes and fins is varied. The diameter of the radiator inlet and outlet pipes is identical ϕ32 mm.

The radiator in Figure 2a consists of a two-row array of circular tubes arranged in a staggered configuration with an outer diameter of 6.5 mm in the number of tube units of 25 in the first row and 26 in the second row. The radiator in Figure 2b consists of straight tubes with a length of 23 mm and a height of 2.5 mm in a number of tube units of 44. The number of rows of fins is 46. The thickness of the fins is identical for both examined radiators, measuring 0.1 mm. Based on the inner dimensions of the tubes, a coolant volume of 9.2×10^{-4} m^3 and 12.9×10^{-4} m^3 was calculated for a cooler with circular tubes and flattened tubes, respectively.

3. Results and Discussion

The results of the experimental measurements are structured to research the influence of the radiator core geometry and the effect of coolant additives on the heating and cooling process in the automobile cooling circuit.

3.1. Effect of Engine Radiator Core Geometry on Cooling Efficiency

The heating and cooling process of the coolant inside the cooling circuit of the automobile engine is influenced by the integrated heat exchanger (radiator), especially the radiator core. Radiator cores can have different geometrical parameters, but in general, as the fin density increases, the length of the cooling process is reduced. Also, due to the influence of ambient impurities, high density finning may not be effective in the long time. The use of circular or flat tubes leads to different heat transfer processes as they are bypassed by the air from the fan. Also, the use of straight or corrugated fins affects the rate of heat dissipation from the radiator to the ambient environment. The results of the experimental measurements are discussed in terms of the process of heating and cooling of the coolant when the geometry of the radiator core is varied.

3.1.1. Coolant Heating Process When Changing the Radiator Core Geometry

The observed variations of the initial coolant temperatures (simulation of the combustion process) in the short and long cooling circuit from the ambient temperature T_a = 20 °C to the operating temperature T_o = 80 °C are shown in Figure 3. The radiator with circular tubes and straight fins achieved a time of 27 min for heating the coolant in a short circuit to an operating temperature of 80 °C (T_{therm} = 80.37 °C). The radiator with flat tubes and corrugated fins achieved a heating time of 30 s longer (27 min and 30 s, T_{therm} = 80.57 °C). At the mentioned time intervals, the initial release of hot coolant into the cores of the radiators occurred. The difference in heating time when using a radiator with flattened tubes is due to the slight increase in coolant volume (~0.15 L) contained in this type of radiator. The waveforms of the initial coolant temperatures in the long cooling circuit remained constant throughout the heating process without change. This is caused by the low temperature of the coolant in the short cooling circuit below the operating temperature (80 °C), which is insufficient for the thermostatic valve to open. It can be concluded that the different geometrical parameters of the investigated engine radiator cores do not cause a change in the time length of the coolant heating up to the operating temperature.

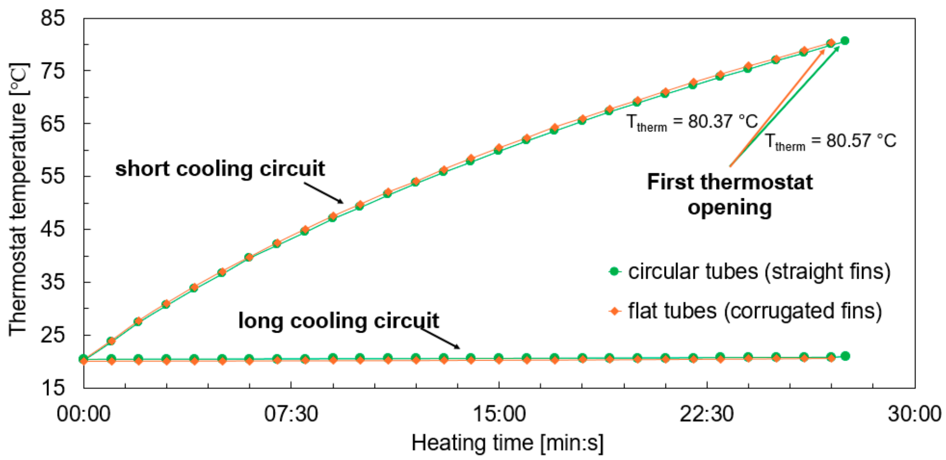


Figure 3. Initial thermostat temperature waveforms while heating the coolant in the short and long coolant circuits when changing the geometry of the radiator core.

Hot coolant, heated above the operating temperature, flows from the short cooling circuit through the open thermostatic valve under pressure, and simultaneously through the water pump blades into the radiator core. This is shown in Figure 4, where the temperature waveforms at the inlet of the radiators are recorded. At a time of 27 min and 30 s, a significant temperature increase can be observed for both types of radiators, indicating that hot coolant has been released into the long cooling circuit. When the hot coolant is released into the radiator at a time of 28 min and 30 s, a gradual closing of the long circuit by the thermostatic valve can be observed. This is due to the cold coolant being pumped out by the water pump to the expansion element of the thermostat. At the same time, a significant difference in inlet temperatures can be observed at this time when comparing the radiator with circular and flattened tubes. The radiator equipped with circular tubes at this time reached a coolant inlet temperature of T_{in} = 61.86 °C, while the radiator with flattened tubes reached a higher temperature of T_{in} = 68.82 °C. This significant difference (ΔT = 6.96 °C) is due to the higher volume of the flattened tubes, where more coolant flows through them compared to a radiator with circular tubes in the same time to the point of thermostat re-closure. Subsequently, a gradual stabilization of coolant temperatures between the short and long cooling circuits can be observed within a time of 44 min (radiator with circular tubes T_{in} = 80.01 °C) and a time of 45 min and 30 s (radiator with flattened tubes T_{in} = 80.04 °C). At this point, the radiator cores have "overheated" (equalization of temperatures between the radiator inlet and outlet), where heat is not sufficiently dissipated by free convection and thus the radiator fan is on. The start of the radiator fan (forced heat dissipation from the radiator core) occurred at 44 min and 45 min and 30 s for the circular-tube radiator and the flattened-tube radiator, respectively. This difference is caused by faster heating of the radiator core due to lower coolant volume (radiator with circular tubes and straight fins). It can be stated that the radiator with circular tubes reached the "overheating" of the radiator core 1 min and 30 s earlier than the radiator with flattened tubes. Faster "overheating" of the radiator cores in real operation is an undesirable phenomenon. The individual engine components are more exposed to thermal stress and the radiator fan has to be run more frequently, which has a negative impact on the optimum operation of the engine.

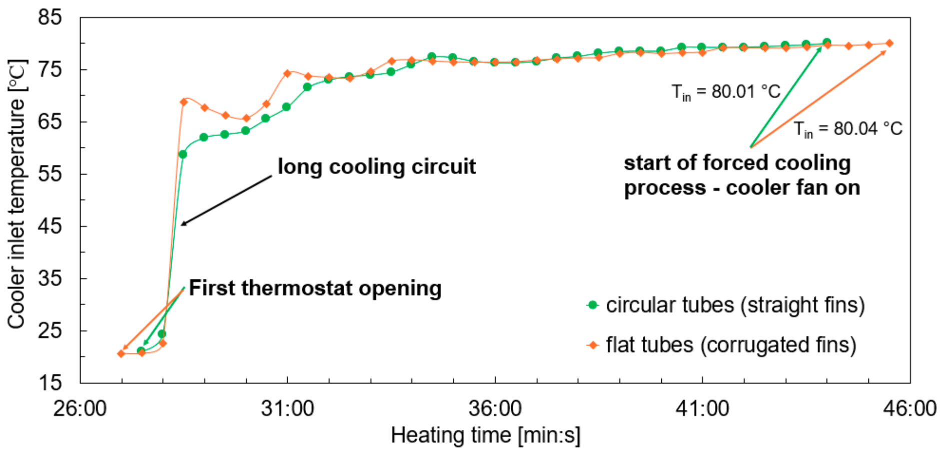


Figure 4. Radiator inlet temperature waveforms in the long cooling circuit influenced by thermostat opening when changing the geometry of the radiator core.

While the experimental setup was operating, the engine speed was set at idle speed, i.e., 900 RPM, to simulate stationary or low driving speeds. This operating mode represents one of the major thermal stresses on the engine, especially in the summer months. At this point, there is no sufficient air supply to dissipate the heat from the overheated engine radiator core. Overheating of the radiator core is caused by the hot coolant flowing in from the short cooling circuit, and at the same time the radiator core is also heated from the engine compartment. The accumulated heat blocks the heat from the radiator core to the ambient and engine compartment by free convection. The radiator outlet temperature waveforms during the hot coolant flow in the radiator core are shown in Figure 5. The coolant circulating from the short to long cooling circuit reached the operating temperature in the thermostat T_{therm} = 80 °C. For the time period of 27 min to 33 min, a relatively steady state coolant outlet temperature waveform can be observed for both radiator core geometries investigated. The coolant temperature at this point was still identical to the ambient temperature T_a = 20 °C. Cold coolant was forced out of the radiator core and into the water pump suction by the flow of hot coolant from the long cooling circuit.

The core heating process of the radiator can be observed in the time period from 33 min (T_{out} = 22.20 °C) to 44 min (T_{out} = 72.61 °C) for the radiator with circular tubes, and in the time period from 33 min (T_{out} = 22.20 °C) to 45 min and 30 s (T_{out} = 72.61 °C) for the radiator with flattened tubes. It can be stated that the radiator with circular tubes reached the core heat of the radiator earlier by 1 min and 30 s, which does not represent a significant advantage in terms of operating efficiency compared to the radiator with flattened tubes. Simultaneously, the geometry of the radiator core with flattened tubes ensured that the cooling circuit was maintained at a lower temperature for a longer period of time. This is also due to the larger heat exchange surface through which more of the heat has been dissipated by free convection.

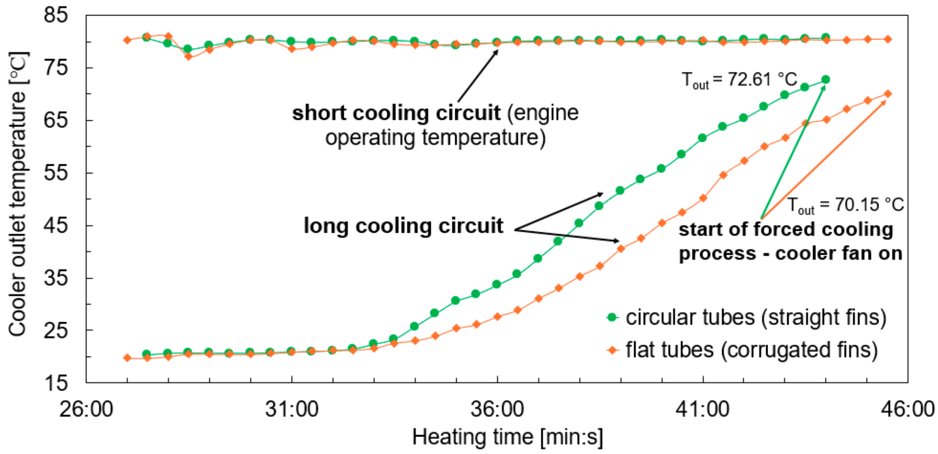
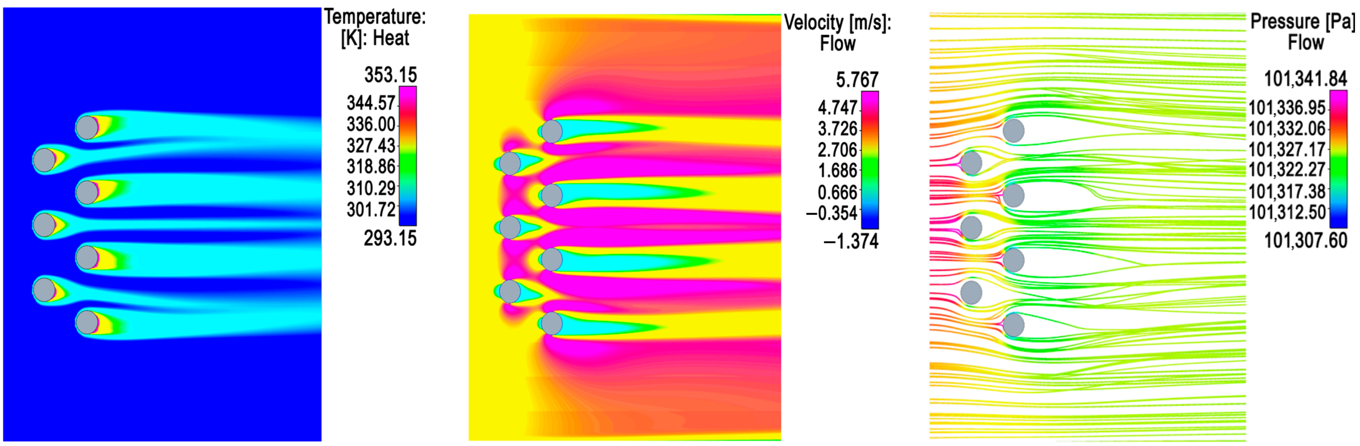


Figure 5. Radiator outlet temperature waveforms influenced by thermostat opening when changing the geometry of the radiator core.

CFD simulations of the temperature and velocity fields in the vicinity of the heated bundle of circular and flattened tubes in forced air convection with a velocity of 3 m/s are shown in Figures 6 and 7. The indicated velocity corresponds to the real airflow velocity when using a pair of fans. The selected section for the simulations corresponds to the tube bundle in the inlet of the radiator when the temperature of the outer surface of the tubes reached 80 °C.



(**a**) (**b**) (**c**)

Figure 6. CFD simulations of cooling air circulation around heated circular tubes: (**a**) temperature fields; (**b**) velocity fields; (**c**) pressure distribution.

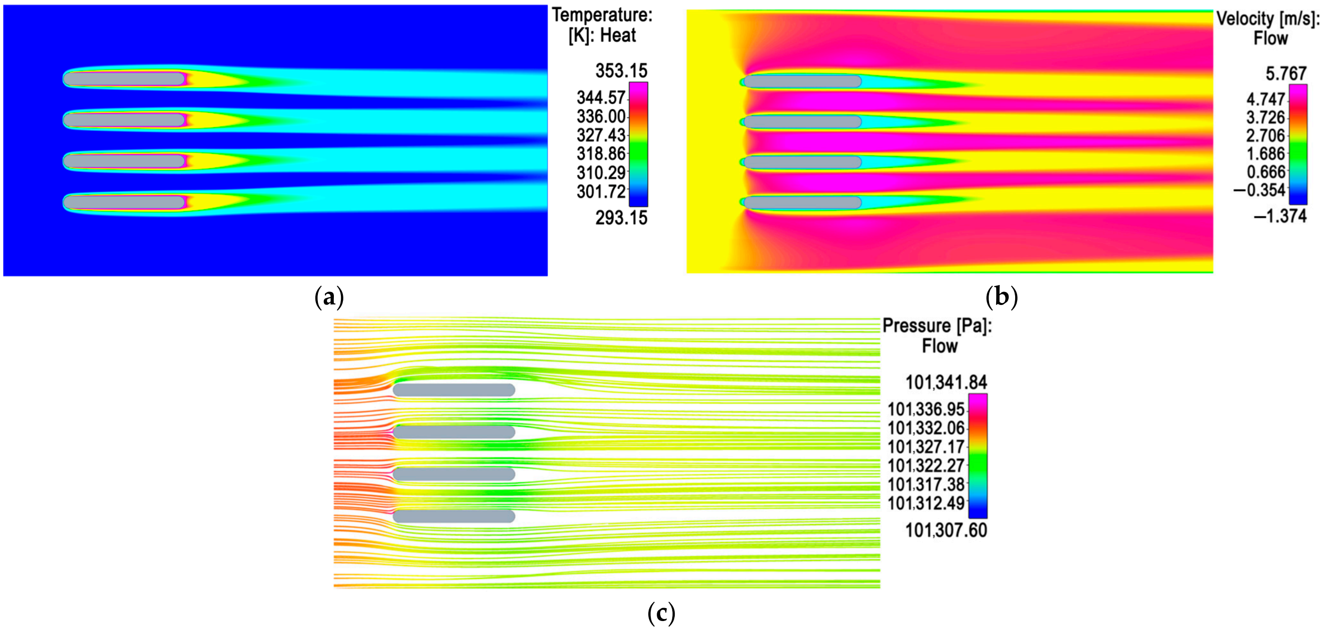


Figure 7. CFD simulations of cooling air circulation around heated flattened tubes: (**a**) temperature fields; (**b**) velocity fields; (**c**) pressure distribution.

From the temperature fields in the vicinity of the circular and flattened tubes, the formation of the temperature boundary layer around the circumference of the tubes can be observed, the thickness of which depends on the shape of the tubes (Figures 6a and 7a) [19]. The second row of circular tubes is influenced by the first row, but the staggered arrangement of the circular tubes helps the cooling air to directly impinge on all the tubes. The flattened tubes, by their shape, achieve a higher thickness of the thermal boundary layer, which increases along the bottom and top edges of the tubes and thus the heat transfer degrades with the length of the flattened tubes (Figure 7a). However, the internal volume of flattened tubes is higher compared to circular tubes, which compensates for the mentioned disadvantage. The distribution of pressures in the vicinity of the circular tubes indicates a gradual decrease in pressure with increasing number of tube rows (Figure 6c). In the narrowed spaces between the tubes, the pressure drops as the velocity increases.

From the velocity fields in the vicinity of the circular tubes, an increase in the cooling air velocity can be observed due to the narrowed cross section between the tubes in the first row, thus forcing the air to reach the second row of tubes more efficiently (Figure 6b). Simultaneously, the second row of circular tubes influences the suction of air from the first row of tubes, thus deflecting the flow off the flow axis. The velocity between the flattened tubes also increases due to the narrow space between them, which contributes to the heat transfer from the tubes to the ambient air (Figure 7b). As the velocity of the flowing air in the narrowed regions between the tubes increases, the pressure decreases, and the decrease is also observable in the length between the flattened tubes (Figure 7c).

3.1.2. The Process of Cooling the Coolant Below Operating Temperature When Changing the Radiator Geometry

At the moment when the heat exchange surface of the radiator overheats due to insufficient flow of ambient air, a fan is started to operate, which removes the excess heat. In the cooling circuits of real automobiles, the forced cooling process of the heat exchange surface of the engine radiator takes approximately 30 s. However, the cooling process can take longer under different conditions, but usually does not exceed 1 min. In the experimental measurements, the radiator fan was started to operate when the coolant temperature in the radiator inlet pipe was above 80 °C and that in the radiator outlet pipe was 75 °C. At this point, there was a gradual equalization of the coolant temperatures at the radiator inlet and outlet, which means that the engine cooling circuit was beginning to overheat due to the insufficient air flow through the heat exchange surface by free convection.

The G12 coolant temperatures at the inlet and outlet of the radiator with circular tubes and flat fins when cooled by three fans with different diameters (ϕ345 mm, ϕ400 mm, ϕ345 mm, and ϕ290 mm) are shown in Figure 8. Within 1 min of starting the pair of fans (ϕ345 mm and ϕ290 mm), the lowest coolant temperature at the inlet of the radiator was recorded to be T_{in} = 78.01 °C. When cooled by a single fan (ϕ345 mm), the inlet temperature of the coolant reached T_{in} = 78.19 °C. The highest temperature was recorded when cooling by a ϕ400 mm diameter fan (T_{in} = 79.29 °C). This is because the fan with the highest diameter ϕ400 mm located in the left part of the radiator core is not mounted on a continuous full cover, which would also serve as a "hot air suction" from the overheated radiator core, as is the case for the fans with diameters ϕ345 mm and ϕ290 mm (Figure 1).

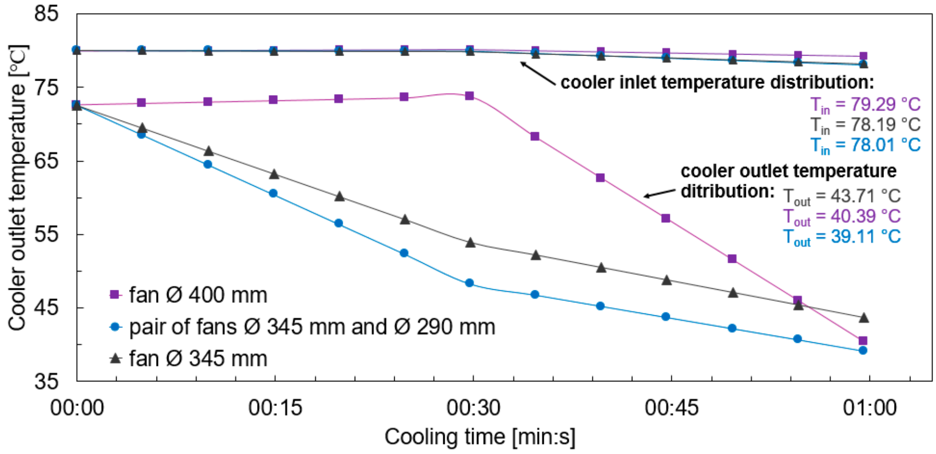


Figure 8. Radiator outlet temperature waveforms at the radiator inlet and outlet—forced cooling by fans (radiator with circular tubes and straight fins).

A significant decrease in G12 coolant temperatures was recorded at the radiator outlet. After 30 s, the temperature decreased the fastest in the process of cooling by a pair of fans (T_{out} = 48.25 °C). A single fan with a diameter of 345 mm was able to cool the coolant to a temperature of T_{out} = 53.93 °C in the same time. When cooled by a single fan (ϕ400 mm), the temperature increased even more during the first 30 s (T_{out} = 73.73 °C). This is caused by undirected suction of hot air (delayed cooling start-up) from the radiator core by the fan. The most significant decrease in the coolant outlet temperature within one minute of fan start-up was recorded by a pair of fans T_{out} = 39.11 °C. Subsequently, it was cooling by a fan with a diameter of ϕ400 mm (T_{out} = 40.39 °C) and by a fan with a diameter of ϕ345 mm (T_{out} = 39.11 °C).

The fastest cooling of the G12 coolant below operating temperature is best achieved by a pair of fans (ϕ345 mm and ϕ290 mm) mounted on a full radiator cover with circular tubes and straight fins. The cover is used to mount the fans, helping to suck hot air from the heat exchange surface of the radiator while separating this surface from hot engine components that can cause a rise in inlet and outlet coolant temperatures. This effect can be observed specifically for a radiator with a diameter ϕ400 mm without a full cover.

The G12 coolant temperatures at the inlet and outlet of the radiator with flat tubes and corrugated fins for the cooling process by three fans with different diameters are shown in Figure 9. Within one minute of starting the single fan (ϕ345 mm), the lowest inlet coolant temperature was recorded with a value of T_{in} = 77.56 °C. When cooled by a ϕ400 mm diameter fan, the inlet temperature of the coolant reached T_{in} = 78.78 °C, while the highest inlet temperature of T_{in} = 79.89 °C was recorded when cooled by a pair of fans. The increased inlet coolant temperature at the end of the cooling process by the fan pair is due to the geometry and design of the radiator core. The fan pair is positioned across the entire width of the radiator core. However, there are still uncovered areas at the hot coolant inlet (top 3 rows of tubes) from the short cooling circuit where the blower exhaust air from the fan is drawn.

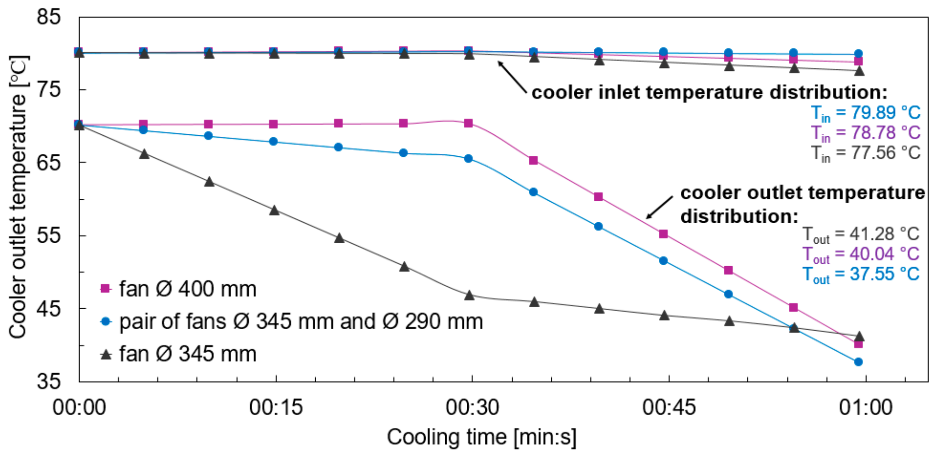


Figure 9. Radiator outlet temperature waveforms at the radiator inlet and outlet—forced cooling by fans (radiator with flat tubes and corrugated fins).

In the other case, the second radiator under investigation has flat tubes and corrugated fins positioned in the horizontal plane and this prevents hot air from being drawn in by the fans from the upper rows of tubes. When cooling the radiator core with circular tubes and straight fins positioned in the vertical plane, hot air is also drawn in from the top inlet tubes as the fan blades rotate. The most significant decrease in coolant outlet temperature within one minute of fan start-up was again recorded by a pair of fans (T_{out} = 37.55 °C). The coolant was able to decrease even more by 1.56 °C in this case compared to the radiator with circular tubes and straight fins (T_{out} = 39.11 °C). When cooled by a ϕ400 mm diameter fan, the coolant outlet temperature reached T_{out} = 40.04 °C, representing a temperature decrease of 0.35 °C compared with circular tubes (T_{out} = 40.39 °C). The lowest temperature decrease was observed when cooling the coolant with a fan with a diameter of ϕ345 mm (T_{out} = 41.28 °C). In this case, there was a 2.43 °C decrease in coolant temperature compared to the circular tubes (T_{out} = 43.71 °C). It can be concluded from the measured results that the radiator core with flattened tubes and corrugated fins provides a significant decrease in coolant temperature compared to a radiator with circular tubes and straight fins, making it more efficient in the cooling process.

3.2. Effect of Coolant Additives on Cooling Efficiency

The other experimental measurements were focused on the monitoring of the heating and cooling time of the coolant in order to determine the influence of coolant additives (coolant enhancers) on the cooling process. The radiator with circular tubes and straight fins used for this study (Figure 2a). Four coolant samples were tested, two of which were additionally additive to investigate the thermal variations during heating and cooling of a new coolant (0 km) and a used coolant (120,000 km) of type G12. The mentioned additive (coolant enhancer) can generally be added to new or used coolant. The coolant cooling process was performed with three variations of fans, as was done in Section 3.1. The cooling process was performed with the single fan (ϕ400 mm, 2130 RPM), the pair of fans simultaneously (ϕ345 mm, 1530 RPM and ϕ290 mm, 2040 RPM), and one of the pair of fans (ϕ345 mm, 1530 RPM).

The operating temperature of the coolant in real operation ranges from 80 °C to 90 °C and is determined by the thermostatic valve used. A thermostatic valve with a temperature value of 80 °C was fitted in the thermostat housing of the experimental setup, when its initial gradual opening occurred. When the operating temperature was reached in the short cooling circuit, the cool liquid was forced out of the radiator outlet by the hot liquid flowing through the open thermostatic valve and the inlet pipe to the radiator. Subsequently, this hot liquid was cooled by the fan in the radiator via the heat exchange surface. For the operating of the internal combustion engine, it is necessary to cool the hot fluid slightly below the operating temperature as quickly as possible. However, the operating temperature of the engine is not only influenced by the combustion process but also by the ambient air temperature T_a.

3.2.1. Coolant Heating Process with and Without Additives

When operating an internal combustion engine, it is important that it does not operate for long periods outside the operating temperature range due to inefficient operation and possible damage due to high temperatures. However, the same can also occur if the engine runs in subcooled mode below the operating temperature for a long period of time. Reaching the operating temperature of the engine earlier is just as important a parameter as cooling it down from temperatures above the operating temperature. This fact led to the investigation of the coolant heating process also in our series of the experimental measurements (Figures 10–12).

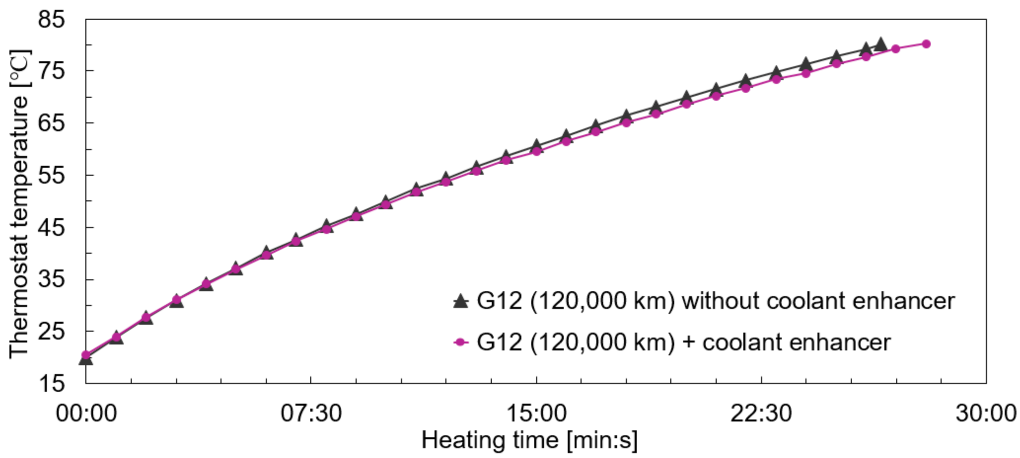


Figure 10. Thermostat temperature waveforms while the coolant in the short cooling circuit is heated to operating temperature for G12 (mileage 120,000 km) and G12 (mileage 120,000 km) + coolant enhancer.

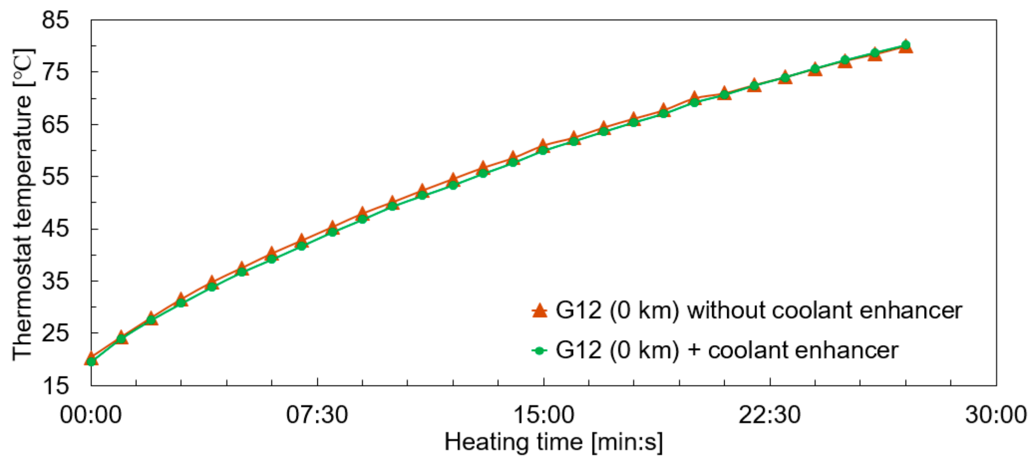


Figure 11. Thermostat temperature waveforms while the coolant in the short cooling circuit is heated to operating temperature for G12 (mileage 0 km) and G12 (mileage 0 km) + coolant enhancer.

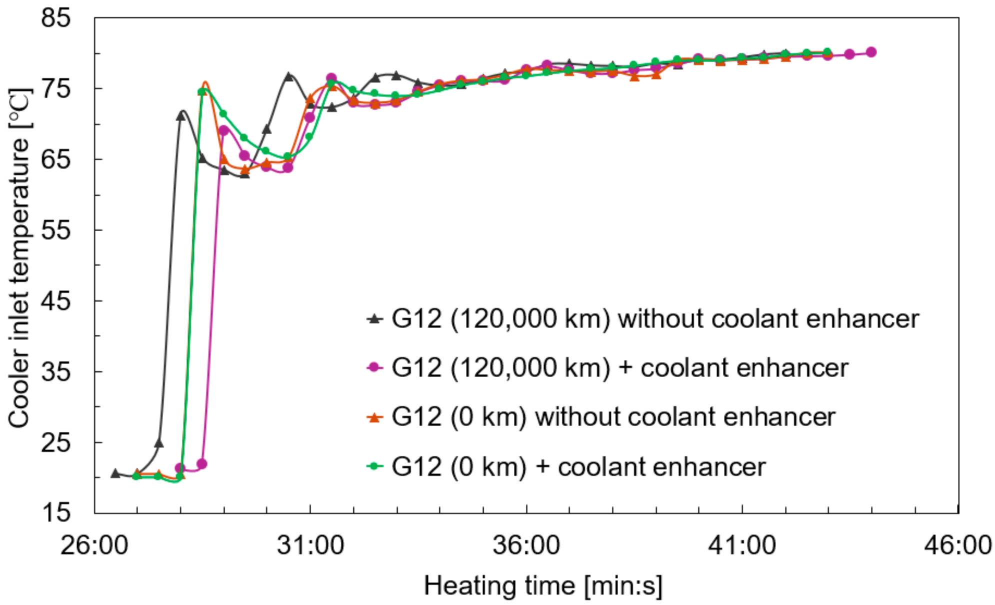


Figure 12. Radiator inlet temperatures while the coolant is heated in the short cooling circuit to operating temperature for G12 coolant with 120,000 km mileage, 120,000 km mileage + coolant enhancer, 0 km mileage, 0 km mileage + coolant enhancer.

By heating, the G12 coolant without additives reached the operating temperature of 80 °C one minute and 30 s earlier compared to the G12 with additive (coolant enhancer) for the 120,000 km mileage. For the 0 km mileage, the coolant enhancer had no effect on the length of the heating process and both samples reached operating temperature in the same time of 27 min (Figure 11). In the experimental measurements using G12 coolant with the 120,000 km mileage, a significant difference in the time required to heat the coolant to operating temperature was observed. This is due to the gradual degradation of the additives in the used fluid with the mileage of 120,000 km due to the long-term operation of the engine. Subsequent refilling of the additive (coolant enhancer) in the used coolant ensured that the heat transfer properties were recovered.

Figure 10 shows the coolant temperature waveforms at the inlet to the radiator (long cooling circuit) after the primary opening of the thermostat as a result of the coolant reaching the operating temperature. When the thermostat opens, some of the hot coolant is released from the short to the long cooling circuit, and then the cooled coolant is forced out of the radiator and into the pump suction. The described process can also be observed in the graphical waveforms (Figure 12), where there was a step increase in Tin for all tested coolant samples at the inlet of the radiator. After a rapid increase in T_{in}, they gradually stabilised to slightly below the operating temperature. Stabilisation was caused by forcing some of the cold ambient temperature liquid (primary thermostat opening) out of the radiator, where it further removed heat by free convection until the thermostatic valve was reopened (releasing hot coolant from the short cooling circuit).

3.2.2. The Process of Cooling the Coolant Below Operating Temperature with and Without Additives—Radiator Outlet

The coolant temperature at the radiator outlet when the coolant is cooled by the single ϕ400 mm diameter fan is shown in Figure 13. One-minute sections were recorded from the cooling process. This is because in case of recording temperatures only from a short period of time (e.g., 10 s, when the temperature drops below the operating temperature), the graphical results could be distorted, e.g., by different start-up of the fans to the maximum speed due to their previous wearing out, etc. Within 1 min of starting the radiator fan, the lowest outlet temperature of 38.46 °C was recorded for G12 coolant with 0 km mileage + coolant enhancer. This was followed by G12 with 0 km mileage without coolant enhancer with an outlet temperature value of 39.01 °C. The same ranking was recorded for the G12 coolants used with the 120,000 km mileage. The used G12 coolant with 120,000 km mileage + coolant enhancer reached T_{out} = 40.30 °C after one minute of cooling, while the G12 with 120,000 km mileage without coolant enhancer reached the outlet temperature higher by 0.64 °C.

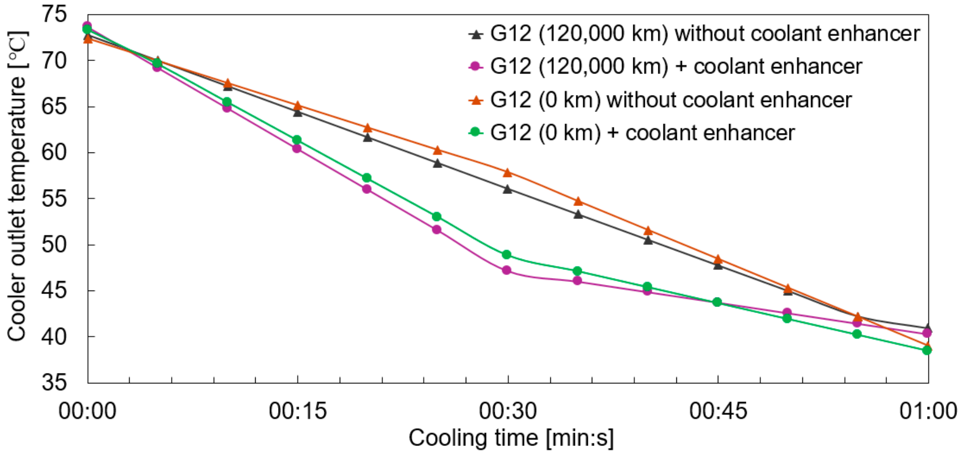


Figure 13. Radiator outlet temperatures while the coolant is cooling by fan (ϕ400 mm, 2130 RPM) for G12 coolant with 120,000 km mileage, 120,000 km mileage + coolant enhancer, 0 km mileage, 0 km mileage + coolant enhancer.

For the next experimental measurements, a pair of fans (ϕ345 mm, 1530 RPM and ϕ290 mm, 2040 RPM) was used (Figure 14). We have assumed that this variant will be able to cool the coolants to the lowest outlet temperature within one minute. Almost the entire heat exchange surface of the radiator is covered by the airflow generated by the pair of fans, which was not possible to achieve compared to a single fan alone. G12 coolant with 0 km mileage + coolant enhancer reached the output lowest temperature T_{out} = 33.63 °C, followed by G12 coolant with 0 km mileage without coolant enhancer T_{out} = 35.28 °C, a difference of 1.65 °C. The used G12 coolant with the mileage of 120,000 km with and without coolant enhancer reached the outlet temperature of 37.16 °C and 38.63 °C, respectively.

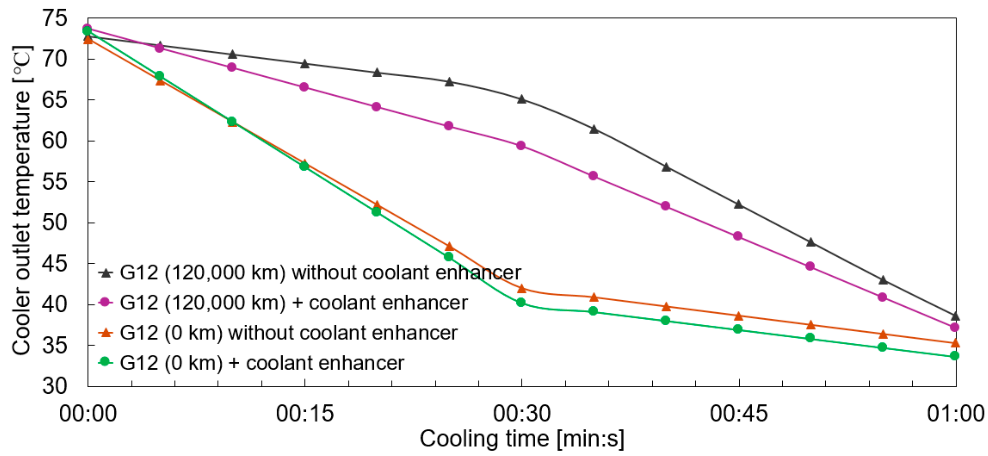


Figure 14. Radiator outlet temperatures while the coolant is cooling by a pair of fans (ϕ345 mm, 1530 RPM and ϕ290 mm, 2040 RPM) for G12 coolant with 120,000 km mileage, 120,000 km mileage + coolant enhancer, 0 km mileage, 0 km mileage + coolant enhancer.

The outlet temperatures for cooling by the single fan with diameter ϕ345 mm are shown in Figure 15. This is the smallest fan diameter among the fans we tested and thus it was assumed that the coolant temperatures at the radiator outlet would be the highest in this case, i.e., the lowest heat dissipation. After 1 min of fan start-up, the G12 coolant outlet temperatures were recorded in order from lowest to highest: 0 km mileage with and without coolant enhancer (T_{out} = 42.02 °C and T_{out} = 43.40 °C), 120,000 km mileage with and without coolant enhancer (T_{out} = 45.36 °C and T_{out} = 46.77 °C). With the addition of the additive (coolant enhancer) to the coolant with the mileage of 0 km and 120,000 km, there was a decrease in T_{out} by 1.38 °C and 1.41 °C, respectively, compared to coolants without coolant enhancers.

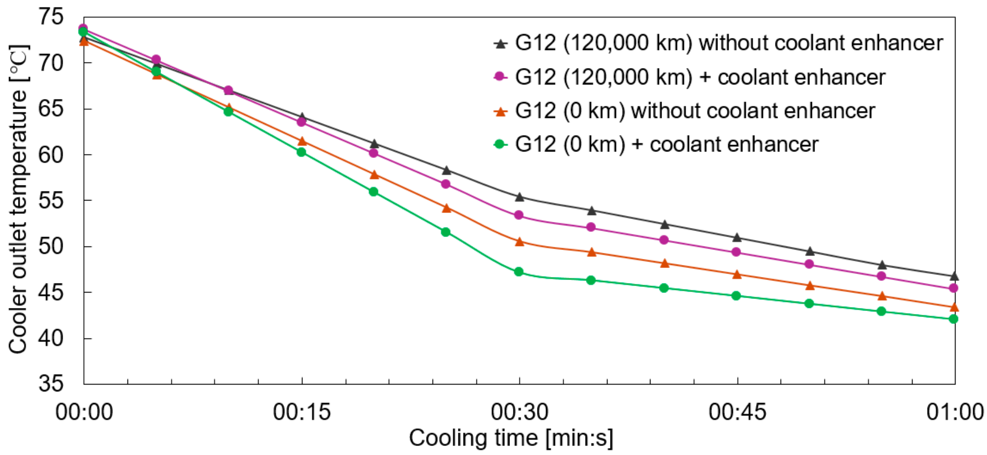


Figure 15. Radiator outlet temperatures while the coolant is cooling by fan (ϕ345 mm, 1530 RPM) for G12 coolant with 120,000 km mileage, 120,000 km mileage + coolant enhancer, 0 km mileage, 0 km mileage + coolant enhancer.

3.2.3. The Process of Cooling the Coolant Below Operating Temperature with and Without Additives—Radiator Inlet

During the coolant cooling process, it is not only important to record the outlet temperature of the cooled coolant flowing from the radiator to the water pump, but also to monitor the inlet temperature of the radiator. According to the inlet temperature, we can determine exactly at what point the thermostatic valve was closed. The closure was caused by the flow of cooled coolant from the radiator outlet through the water pump to the thermostat. In this, the cooled liquid with the temperature lower than the operating temperature caused the expansion element of the thermostat to cool down and closed the long cooling circuit, which at the same time caused the coolant flow in the radiator to stop. The earlier the thermostatic valve was closed, the earlier the engine was operating in optimum operation. The stated reasons served as a stimulus for recording the temperatures at the inlet to the radiator.

The graphical waveforms of the coolant inlet temperatures to the radiator when using the three variants of fans on the radiator during cooling are shown in Figures 16–18. Figure 16 shows the time taken for the coolants to cool below operating temperature (thermostatic valve closure) by using the ϕ400 mm diameter fan. For the initial 30 s, the steady-state temperature can be monitored, which fluctuated slightly about 80 °C operating temperature. This is because hot coolant is still flowing into the cooling circuit from the short cooling circuit until the cooled coolant is pumped to the thermostatic valve by the water pump. This process (from about 30 s onwards) can be observed from Figure 16 by a more rapid drop in temperatures below the operating temperature. The drop is caused by the interruption of the hot coolant flow from the short cooling circuit. Within one minute, the lowest inlet temperature was recorded when using G12 coolant with 0 km mileage + coolant enhancer (T_{in} = 78.01 °C). The G12 coolant with 0 km mileage but without coolant enhancer reached T_{in} = 78.23 °C, a difference of only 0.22 °C. The highest temperature was recorded for G12 coolant with the mileage of 120,000 km without coolant enhancer with T_{in} = 78.55 °C. After the addition of coolant enhancer, there was a 0.19 °C decrease in temperature (T_{in} = 78.36 °C).

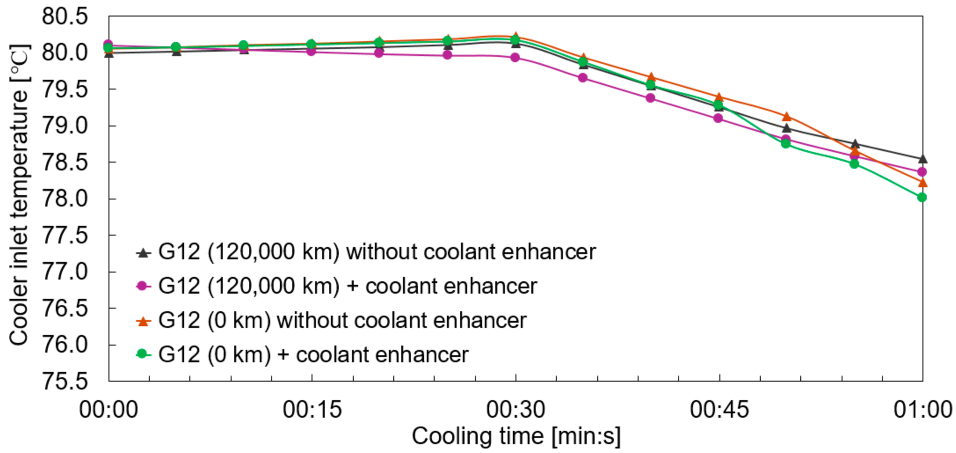


Figure 16. Radiator inlet temperatures while the coolant is cooling by fan (ϕ400 mm, 2130 RPM) for G12 coolant with 120,000 km mileage, 120,000 km mileage + coolant enhancer, 0 km mileage, 0 km mileage + coolant enhancer.

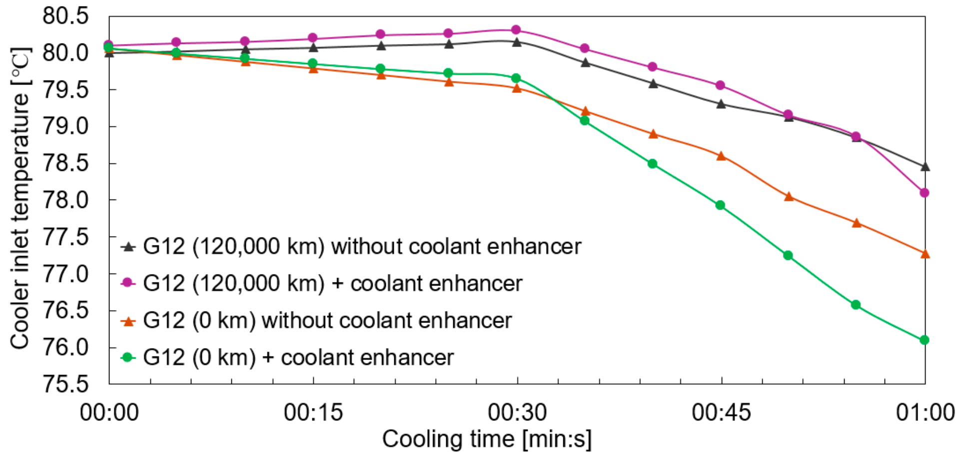


Figure 17. Radiator inlet temperatures while the coolant is cooling by pair of fans (ϕ345 mm, 1530 RPM and ϕ290 mm, 2040 RPM) for G12 coolant with 120,000 km mileage, 120,000 km mileage + coolant enhancer, 0 km mileage, 0 km mileage + coolant enhancer.

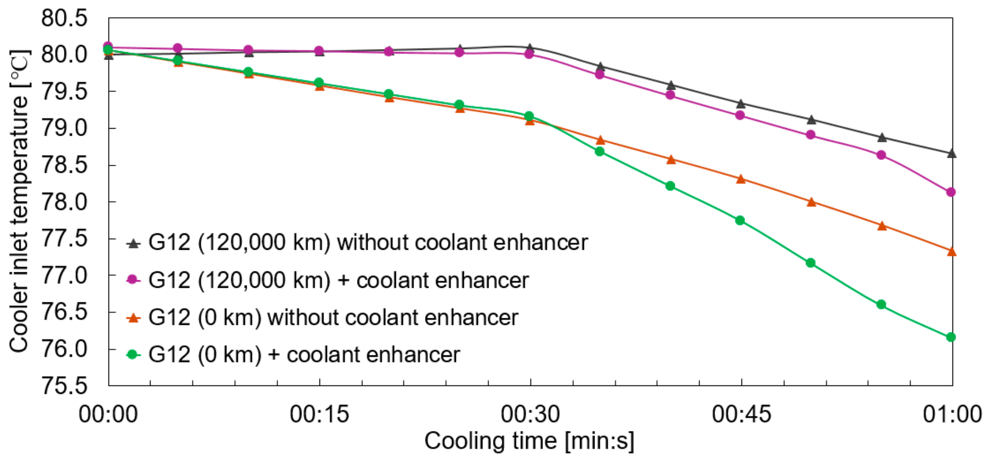


Figure 18. Radiator inlet temperatures while the coolant is cooling by fan (ϕ345 mm, 1530 RPM) for G12 coolant with 120,000 km mileage, 120,000 km mileage + coolant enhancer, 0 km mileage, 0 km mileage + coolant enhancer.

The coolant inlet temperature distribution at the radiator inlet during the cooling process by a pair of fans (ϕ345 mm and ϕ290 mm) is shown in Figure 17. During the cooling process, the lowest coolant inlet temperatures were recorded when cooling with a pair of fans, representing their highest heat dissipation efficiency. The lowest inlet temperature was achieved by G12 coolant with 0 km mileage + coolant enhancer with T_{in} = 76.09 °C, followed by the same sample but without coolant enhancer (T_{in} = 77.28 °C), a difference of 1.19 °C. The used G12 coolants with 120,000 km mileage + coolant enhancer and without coolant enhancer reached T_{in} = 78.09 °C and T_{in} = 78.46 °C, respectively, representing a difference of 0.37 °C.

The coolant inlet temperature waveforms for the cooling process with the fan diameter ϕ345 mm is shown in Figure 18. The lower ability to dissipate excess heat through the smallest diameter fan of the tested ones can be observed here. Therefore, the inlet coolant temperatures were also the highest among the tested samples. The lowest inlet temperature was recorded for G12 coolant with 0 km mileage + coolant enhancer (T_{in} = 76.15 °C) and the same coolant but without coolant enhancer (T_{in} = 77.33 °C), a difference of 1.18 °C. The highest inlet temperature was recorded for G12 coolant with the mileage of 120,000 km without coolant enhancer (T_{in} = 78.66 °C). After the addition of coolant enhancer, there was a 0.54 °C decrease in inlet temperature (T_{in} = 78.12 °C).

4. Conclusions

The influence of radiator core geometry and coolant additives on the heating and cooling process of the automobile engine radiator has been demonstrated by experimental measurements. The results of the influence of the core radiator geometry on the heating and cooling time of the G12 coolant can be summarized as follows:

- Coolant heating to 80 °C was 1 min and 30 s longer for the cooling circuit with the radiator consisting of the flat tubes and corrugated fins compared to the radiator consisting of the circular tubes and straight fins. This is due to the higher volume of the coolant flowing in the flattened tubes by 0.15 L. Faster overheating of the radiator cores in real operation is an undesirable phenomenon, because the individual engine components are more exposed to thermal stress and the radiator fan has to be run more frequently.
- In the cooling process, the radiator consisting of circular tubes and straight fins reached the lowest temperature at the inlet of the radiator when using a pair of fans with diameters of ϕ345 mm and ϕ290 mm (T_{in} = 78.01 °C). The single fans with diameters of ϕ345 mm and ϕ400 mm were less efficient by 0.18 °C and 1.28 °C, respectively. When considering the T_{out} values after one-minute cooling, the most intense coolant temperature decrease was recorded using a pair of fans (T_{out} = 39.11 °C). The single fans with diameters of ϕ400 mm and ϕ345 mm were less efficient by 1.28 °C and 4.60 °C, respectively.
- In the cooling process, the radiator consisting of flat tubes and corrugated fins reached the lowest temperature at the inlet of the radiator when using the fan with a diameter of ϕ345 mm (T_{in} = 77.56 °C). The ϕ400 mm diameter fan and the fan pair were less efficient by 1.22 °C and 2.33 °C, respectively. When considering the T_{out} values after one-minute cooling, the most intense decrease in G12 coolant temperature was recorded using a pair of fans T_{out} = 37.55 °C. The single fans with diameters of ϕ400 mm and ϕ345 mm were less efficient by 2.49 °C and 3.73 °C, respectively.
- When comparing the outlet temperatures of the radiators when varying their core geometry, the radiator with flat tubes and corrugated fins achieved lower values by 0.35 °C, 1.56 °C, and 2.43 °C for the fan with a diameter of ϕ400 mm, the fan pair, and the fan with a diameter of ϕ345 mm, respectively. It can be concluded that the radiator core with flattened tubes and corrugated fins provides a decrease in coolant temperature compared to a radiator with circular tubes and straight fins, making it more efficient in the cooling process.

- In summary, after 1 min of cooling, the flattened tubes with corrugated fins were the most effective, achieving the lowest inlet and outlet temperatures.
- The cover of the fan also had a significant effect on the fan cooling process, i.e., the fan ϕ400 mm without the cover had a lower efficiency of the cooling process in each case compared to the covered fan pair or the covered fan ϕ345 mm.

The results of the influence of coolant additives on the heating and cooling time of the new and used G12 coolant can be summarized as follows:

- By heating, the coolant without additives reached the operating temperature of 80 °C one minute and 30 s earlier compared to the G12 with additive (coolant enhancer) for the 120,000 km mileage (used coolant). For the 0 km mileage, the coolant enhancer had no effect on the length of the heating process. The addition of the coolant enhancer to the used coolant was significant because a difference in the time required to heat the coolant to operating temperature was achieved. Refilling of the coolant enhancer in the used coolant ensured that the heat transfer properties were recovered.
- When evaluating the outlet temperatures from the cooling process, the 120,000 km coolant achieved higher T_{out} compared to the new coolant in every case studied. The addition of the coolant enhancer to the used G12 decreased the T_{out} by 0.64 °C, 1.41 °C, and 1.47 °C for the radiator with fan diameter ϕ400 mm, ϕ345 mm, and a pair of fans (ϕ345 mm and ϕ290 mm), respectively. The addition of the coolant enhancer to the new G12 has decreased the T_{out} by 0.55 °C, 1.38 °C, and 1.65 °C for the radiator with the fan diameter of ϕ400 mm, ϕ345 mm, and a pair of fans (ϕ345 mm and ϕ290 mm) respectively.
- The radiator with a pair of fans achieved the lowest coolant outlet temperatures for all fan variants studied. The fan pair covered almost the entire heat exchange surface of the radiator with airflow, which was not possible with the single fans. The fan with a diameter of 400 mm without the cover showed the least efficient cooling in each case.
- When evaluating the radiator inlet temperatures from the cooling process, coolant with 120,000 mileage achieved higher T_{in} compared to the new coolant in every case studied. The addition of the coolant enhancer to the used G12 decreased the T_{in} by 0.19 °C, 0.54 °C, and 0.37 °C for the radiator with a single fan diameter of ϕ400 mm, ϕ345 mm, and a pair of fans (ϕ345 mm and ϕ290 mm), respectively. The addition of the coolant enhancer to the new G12 has decreased the T_{in} by 0.22 °C, 1.18 °C, and 1.19 °C for the radiator with a single fan diameter of ϕ400 mm, ϕ345 mm, and a pair of fans (ϕ345 mm and ϕ290 mm).

In summary, it can be concluded that the additive (coolant enhancer) added to the used G12 coolant has enhanced the cooling process. The coolant core composed of flattened tubes and corrugated fins reduced the cooling time of the coolant and thus enhanced the heat dissipation process.

Author Contributions: Conceptualization, Z.B. and M.L.; methodology, Z.B.; software, P.B.; validation, Z.B.; investigation, M.L. and Z.B.; resources, M.L. and Z.B.; data curation, Z.B.; writing—original draft preparation, Z.B. and M.L.; writing—review and editing, M.L.; supervision, Z.B.; project administration, M.L.; funding acquisition, M.L. All authors have read and agreed to the published version of the manuscript.

Funding: This research was funded by the EU NextGenerationEU through the Recovery and Resilience Plan for Slovakia under the project No. 09I03-03-V05-00016.

Data Availability Statement: The raw data supporting the conclusions of this article will be made available by the authors on request.

Conflicts of Interest: The authors declare no conflicts of interest.

References

1. Pang, S.C.; Kalam, M.A.; Masjuki, H.H.; Hazrat, M.A. A review on air flow and coolant flow circuit in vehicles—Cooling system. *Int. J. Heat Mass Transf.* **2012**, *55*, 6295–6306. [CrossRef]
2. Bougeard, D. Infrared thermography investigation of local heat transfer in a plate fin and two-tube rows assembly. *Int. J. Heat Fluid Flow* **2007**, *28*, 988–1002. [CrossRef]
3. Taler, J.; Ocłoń, P.; Taler, D.; Nowak-Ocłoń, M. Computer-aided determination of the air-side heat transfer coefficient and thermal contact resistance for a fin-and-tube heat exchanger. In *Heat Transfer Studies and Applications*; Kazi, S.N., Ed.; IntechOpen: Rijeka, Croatia, 2015; pp. 259–292.
4. Aytunc Erek, A.; Ozerdem, B.; Bilir, L.; Ilken, Z. Effect of geometrical parameters on heat transfer and pressure drop characteristics pf plate fin and tube heat exchangers. *Appl. Therm. Eng.* **2005**, *25*, 2421–2431. [CrossRef]
5. Wang, C.; Chi, K.; Chang, C. Heat transfer and friction characteristics of plain fin-and-tube heat exchangers, part II: Correlation. *Int. J. Heat Mass Transf.* **2000**, *43*, 2693–2700. [CrossRef]
6. Balanna, K.; Kishore, P.S. Evaluation of heat transfer and friction factor on wavy fin automotive radiator. *Int. J. Sci. Res. Dev.* **2015**, *3*, 144–147.
7. Turizo-Santos, J.; Barros-Ballesteros, O.; Fontalvo-Lascano, A.; Vasquez-Padilla, R.; Bula-Silvera, A. Experimental characterization of thermal hydraulic performance of louvered brazed plate fin heat exchangers. *Rev. Fac. De Ing. De Antioq.* **2015**, *74*, 108–116.
8. Pitambar, G.; Shambhu, K. Enhancement of Forced Convection Heat Transfer over Dimple Surface—Review. *Int. Multidiscip. Res. J.* **2012**, *1*, 51–57.
9. Shariff, K.B.; Abdullahi, B.; Abubakar, S.B. Modelling and simulations of car radiator: Effects of fins under the atmospheric condition of Kano, Nigeria. *J. Adv. Res. Fluid Mech. Therm. Sci.* **2018**, *48*, 1–16.
10. Patil, V.R.; Patil, S.S.; Kumbhar, V.; Kolhe, K. Review of problems of heat transfer in car radiator and suggested solutions. *Int. J. Sci. Res. Dev.* **2017**, *2*, 94–98.
11. Zhang, L.-Z.; Ouyang, Y.-W.; Zhang, Z.-G.; Wang, S.-F. Oblique fluid flow and convective heat transfer across a tube bank under uniform wall heat flux boundary conditions. *Int. J. Heat Mass Transf.* **2015**, *91*, 1259–1272. [CrossRef]
12. Patel, H.V.; Subhedar, D.G.; Ramani, B. Numerical Investigation of performance for Car Radiator Oval Tube. *Mater. Today Proc.* **2017**, *4*, 9384–9389. [CrossRef]
13. Ahmadi, M.H.; Ghazvini, M.; Maddah, H.; Kahani, M.; Pourfarhang, S.; Pourfarhang, A.; Heris, S.Z. Prediction of the pressure drop for CuO/(Ethylene glycol-water) nanofluid flows in the car radiator by means of Artificial Neural Networks analysis integrated with genetic algorithm. *Phys. A Stat. Mech. Its Appl.* **2020**, *546*, 1–15. [CrossRef]
14. Subhedar, D.G.; Ramani, B.M.; Gupta, A. Experimental Investigation of Heat Transfer Potential of Al_2O_3/Water-Mono Ethylene Glycol Nanofluids as a Car Radiator Coolant. *Case Stud. Therm. Eng.* **2018**, *11*, 26–34. [CrossRef]
15. Gollin, M.; Bjork, D. Comparative performance of ethylene glycol/water and propylene glycol/water coolants in automobile radiators. In Proceedings of the International Congress & Exposition, Atlanta, GA, USA, 17–22 November 1996; SAE Technical Paper 960372. pp. 115–123.
16. Roslan, A.; Arbanah, M.; Tukiman, N.; Ibrahim, N.N.; Juanil, A.R. The Effects of Ethylene Glycol to Ultrapure Water on Its Specific Heat Capacity and Freezing Point. *J. Appl. Environ. Biol. Sci.* **2017**, *7*, 54–60.
17. Rebsdat, S.; Mayer, D. *Ethylene Glycol*; Ullmann's Encyclopedia of Industrial Chemistry, Wiley VCH Verlag GmbH & Co. KGaA: Hoboken, NJ, USA, 2000; pp. 1–16.
18. Al-Amayreh, M. Experimental Study of Thermal Conductivity of Ethylene Glycol Water Mixtures. *Eur. J. Sci. Res.* **2020**, *10*, 62–68.
19. Brodnianská, Z.; Kotšmíd, S. Numerical Study of Heated Tube Arrays in the Laminar Free Convection Heat Transfer. *Energies* **2020**, *13*, 973–997. [CrossRef]