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Comparison of the experimental performance of round and flat tube automobile radiators for various coolants

Çeşitli soğutma sıvıları için dairesel ve düz tüplü otomobil radyatörlerinin deneysel performanslarının karşılaştırılması

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Comparison of the Experimental Performance of Round and Flat Tube Automobile Radiators for Various Coolants

Highlights

- *An experimental system for testing the performance of automobile radiators was developed.*
- *The performances of round and flat tube radiators were evaluated for four different engine coolants.*
- *In the tests, the air speed, air temperature at the radiator inlet and coolant flow rate were varied.*
- *For water, the flat tube radiator dissipated on average 4.8% more heat than the round tube one.*
- *For ethylene glycol, the flat tube radiator dissipated averagely 66.4% more heat than the round tube one.*

Graphical Abstract

In this study, the heat transfer performance of round tube and flat tube automobile radiators were experimentally evaluated and compared for various engine coolants using the test system shown below.

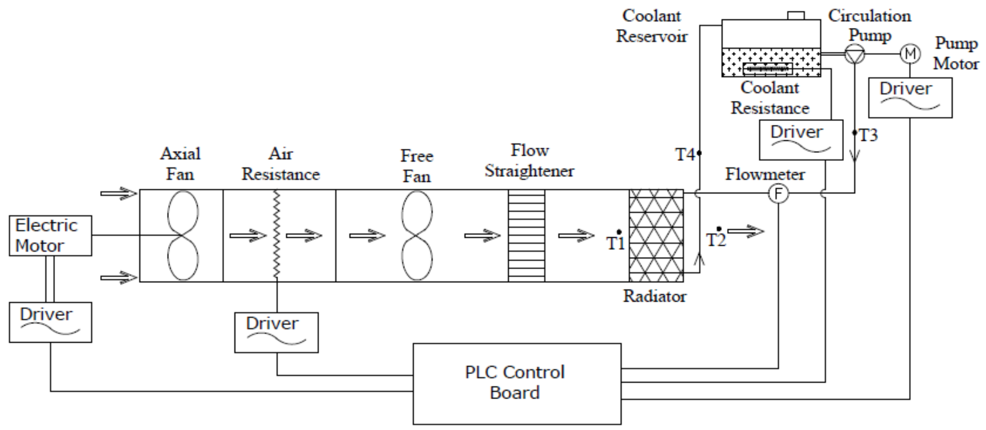

Figure. Schematic view of the test system

Aim

The aim of the study was to evaluate and compare the experimental heat transfer performances of round and flat tube automobile radiators for various engine coolants, namely water, ethylene glycol, their 50/50 mixture and a commercial heat transfer oil.

Design & Methodology

An experimental test system capable of using various types of radiators and coolants was developed. The radiators were tested by changing the air speed, air temperature entering the radiator and coolant flow rate in a broad range.

Originality

The performances of two types of automobile radiators serving the same engine were experimentally compared for four different engine coolants.

Findings

The flat tube radiator dissipated on average 4.8% and 66.4% more heat than the round tube one when the coolants were water and ethylene glycol, respectively.

Conclusion

The flat tube radiator rejects more heat and water is the best among the tested coolants in terms of heat transfer performance.

Declaration of Ethical Standards

The authors of this article declare that the materials and methods used in this study do not require ethical committee permission and/or legal-special permission.

Comparison of the Experimental Performance of Round and Flat Tube Automobile Radiators for Various Coolants

Araştırma Makalesi / Research Article

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ABSTRACT

A radiator test system was developed to test the heat transfer performance of automobile radiators for various engine coolants. The system was made up from a circulation pump, coolant reservoir, axial fan, electric heaters, PLC controlled drivers and instruments for various mechanical measurements along with the tested radiators. Two different radiators, namely round and flat tube ones, and four different engine coolants, namely water, ethylene glycol, their 50/50 mixture and a commercial heat transfer oil, were tested. The experimental heat dissipation rates of the radiators were evaluated under a broad range of operating conditions. The air speed was changed between 2 and 4 m s^{-1}, the coolant flow rate was varied between 0.1 and 0.3 l s^{-1}, and the air temperature at the inlets of the radiators was changed between 25 and 35 °C, while the coolant temperature was fixed at 90 °C in all tests. The flat tube radiator dissipated on average 4.8% more heat than the circular tube one for water coolant, while it rejected on average 66.4% more heat than the circular tube one for ethylene glycol. Furthermore, when the heat transfer oil was used as coolant, the flat tube radiator dissipated on average 101.6% more heat than the circular tube one.

Keywords: Engine cooling, radiator, coolant, automobile, ethylene glycol.

Çeşitli Soğutma Sıvıları İçin Dairesel ve Düz Tüplü Otomobil Radyatörlerinin Deneysel Performanslarının Karşılaştırılması

ÖZ

Bu çalışmada, otomobil radyatörlerinin ısı transferi performanslarını çeşitli motor soğutma sıvıları için test etmek amacıyla bir radyatör test sistemi geliştirilmiştir. Sistem, sirkülasyon pompası, soğutma sıvısı tankı, eksenel fan, elektrikli ısıtıcılar, PLC kontrollü sürücüler, çeşitli mekanik ölçüm cihazları ve test edilen radyatörlerden oluşturulmuştur. Testlerde dairesel ve düz tüplü iki farklı otomobil radyatörü ile su, etilen glikol, bunların 50/50 karışımı ve ticari bir ısı transfer yağı olmak üzere dört farklı soğutma sıvısı kullanılmıştır. Radyatörlerin attığı ısılar, geniş bir test koşulu aralığında deneysel olarak belirlenmiştir. Radyatörlerden geçen hava hızı 2 ve 4 m s^{-1} arasında, soğutma sıvısı debisi 0.1 ve 0.3 l s^{-1} arasında, radyatörlere giren hava akımının sıcaklığı 25 ve 35 °C aralığında değiştirilmiştir. Radyatörlere giren soğutma sıvısı sıcaklığı ise tüm testler için 90 °C'de sabit tutulmuştur. Soğutma sıvısı olarak su kullanıldığında, düz tüplü radyatörün dairesel tüplüye göre ortalama % 4.8 daha fazla ısı attığı; etilen glikol kullanıldığında ise düz tüplü radyatörün dairesel tüplüden ortalama % 66.4 daha fazla ısı attığı tespit edilmiştir. Soğutma sıvısı olarak ısı transfer yağı kullanılması durumunda, düz tüplü radyatörün dairesel tüplüye kıyasla ortalama % 101.6 daha fazla ısı attığı belirlenmiştir.

Anahtar Kelimeler: Motor soğutma, radyatör, soğutma sıvısı, otomobil, etilen glikol.

1. INTRODUCTION

In internal combustion engines, some portion of the energy released during combustion is transferred to the surroundings as heat. Although the exhaust gas expels a ratio of this heat to the atmosphere, the engine cooling system dissipates about 17−26% and 16−35% of the fuel heating value to the atmospheric air in spark ignition (SI) and Diesel engines, respectively [1]. In order to prevent distortion and fatigue cracking caused by thermal stresses, the temperature to be experienced by engine components made from aluminium alloys is limited to 300 °C [1]. Furthermore, the maximum allowable cylinder surface temperature is 180 °C to avoid from excessive wear and engine damage caused by deterioration of the lubricating oil film due to high temperatures [2]. In SI engines, overheating may also cause pre-ignition and knock. On the other hand, insufficient engine temperatures may cause incomplete burning of fuel, thus leading to low engine thermal efficiency and deteriorated exhaust emissions.

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In liquid-cooled internal combustion engines, a coolant is commonly circulated by a pump through the water jackets located in the engine block to remove the heat released by combustion so that the engine operating temperature is maintained at desirable values. This excess heat absorbed by the engine coolant is conveyed from the engine block to the radiator, and dissipated into the atmospheric air with the aid of an axial fan. When the engine temperature is below the desired value, the coolant is bypassed the radiator, and recirculated in the engine block. When the coolant reaches a certain temperature, the thermostat opens and sends the coolant to the radiator for heat dissipation.

Nowadays, volume and weight limitations compel automotive manufacturers to develop more efficient radiators with higher heat rejection rates for a certain volume allocated for the radiator. The heat transfer performance of a specific radiator depends substantially on the heat transfer characteristics of the coolant and geometric characteristics of the radiator. Manufacturers usually employ either round or flat tube radiators with different channel and fin geometries together with water and ethylene glycol mixture as engine coolant.

The open literature contains limited number of studies on the performance comparison of various types of automotive radiators and engine coolants due to the competitive nature of the sector.

Gollin and Bjork evaluated and compared the experimental performance of water, ethylene glycol, propylene glycol, and their blends in automotive radiators [3]. They determined that water yielded the best heat transfer performance followed by 50/50 ethylene glycol/water, 50/50 propylene glycol/water, 70/30 ethylene glycol/water, 70/30 propylene glycol/water, and finally propylene glycol in decreasing order.

Oliet et al. developed a heat exchanger model to perform parametric studies on automotive radiators [4]. As a compromise between classical ε-NTU method and CFD, their model determined the heat transfer and hydraulic performance of automobile radiators as a function of the air and water mass flow rates, air and water inlet temperatures, fin pitch, louver angle and coolant flow layout. Their results showed the utility of their numerical model as a rating and design tool for automotive radiators.

Sany et al. evaluated experimental performance of a car radiator under various test conditions [5]. Based on experimental data, they developed a method to determine the heat transfer coefficient using ε-NTU method. The practical usefulness of their calculation method is its provision of empirical data, which can be used in the design state.

Peyghambarzadeh et al. compared the performance of automotive radiators using pure water, pure ethylene glycol and their blends with different ratios [6]. They also experimentally investigated the effect of Al2O3 nano particles added to these blends on the heat convection inside the radiator tubes. They determined that addition of nano particles increased the Nusselt number by up to 40% inside the tubes.

Amrutkar and Patil performed a theoretical analysis of automobile radiators using ε-NTU method, and validated their one-dimensional simulation software [7]. They concluded that the simulated heat dissipation rate and coolant temperature were very close to the experimental ones.

Nieh et al. investigated the experimental performance of some nano coolants, which were water and ethylene glycol containing various ratios of Al2O3 and TiO2 nano particles, in vehicle radiators [8]. Their test results showed that the use of nano coolants increased the radiator heat dissipation rate by 25.6%.

Sheikhzadeh et al. prepared a computer software for simulating the heat transfer performance of a car radiator with ethylene glycol containing copper nano particles at various ratios [9]. They found that the increase in the radiator heat dissipation rate was about 26.9% when the volume fraction of these nano particles was increased from 0 to 5%.

Vajjha et al. analysed the heat transfer of flat tube automotive radiators with Al2O3 and CuO nano particles in ethylene glycol and water mixture using ANSYS software [10]. They determined that, for the same pumping power, Al2O3 and CuO nano fluids up to 3% and 2% particle volumetric concentrations, respectively, provided higher heat transfer coefficients than that of the base fluid.

Ahmed et al. evaluated the experimental performance of a car radiator using pure water and TiO2 -water nano fluid as coolants [11]. They tested the nano fluids containing three different ratios of TiO2, namely 0.1, 0.2 and 0.3% in volume. They found that the 0.2% TiO2-water coolant provided up 47% higher heat transfer coefficient compared to the pure water.

Keklik and Hosoz developed a rig to test the heat transfer performance of automobile radiators [12]. They evaluated the heat dissipation rates of a round tube radiator using water, ethylene glycol, 50/50 mixture of ethylene glycol/water and a commercial heat transfer oil as coolants under a broad range of test conditions. They found that the highest radiator heat dissipation rates were obtained with water, followed by 50/50 ethylene glycol/water mixture, ethylene glycol and heat transfer oil, respectively. According to their results, water yielded 77−112% higher heat dissipation rates than ethylene glycol and 234−264% higher rates than the heat transfer oil.

In this study, the experimental heat transfer performance of two different types of automobile radiators, namely round and flat tube ones, were evaluated and compared for four different engine coolants, which were water, ethylene glycol, their 50/50 mixture and a commercial heat transfer oil.

2. DESCRIPTION OF THE EXPERIMENTAL SETUP

The layout and photograph of the experimental system used for evaluating the heat transfer performance of automobile radiators are shown in Figures 1 and 2, respectively. The heat dissipation rate from a specific radiator depends on the speed and inlet temperature of the air steam passing over the radiator along with the flow rate and inlet temperature of the coolant passing through the radiator. The effects of all these variables were taken into account in the experimental system. The system mainly consists of coolant and air circuits, a test radiator, electric motors, drivers, a PLC control board for the drivers and various instruments for mechanical measurements.

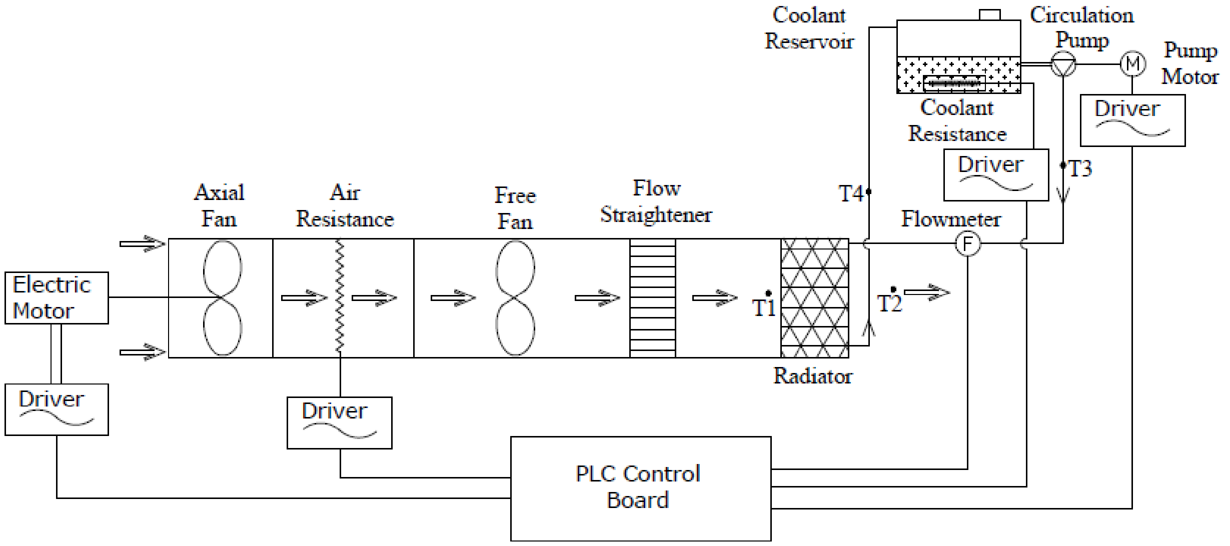

Figure 1. Layout of the experimental system used for testing radiators.

The components located in the air circuit are an axial fan used for providing the air stream passing over the radiator, an air heater resistance used for heating the air stream to the required test temperature and a free fan-flow straightener couple used for obtaining both a uniform air speed and air temperature at the radiator inlet. The axial fan has a maximum air flow rate of 2.66 $m^3\ s^{-1}$, and it was operated by a three-phase 550 W AC motor. The fan motor was driven via a driver connected to a PLC so that the air speed could be adjusted to the desired value by changing the frequency of the electric power supplied to the motor. The operation of the air heater, which had a maximum heating capacity of 10 kW, was also controlled by the PLC through solid state relays so that the air temperature at the radiator inlet could be adjusted to the desired value.

On the other hand, the components located in the coolant circuit are a coolant reservoir, a coolant heater resistance placed in the reservoir, a coolant circulation pump along with the test radiator. The coolant reservoir was made from steel sheet, and has dimensions of 416×296×248 mm^3. The coolant resistance, which had a total heating capacity of 15 kW, was also controlled by the PLC through solid state relays so that the coolant temperature entering the radiator could be kept at the desired value. Similarly, the motor driving the circulation pump was operated with a driver connected to the PLC to simulate changing engine speed, thereby providing various coolant flow rates.

In order to evaluate the heat dissipation rate of the test radiator, various mechanical measurements were performed. For this aim, the coolant volume flow rate was measured by a turbine type flow meter installed upstream of the radiator. The air speed passing over the radiator was measured at the inlet of the radiator by an air velocity transmitter. Moreover, the air and coolant temperatures were measured at the inlet and outlet of the radiator, as shown in Figure 1, by type J thermocouples connected to the thermocouple module of the PLC. Technical specifications of the measurement devices are reported in Table 1.

Figure 2. Photograph of the experimental setup.

Table 1. Technical specifications of the instruments.

Physical quantity	Instrument	Range	Accuracy
Temperature	Type J thermocouple	0−400 °C	±2.2 °C
Air speed	Air velocity transmitter	0−20 m s^{-1}	<0.2 m s^{-1}
Coolant flow rate	Turbine meter	0−6 m^{3}h^{-1}	±1%

As mentioned above, the resistance heaters as well as the fan and circulation pump motors were operated via the PLC to obtain the required test conditions. The PLC also served for acquiring the coolant flow rate, air speed and temperature data. Before performing a test, the desired coolant flow rate, air speed, air inlet temperature and coolant inlet temperature were entered the PLC via a touchpad screen, which is indicated in Figure 3. Because the PLC was previously programmed to maintain the required test conditions, it operated the system at these conditions by controlling the motor and resistance drivers. During the tests, the temperatures of the coolant and air streams entering and leaving the radiator as well as coolant volume flow rate and air speed were monitored on the control screen of the PLC, as shown in Figure 3.

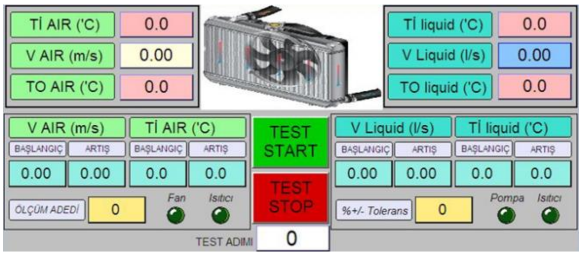

Figure 3. Touchpad control screen of the PLC.

The tested radiators were a finned round tube aluminium radiator and a louvered-fin flat tube aluminium radiator. Both of these radiators belonged to a light commercial vehicle having a four-cylinder diesel engine with a stroke volume of 1910 cm^3 and a maximum power of 77 kW. The photographs of the round tube (radiator 1) and flat tube (radiator 2) test radiators are shown in Figures 4 and 5, respectively, while their geometric characteristics are reported in Table 2.

Figure 4. Photographs of the round tube radiator (radiator 1) from different perspectives.

Figure 5. Photographs of the flat tube radiator (radiator 2) from different perspectives.

Table 2. Geometric characteristics of the round and flat tube radiators.

Characteristics	Round tube radiator	Flat tube radiator
Frontal area (m^2)	0.219	0.175
Depth (mm)	34	30
Fin pitch (fpi)	1.6	2.7
Core size (mm)	700 x 312.5 x 34	770 × 293 × 30
Core volume (l)	7.44	6.77
Tube no.	32	48
Pass no.	2	2
Tube outside diameter (mm)	10.3	2 × 12
Tube hydraulic diameter (mm)	9.8	2.9
Fin thickness (mm)	0.07	0.05
Minimum fin length between the tubes (mm)	8.3	9.0

It is seen in Table 2 that the core volume of the flat tube radiator is about 9.0 % lower than that of the round tube radiator employed in the same vehicle. Since the manufacturing processes of the flat and round tube radiators are completely different, they have different tube numbers as well as fin pitches and thicknesses even though their volumes are close to each other.

The coolants used in the tests were water, ethylene glycol (EG), their 50/50 mixture on a volume basis, and a commercial heat transfer oil (HTO). Table 3 reports important thermophysical properties of these coolants.

In all tests, the coolant temperature at the radiator inlet ($T_{c,in}$) was maintained at 90 °C, while the air temperature at the radiator inlet ($T_{a,in}$) was changed between 25 and 35 °C with intervals of 5 °C. On the other hand, the air speed passing over the radiator was varied between 2 and 4 m s^{-1} with intervals of 1 m s^{-1}. Because the test radiators belonged to an engine with a maximum power of 77 kW, it was assumed that the radiators dissipated about 25 kW at this power. However, due to the technical limitations of the laboratory where tests were performed, the maximum heating capacity of the coolant resistance was determined as 15 kW. Another constraint was to keep the coolant temperature entering the radiator at 90 °C. When the speed of the air stream passing over the radiators exceeded 4 m s^{-1}, the heating capacity of the coolant resistance became insufficient to maintain 90 °C coolant temperature. Therefore, the upper limit of the air speed passing over the radiators was determined as 4 m s^{-1}. On the other hand, the lower limit of the air speed was selected as 2 m s^{-1} to simulate the operations at low vehicle speeds, which are more critical for the radiators due to the deteriorated heat transfer at low speeds. Furthermore, in accordance with the studies of Patel et al. [14] and Tasuni et al. [15], the coolant flow rate passing through the radiator was changed between 0.1 and 0.3 l s^{-1} with intervals of 0.1 l s^{-1}.

In the study, each radiator was tested under totally 27 different combinations of test conditions for each coolant. Furthermore, each test was repeated five times and the averaged steady-state data were employed for the performance evaluation. Thus, totally 1080 test runs were carried out for all radiators and coolants. The experimental system reached the steady-state in maximum 15 min after changing the test conditions. The radiator heat dissipation rate was evaluated from

$$\dot{Q} = \rho_c\, \dot{V}_c\, c_c\left(T_{c,in} - T_{c,out}\right) \quad (1)$$

where ρ_c is the coolant density, $\dot{V}_c$ is the coolant volume flow rate, c_c is the coolant specific heat, $T_{c,in}$ is the coolant temperature at the radiator inlet and $T_{c,out}$ is the coolant temperature at the radiator outlet.

Table 3. Thermophysical properties of the coolants used in the tests at 90 °C [13].

Coolant	**Density** (kg m^{-3})	**Specific heat** (kJ kg^{-1} K^{-1})	**Dynamic viscosity** (mPa s)	**Thermal conductivity** (W m^{-1} K^{-1})
Water	965	4.205	0.316	0.676
EG	1064	2.677	2.640	0.261
50% water + 50% EG (volume basis)	1019	3.616	0.819	0.432
HTO	860	2.150	6.70	0.13

2.1. Uncertainty Analysis

The uncertainty for the radiator heat dissipation rate was determined using the method suggested by Moffat [16]. This method assumes that the function R is to be calculated from a set of totally N independent variables represented by

$$R = R(X_1, X_2, \ldots, X_N) \quad (2)$$

The uncertainty of the result R can be found by combining the uncertainties of the individual terms using a root-sum-square method as expressed below.

$$\Delta R = \left[\sum_{i=1}^{N}\left(\frac{\partial R}{\partial X_i}\Delta X_i\right)^2\right]^{1/2} \quad (3)$$

Using the accuracies for the measured variables reported in Table 1 and evaluating Eq. (1) in Eq. (3), the total uncertainty of the radiator heat dissipation rate was estimated as 4.1%.

3. RESULTS AND DISCUSSION

The heat dissipation rates of the round and flat tube radiators as a function of the test conditions and coolant types were indicated in Figures 6−11.

In Figure 6, the heat dissipation rates of the radiators were compared for two different coolant volume flow rates, namely 0.1 and 0.3 l s^{-1}, as a function of the air inlet temperature when the air speed passing over the radiators was maintained at 2 m s^{-1}. For a specific coolant flow rate, the heat dissipation rates decreased

for both radiators and all coolants with rising air inlet temperature. Furthermore, water rejects the highest heat, followed by 50/50 water/EG mixture, EG and HTO in decreasing order for both radiators. For the round tube radiator and a coolant flow rate of 0.1 l s^{-1}, the average heat dissipation rates for water, mixture, EG and HTO were 8.5 kW, 6.8 kW, 4.0 kW and 3.0 kW, respectively. When the coolant flow rate was increased to 0.3 l s^{-1} in the round tube radiator, the average heat dissipation rates for water, mixture, EG and HTO reached to 12.1, 9.6, 5.7 and 4.2 kW, respectively. On the other hand, for the flat tube radiator and a coolant flow rate of 0.1 l s^{-1}, the average heat dissipation rates for water, mixture, EG and HTO were 9.4, 8.7, 7.0 and 6.2 kW, respectively. When the coolant flow rate was increased to 0.3 l s^{-1} in the flat tube radiator, the average heat dissipation rates for water, mixture, EG and HTO became 11.5, 10.6, 8.6 and 7.7 kW, respectively. These results reveal that the flat tube radiator usually rejects greater heat than the round tube radiator except the operation with water for the coolant volume flow rate of 0.3 l s^{-1}. On the other hand, for the air speed of 2 m s^{-1} and coolant flow rate of 0.1 l s^{-1}, the flat tube radiator rejected 41.7% more heat than the round tube one as an average of the four coolants in the range of test conditions. However, when the coolant flow rate was increased to 0.3 l s^{-1} for the same air speed, the flat tube radiator rejected on average 22.7% more heat than the round tube one.

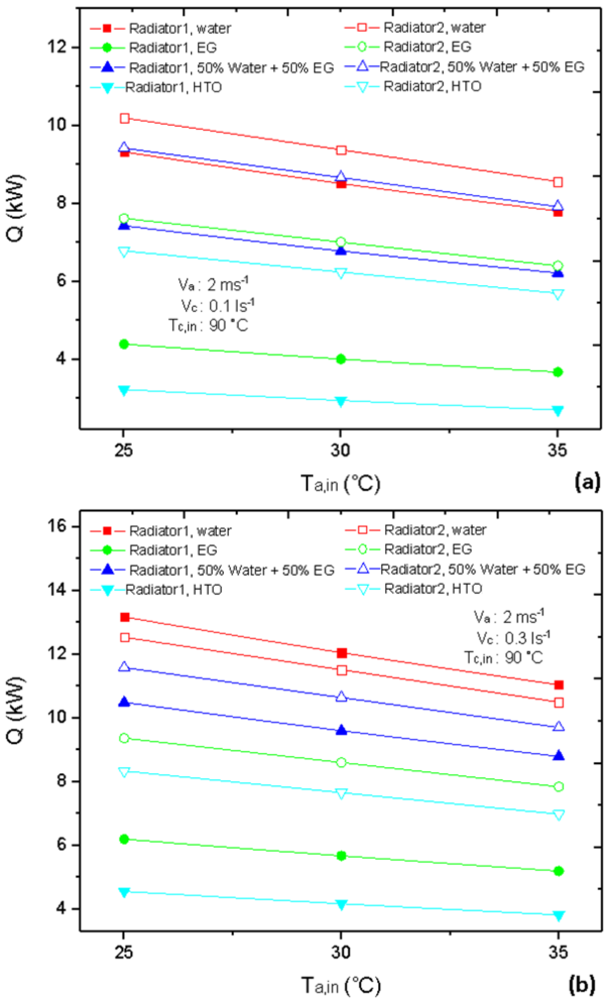

Figure 6. The radiator heat dissipation rates for 2 m s^{-1} air speed as a function of the air inlet temperature for two different coolant flow rates; (a) $\dot{V}_c = 0.1\,\mathrm{l\,s^{-1}}$, (b) $\dot{V}_c = 0.3\,\mathrm{l\,s^{-1}}$.

Because water was the coolant with the highest specific heat and thermal conductivity, it yielded the largest convection heat transfer inside the radiator tubes, thus providing the highest radiator heat rejection rates. On the other hand, the HTO resulted in the lowest heat rejection rates since it had the lowest specific heat and thermal conductivity. The total surface area of the liquid side of the flat tube radiator was 0.93 m^2, while that of the round tube radiator was 0.31 m^2. Although the total surface area of the air side of the flat tube radiator was 3.85 m^2 while that of the round tube radiator was 9.30 m^2, the liquid side area is more influential than the air side one. Therefore, the flat tube radiator rejects more heat than the round tube one.

Figure 7 indicates the heat dissipation rates of the radiators for two different coolant volume flow rates as a function of the air inlet temperature when the air speed passing over the radiators was kept at 4 m s^{-1}. Similar to the previous results, water rejected the highest heat and HTO rejected the lowest heat for both radiators. For the round tube radiator and coolant flow rate of 0.1 l s^{-1}, the average heat dissipation rates for water, mixture, EG and HTO were 10.2, 8.1, 4.8 and 3.5 kW, respectively. However, the average heat dissipation rates for water, mixture, EG and HTO reached to 16.0, 12.8, 7.5 and 5.5 kW, respectively, when the coolant flow rate was raised to 0.3 l s^{-1} for the round tube radiator. On the other hand, for the flat tube radiator and coolant flow rate of 0.1 l s^{-1}, the average heat dissipation rates for water, mixture, EG and HTO were 12.3, 11.3, 9.2 and 8.2 kW, respectively. When the coolant flow rate was raised to 0.3 l s^{-1} for the flat tube radiator, the average heat dissipation rates for water, mixture, EG and HTO became 16.3, 15.0, 12.2 and 10.8 kW, respectively. For the air speed of 4 m s^{-1} and coolant flow rate of 0.1 l s^{-1}, the flat tube radiator rejected 57.1% more heat than the round tube one as an average of the four coolants in the range of test conditions. On the other hand, when the coolant flow rate was raised to 0.3 l s^{-1} for the same air speed, the flat tube radiator rejected on average 32.4% more heat than the round tube one.

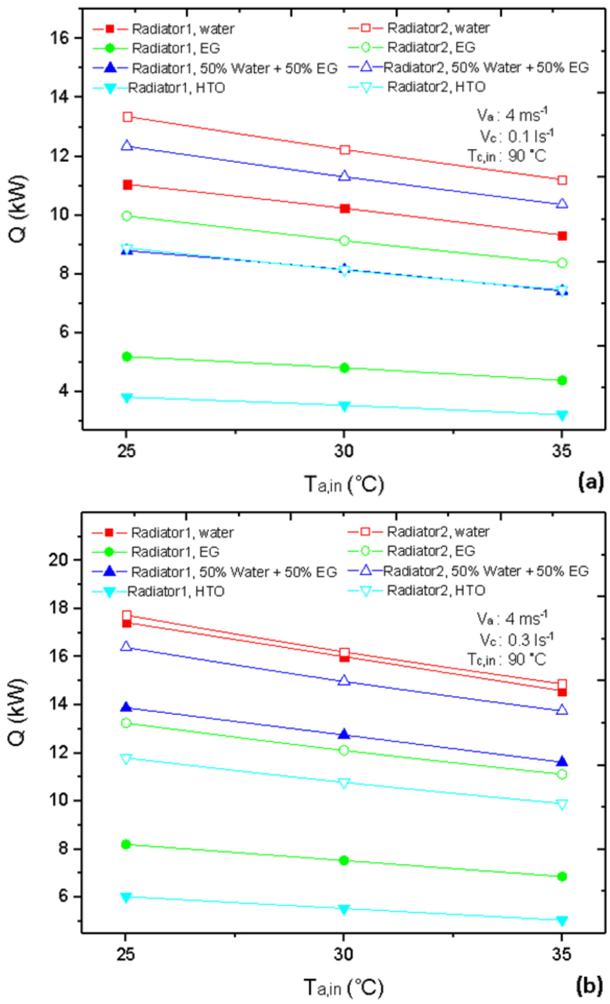

Figure 7. The radiator heat dissipation rates for 4 m s^{-1} air speed as a function of the air inlet temperature for two different coolant flow rates; (a) $\dot{V}_c = 0.1\,\mathrm{l\,s^{-1}}$, (b) $\dot{V}_c = 0.3\,\mathrm{l\,s^{-1}}$.

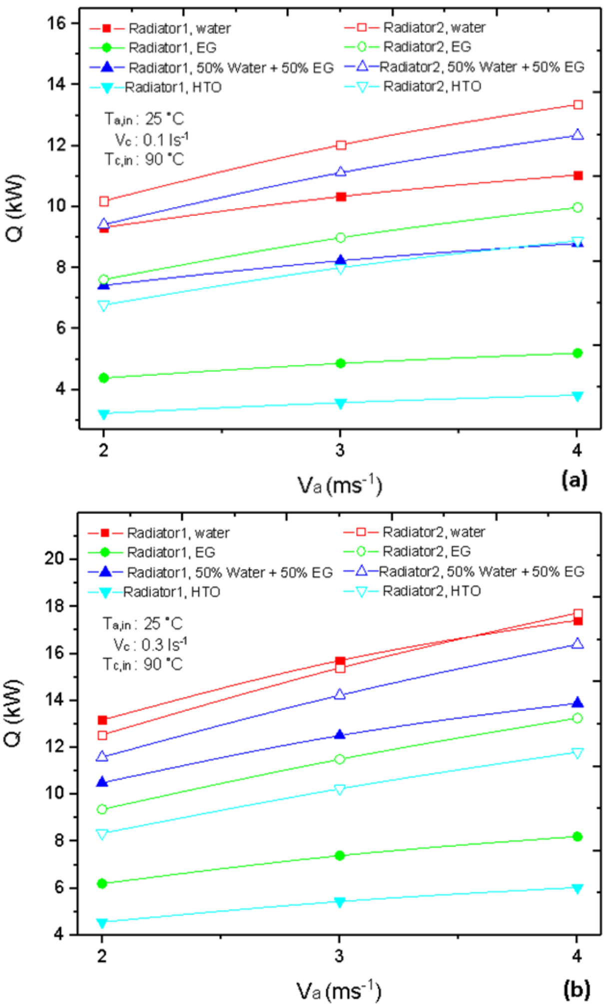

Figure 8. The radiator heat dissipation rates for 25 °C air inlet temperature as a function of the air speed for two different coolant flow rates; (a) $\dot{V}_c = 0.1\,\mathrm{l\,s^{-1}}$, (b) $\dot{V}_c = 0.3\,\mathrm{l\,s^{-1}}$.

Figure 8 shows the heat dissipation rates of both radiators for two different coolant volume flow rates, namely 0.1 and 0.3 l s^{-1}, as a function of the air speed when the air temperature entering the radiators was maintained at 25 °C. When the air speed was raised, the heat dissipation rate increased for both radiators and all coolants. For the round tube radiator and coolant flow rate of 0.1 l s^{-1}, the average heat dissipation rates for water, mixture, EG and HTO were 10.2, 8.2, 4.8 and 3.5 kW, respectively. When the coolant flow rate was increased to 0.3 l s^{-1} in the round tube radiator, the average heat dissipation rates for water, mixture, EG and HTO reached to 15.4, 12.3, 7.3 and 5.3 kW, respectively. On the other hand, for the flat tube radiator and coolant flow rate of 0.1 l s^{-1}, the average heat dissipation rates for water, mixture, EG and HTO were 11.9, 11.0, 8.9 and 7.9 kW, respectively. When the coolant flow rate was increased to 0.3 l s^{-1} in the flat tube radiator, the average heat dissipation rates for water, mixture, EG and HTO became 15.2, 14.1, 11.4 and 10.1 kW, respectively. For the inlet air temperature of 25 °C and coolant flow rate of 0.1 l s^{-1}, the flat tube radiator rejected 50.9% more heat than the round tube one as an average of the four coolants. However, when the coolant flow rate was raised to 0.3 l s^{-1} for the same air inlet temperature, the flat tube radiator rejected on average 28.0% more heat than the round tube one.

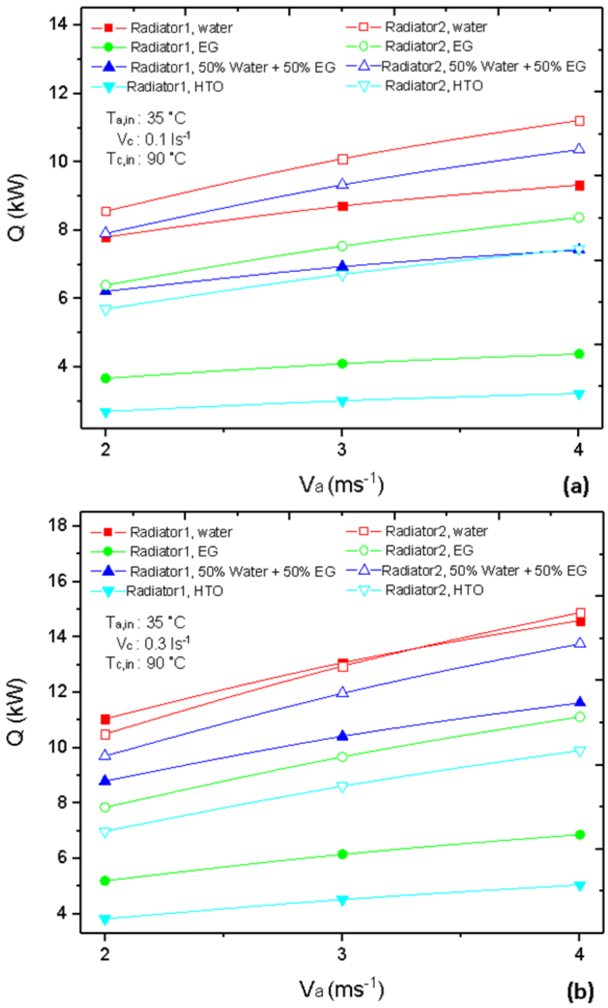

Figure 9. The radiator heat dissipation rates for 35 °C air inlet temperature as a function of the air speed for two different coolant flow rates; (a) $\dot{V}_c = 0.1\,\mathrm{l\,s^{-1}}$, (b) $\dot{V}_c = 0.3\,\mathrm{l\,s^{-1}}$.

Figure 9 reports the heat dissipation rates of both radiators for two different coolant volume flow rates as a function of the air speed when the air temperature entering the radiators was kept at 35 °C. The heat dissipation rate increased for both radiators and all coolants with rising air speed. For the round tube radiator and coolant flow rate of 0.1 l s^{-1}, the average heat dissipation rates for water, mixture, EG and HTO were 8.6, 6.9, 4.1 and 3.0 kW, respectively. However, the average heat dissipation rates for water, mixture, EG and HTO in the round tube radiator reached to 12.9, 10.3, 6.1 and 4.5 kW, respectively, when the coolant flow rate was increased to 0.3 l s^{-1}. On the other hand, when the flat tube radiator was employed and coolant flow rate was kept at 0.1 l s^{-1}, the average heat dissipation rates for water, mixture, EG and HTO were 10.0, 9.2, 7.4 and 6.6

kW, respectively. On the other hand, the average heat dissipation rates for water, mixture, EG and HTO in the flat tube radiator reached to 12.8, 11.8, 9.6 and 8.5 kW, respectively, when the coolant flow rate was raised to 0.3 l s^{-1}. For the inlet air temperature of 35 °C and coolant flow rate of 0.1 l s^{-1}, the flat tube radiator rejected 49.6% more heat than the round tube one as an average of the four coolants. However, when the coolant flow rate was raised to 0.3 l s^{-1} for the same air inlet temperature, the flat tube radiator rejected 28.6% more heat than the round tube one.

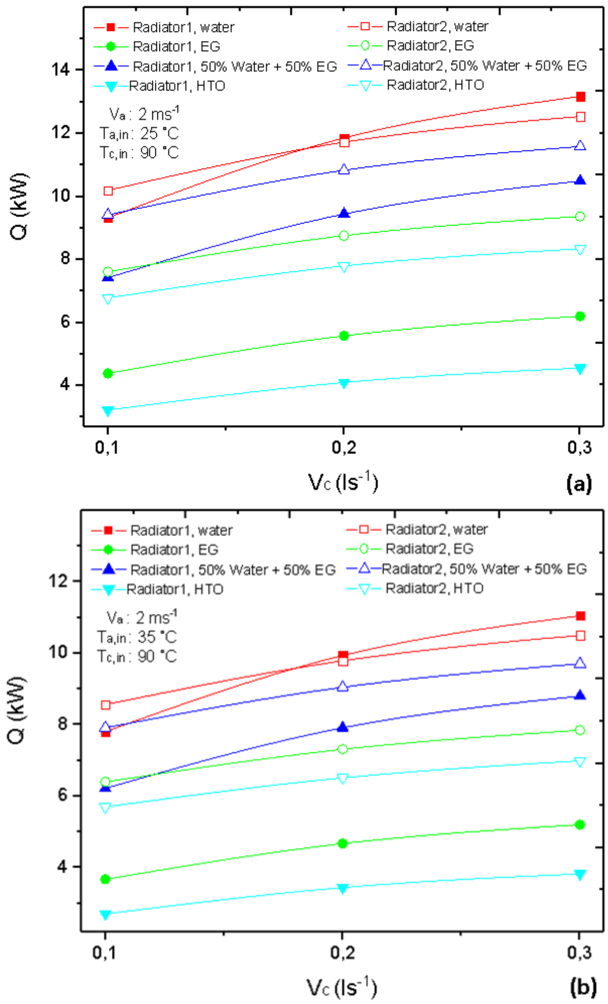

Figure 10. The radiator heat dissipation rates for 2 m s^{-1} air speed as a function of the coolant flow rate for two different air inlet temperatures; (a) $T_{a,in} = 25$ °C, (b) $T_{a,in} = 35$ °C .

Figure 10 shows the heat dissipation rates of both radiators for two different air inlet temperatures, namely 25 and 35 °C, as a function of coolant flow rate when the air speed was maintained at 2 m s^{-1}. When the coolant flow rate was raised, the heat dissipation rate increased for both radiators and all coolants. For the round tube radiator and air inlet temperature of 25 °C, the average heat dissipation rates for water, mixture, EG and HTO were 11.5, 9.1, 5.4 and 4.0 kW, respectively. When the air inlet temperature was increased to 35 °C in the round tube radiator, the average heat dissipation rates for water, mixture, EG and HTO dropped to 9.6, 7.6, 4.5 and 3.3 kW, respectively. On the other hand, for the flat tube radiator and air inlet temperature of 25 °C, the average heat dissipation rates for water, mixture, EG and HTO were 11.5, 10.6, 8.6 and 7.6 kW, respectively. However, the average heat dissipation rates for water, mixture, EG and HTO reduced to 9.6, 8.9, 7.2 and 6.4 kW, respectively, when the air inlet temperature was increased to 35 °C in the flat tube radiator. For the inlet air temperature of 25 °C and air speed of 2 m s^{-1}, the flat tube radiator rejected 27.7% more heat than the round tube one as an average of the four coolants in the range of test conditions. However, when the air inlet temperature was raised to 35 °C for the same air speed, the flat tube radiator rejected on average 28.4% more heat than the round tube one.

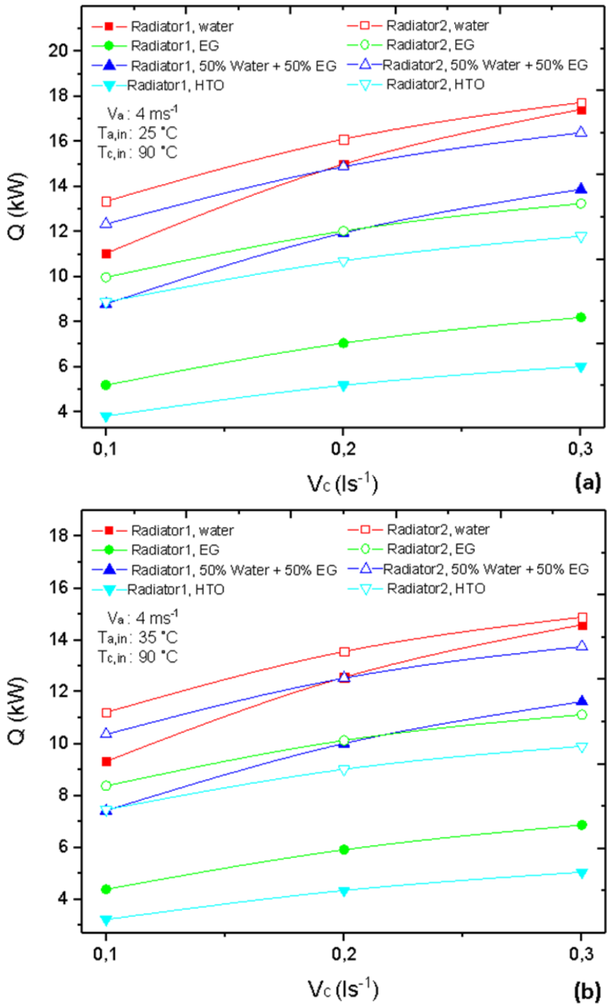

Figure 11. The radiator heat dissipation rates for 4 m s^{-1} air speed as a function of the coolant flow rate for two different air inlet temperatures; (a) $T_{a,in} = 25$ °C, (b) $T_{a,in} = 35$ °C.

Figure 11 reports the heat dissipation rates of both radiators for two different air inlet temperatures as a function of coolant flow rate when the air speed was maintained at 4 m s^{-1}. For the round tube radiator and air inlet temperature of 25 °C, the average heat dissipation rates for water, mixture, EG and HTO were 14.5, 11.5, 6.8 and 5.0 kW, respectively. When the air inlet temperature was raised to 35 °C in the round tube radiator, the average heat dissipation rates for water, mixture, EG and HTO reduced to 12.2, 9.7, 5.7 and 4.2 kW, respectively. On the other hand, for the flat tube radiator and air inlet temperature of 25 °C, the average heat dissipation rates for water, mixture, EG and HTO were 15.7, 14.5, 11.8 and 10.5 kW, respectively. However, the average heat dissipation rates for water, mixture, EG and HTO dropped to 13.2, 12.2, 9.9 and 8.8 kW, respectively, when the air inlet temperature was raised to 35 °C in the flat tube radiator. For the inlet air temperature of 25 °C and air speed of 4 m s^{-1}, the flat tube radiator rejected 38.9% more heat than the round tube one as an average of the four coolants in the range of test conditions. On the other hand, when the air inlet temperature was raised to 35 °C for the same air speed, the flat tube radiator rejected on average 38.7% more heat than the round tube one.

4. CONCLUSIONS

A radiator test system was developed, and the heat transfer performance of round and flat tube automobile radiators were experimentally evaluated for four different engine coolants, namely water, ethylene glycol, their 50/50 mixture and a commercial heat transfer oil. In all tests, the coolant temperature entering the radiator was kept at 90 °C, while the air stream temperature at the radiator inlet was changed between 25 and 35 °C with 5 °C intervals. On the other hand, the coolant flow rate was varied between 0.1 and 0.3 l s^{-1} with 0.1 l s^{-1} intervals, while the air speed was changed between 2 and 4 m s^{-1} with 1 m s^{-1} intervals. Thus, the tests were performed in a broad range of working conditions to simulate the operation of an actual automobile radiator. After evaluating the heat transfer performance of the radiators based on the steady-state test data, the following conclusions have been attained.

- Water resulted in the highest radiator heat dissipation rates in all tests, followed by 50/50 EG/water mixture, EG and HTO, respectively. Water yielded on average 16.0% higher radiator heat dissipation rate than 50/50 mixture, 63.4% higher dissipation rate than EG and 96.3% higher dissipation rate than HTO.
- When water was employed as coolant, the flat tube radiator rejected on average 4.8% more heat than the circular tube one. On the other hand, when the coolant was EG, the flat tube radiator rejected on average 66.4% more heat than the circular tube one. Moreover, when the coolant was 50/50 EG/water mixture, the flat tube radiator rejected on average 21.6% more heat than the circular tube one. Furthermore, the flat tube radiator rejected on average 101.6% more heat than the circular tube one when the coolant was HTO.
- As an average of the related tests, the flat tube radiator rejected 16.1% less heat when the air temperature at the radiator inlet was increased from 25 °C to 35 °C. Furthermore, it rejected on average 36.2% more heat when the air speed passing over the radiator was increased from 2 to 4 m s^{-1}. Moreover, it rejected on average 28.4% more heat when the coolant flow rate was increased from 0.1 to 0.3 l s^{-1}.
- As an average of the related tests, the round tube radiator rejected 16.3% less heat when the air temperature entering the radiator was raised from 25 °C to 35 °C. Moreover, it rejected on average 25.8% more heat when the air speed passing over the radiator was increased from 2 to 4 m s^{-1}. Finally, it rejected on average 50.3% more heat when the coolant flow rate was increased from 0.1 to 0.3 l s^{-1}.

These findings reveal that water yields the best heat transfer performance in expense of its corrosive nature and high freezing point, while the flat tube radiator rejects considerably more heat than the round tube one.

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DECLARATION OF ETHICAL STANDARDS

The author(s) of this article declare that the materials and methods used in this study do not require ethical committee permission and/or legal-special permission..

REFERENCES

[1] Heywood J.B., "***Internal Combustion Engine Fundamentals***", Mc-Graw-Hill, Inc., New York, (1998).

[2] Patil P.S., Belsare S.N., Borse S.L., "Analysis of internal combustion engine heat transfer rate to improve engine efficiency, specific power & combustion performance prediction", ***International Journal of Mechanical Engineering and Technology***, 3: 447−452, (2012).

[3] Gollin M. and Bjork D., "Comparative performance of ethylene glycol/water and propylene glycol/water coolants in automobile radiators", ***International Congress & Exposition***, Detroit, Michigan, U.S.A., February 26−29, (1996).

[4] Oliet C., Oliva A., Castro J. and Segarra C.D.P., "Parametric studies on automotive radiators", ***Applied Thermal Engineering***, 27: 2033−2043, (2007).

[5] Sany A.R.E., Saidi M.H. and Neyestani J., "Experimental prediction of Nusselt number and coolant heat transfer coefficient in compact heat exchanger performed with ε-NTU method", ***The Journal of Engine Research***, 18: 62−70, (2010).

[6] Peyghambarzadeh S.M., Hashemabadi S.H., Hoseini S.M. and Jamnani M.S., "Experimental study of heat transfer enhancement using water/ethylene glycol based nanofluids as a new coolant for car radiators",

International Communications in Heat and Mass Transfer, 38: 1283−1290, (2011).

[7] Amrutkar P.S. and Patil S.R., "Automotive radiator sizing and rating – simulation approach", ***IOSR Journal of Mechanical and Civil Engineering***, 01−05, (2013).

[8] Nieh H.M., Teng T.P. and Yu C.C., "Enhanced heat dissipation of a radiator using oxide nano-coolant", ***International Journal of Thermal Sciences***, 77: 252−261, (2014).

[9] Sheikhzadeh G., Hajilou M. and Jafarian H., "Analysis of thermal performance of a car radiator employing nanofluid", ***International Journal of Mechanical Engineering and Applications***, 2: 47−51, (2014).

[10] Vajjha R.S., Das D.K. and Ray D.R., "Development of new correlations for the Nusselt number and the friction factor under turbulent flow of nanofluids in flat tubes", ***International Journal of Heat and Mass Transfer***, 80: 353–367, (2015).

[11] Ahmed S.A., Ozkaymak M., Sözen A., Menlik T. and Fahed A., "Improving car radiator performance by using TiO2-water nanofluid", ***Engineering Science and Technology, an International Journal***, 21: 996−1005, (2018).

[12] Keklik E. and Hosoz M., "Development of a test rig for automotive radiators and preliminary tests using various engine coolants", ***9th International Automotive Technologies Congress***, Bursa, 559−568, (2018).

[13] Klein, S.A., "***Engineering Equation Solver (EES) Software***", F-Chart Software, Madison, WI, (2016).

[14] Patel B.M., Modi A.J. and Rathod P.P., "Analysis of engine cooling water pump of car & significance of its geometry", ***International Journal of Mechanical Engineering and Technology***, 4: 100−107, (2013).

[15] Tasuni M.L.M., Latiff Z.A., Nasution H., Perang M.R. M., Jamil H.M. and Misseri M.N., "Performance of a water pump in an automotive engine cooling system", ***UTM Jurnal Teknologi***, 78: 47−53, (2016).

[16] Moffat R.J., "Describing the uncertainties in experimental results", ***Experimental and Thermal Fluid Science***, 1: 3−17, (1988).