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Experimental Investigation of the Use of Refrigerant in the Cooling System of the Internal Combustion Engine: Preliminary Study

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Abstract: A new cooling system has been designed to reduce the adverse effects of the cooling system used in internal combustion engines and to create a more efficient system. R134a gas was used as the refrigerant in the engine cooling system in the experimental setup set up for the preliminary study. The cooling effect of gas masses of different weights (400, 450, 500, 550, 600 g) filled into the cooling system for cooling the simulated cylinder fixed at different temperatures (50, 60, 70, 80, 90, 100, 110, 120 °C) was determined. By measuring the temperature and pressure values in the system, comparisons were made by calculating the heat transfer coefficient, fluid flow rate and heat transfer rates of the system. The most efficient fluid weight used in the study was determined as 550 g. In the system without a circulation pump, the pressure difference between the heat receiving and heat giving sections allows the refrigerant gas to circulate. It has been observed that this situation is made possible by the increase in the heat transfer coefficient depending on the temperature increase. The absence of a pump in the system will contribute to engine performance. Having the refrigerant in gaseous form will enable smaller engine structures, lower vehicle weight, less fuel consumption and less exhaust emissions.

Keywords: Internal combustion engines, cooling system, R134a, refrigerants, heat transfer rate.

İçten Yanmalı Motorun Soğutma Sisteminde Soğutucu Akışkan Kullanımının Deneysel İncelenmesi: Ön Çalışma

Öz: İçten yanmalı motorlarda kullanılan soğutma sisteminin olumsuz etkilerini azaltmak ve daha verimli bir sistem oluşturmak amacıyla yeni bir soğutma sistemi tasarlanmıştır. Ön çalışma için kurulan deney düzeneğinde, motor soğutma sisteminde soğutucu akışkan olarak R134a gazı kullanılmıştır. Farklı sıcaklıklarda (50, 60, 70, 80, 90, 100, 110, 120 °C) sabitlenmiş simüle edilmiş silindiri soğutmak için soğutma sistemine doldurulan farklı ağırlıklardaki (400, 450, 500, 550, 600 g) gaz kütlelerinin soğutma etkisi belirlenmiştir. Sistemdeki sıcaklık ve basınç değerleri ölçülerek sistemin ısı transfer katsayısı, akışkan debisi ve ısı transfer hızları hesaplanarak karşılaştırmalar yapılmıştır. Çalışmada kullanılan en verimli akışkan ağırlığı 550 g olarak belirlenmiştir. Sirkülasyon pompası olmayan sistemde, ısı alan ve ısı veren bölümler arasındaki basınç farkı, soğutucu gazın dolaşımına olanak sağlamaktadır. Bu durumun, sıcaklık artışına bağlı olarak ısı transfer katsayısının artmasıyla mümkün olduğu gözlemlenmiştir. Sistemde pompa bulunmaması, motor performansına katkı sağlayacaktır. Soğutucu akışkanın gaz halinde olması, daha küçük motor yapıları, daha düşük araç ağırlığı, daha az yakıt tüketimi ve daha az egzoz emisyonu sağlayacaktır.

Anahtar kelimeler: İçten yanmalı motorlar, soğutma sistemi, R134a, soğutucu akışkanlar, ısı transfer oranı.

1. Introduction

Transportation has become an indispensable element of daily life. It is a connecting bridge between production and consumption. At the same time, besides being an indicator of development, it has also undertaken an essential economic and social development task. Vehicles, which are means of transportation, are in constant motion to realize this critical mission. Internal combustion engines provide an essential part of the energy needed for the movement of vehicles. While using this important vehicle is aimed to achieve the highest performance with the least fuel consumption and to give the least damage to the environment for humanity. In order to achieve the highest engine performance, the engine cooling system has important duties to provide fuel economy and to keep harmful emission values at low levels. Therefore, the differences in the studies in this direction of the researchers are high and varied. Recently, they have focused on using nano-refrigerants in the cooling system [1–6]. As a result of the studies, it has also been seen that lower-volume engines can be produced with the same power. In

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addition, It is possible to diversify research on cooling systems with the cylinder and piston surface coating method [7], with the technique of using different radiator materials [8], with the use of various additives included in the fuel [9], with the addition of additives to the engine oil [10], with the anti-wear studies [11], by making new engine part designs [12, 13] etc. with different studies.

The cooling system's responsibility is to keep the vehicle engine's temperature within the desired limit. This responsibility is to remove excess heat generated by the internal combustion engine to prevent overheating engine parts and engine oil [14]. In the cooling systems of internal combustion engines, either air or liquid refrigerant coolers are used. While air or liquid coolants prevent the engine from overheating, at the same time also undertake the task of enabling the highest engine performance to be obtained. However, to avoid damaging the engine parts of the heat energy obtained in the engine cylinders, it is a must that some of it is discharged to the outside through the cooling system without being used. However, in order not to damage the engine parts of the heat energy obtained in the engine cylinders, it is a necessity that some of it are discharged to the outside through the cooling system without being used. The instantaneously occurring temperature of approximately 2000 °C in the cylinder decreases to an average value with the cold mixture entering the combustion chamber. The temperature at which the engines are theoretically most efficient is considered to be between 500 - 900 °C, depending on the structure of the engines and the work they will do.

Considered the thermal and mechanical strengths of the engine parts, the high temperatures occurring in the cylinder must be removed from the engine [15]. The cooling system used to fulfill this task removes 33-39% of the heat energy released by the combustion of fuel in gasoline engines and 28-35% in diesel engines [16]. Along with that, even this average temperature reduces the mechanical strength of the engine parts [17].

Air-cooled systems cause significant reductions in engine performance in hot climate conditions. The air-refrigerant cooling system is simple and is generally preferred for cold climate conditions. On the other hand, a liquid refrigerant cooling system is preferred for all-season conditions. The pump that circulates refrigerants in both air-refrigerant and liquid-refrigerant cooling systems causes the engine power to decrease [18]. On the other hand, ethylene glycol, which is used to improve the boiling and freezing points of the refrigerant in liquid refrigerant cooling systems, adversely affects thermal conduction. Many studies have been carried out in the literature on cooling liquids and cooling systems in internal combustion piston engines. These studies show that it is possible to use refrigerants in different systems in the future, as well as that the cooling system in engines is open to development and change. The availability of gaseous refrigerants in engine cooling systems, which has yet to find a place in the literature, can be seen as one of these developments and change methods.

The preliminary study system installed can be thought of as the application of heat pipe design to the engine cooling system. Many studies on heat pipe systems have been carried out in the past. Domestic water heating [19] and automotive headlight cooling [20] are some of these works. In addition, many studies are being carried out today, such as cooling electronic systems such as mobile phones [21], air conditioning systems [22], and heating by utilizing geothermal resources [23]. The system, which is established with a heat pipe working logic in engine cooling systems, is thought to be an important preliminary study that needs to be worked on in this field. As stated above, it will be possible to eliminate the deficiencies in classical systems by using heat pipe design. The transportation of heat from a hot environment to a cold environment and the fact that no pumping mechanism is needed and at the same time refrigerant gases can be used in these systems is seen as a unique subject of study.

It is seen that R134a refrigerant, which is planned to be used as a refrigerant in the engine, has the possibility of being used in different current studies both experimentally and theoretically. For example, while evaluating the effect of R134a gas charge amounts on cooling in a system with variable compressor strokes [24], in the study conducted by Zhai et al. (2024) the flame retardancy properties of R134a gas was examined [25]. In addition, the comparison of the effect of using different amounts of R134a and R1234yf refrigerants in different amounts [26], as well as studies comparing these gases in terms of cooling are seen in the literature. In the studies that have been conducted, the statements that R134a is a better option [26–28] are included in these studies. At the same time, it is thought that the use of R134a refrigerant gas in the heat pipe, which can be preferred for space constraints in small-sized devices such as mobile phones and computers, will make significant contributions to engine cooling.

In the study a new cooling system using gaseous refrigerant as a coolant was studied to eliminate the existing deficiencies in the existing cooling systems and minimize the existing problems. Here, the possible contribution of the latent heat of evaporation of the refrigerant to the engine cooling performance is investigated. Gas refrigerants are an ideal option to solve the problems encountered in all climate conditions in cooling systems using air or liquid refrigerants. In other words, the problem of freezing in the refrigerant in winter conditions and the problems of boiling and overheating by insufficient cooling in summer season conditions will be eliminated. Since there will be no temperature difference around the cylinder when the engine is not running, there will be no refrigerant movement in engine cooling. However, the heat energy generated by the combustion effect that will occur in the cylinder as a result of the engine's operation will lead to the realization of a forward and upward movement of the refrigerant by heating. In the cooling system, there will be no need to use a pump with the principle that the refrigerant will naturally circulate with the pressure change difference caused by the temperature effect. Therefore, it is seen as more advantageous than the classical systems used.

It is helpful to pay attention to the global warming potential (GWP) and ozone layer depletion potential (ODP) values in terms of the damages caused to the environment by the gases used as the refrigerant in the system, which is considered a new design. Hydrofluorocarbon (HFC) refrigerants used in cooling systems should have no or low global warming values within the framework of legal norms. Due to this situation, the EURO-F gas regulations, the Montreal Protocol, and the Kyoto protocol have placed significant restrictions on refrigerants with high global warming potential [29,30]. In addition to these values, it is important to determine whether the gases used as refrigerants are toxic and flammable. In this regard, gases have been determined by classifying them as seen in Table 1 below by ASHRAE (American Society of Heating, Refrigerating and Air-Conditioning Engineers). Thanks to this classification, the effects of gas refrigerants on health can be understood [31,32]. At the point of determining the refrigerants to be used, as it can be understood when Table 1 is examined, the importance of refrigerants with low toxicity levels and A1 level, which is the lowest level of toxicity, should be addressed. With a new cooling system design that is planned to be used, it is aimed at creating a more environmentally friendly product.

The aim of this study was to determine whether a gas phase refrigerant could be used in an internal combustion engine cooling system installed as a prototype. In the study, surface temperature values were determined by using different amounts of refrigerants, and heat transfer rates were calculated.

Table 1. ASHRAE classification.

		Toxicity	
		Low	**High**
Flammability	**High**	A3	B3
	Mid	A2	B2
	Low	A2L	B2L
	None	A1	B1

As far as the authors know, it will contribute to the literature since there is no experimental and theoretical study in this area. In addition, the structures of the cooling systems will undoubtedly be reduced thanks to the gaseous refrigerants planned to be used. The shrinkage that will occur in the cooling system will also allow the engine structure to shrink; therefore, the fact that it will affect the weight of the vehicle will also allow the fuel consumption to decrease. This will allow for the reduction of harmful emissions caused by internal combustion engines. It should not be ignored that it will positively affect both economic and environmental health by producing less harmful emissions with less fuel consumption.

2. Material and Method

2.1. Test refrigerant

R134a gas, whose properties are given in Table 2, is preferred as the refrigerant used in the internal combustion engine cooling system. The following features were taken into consideration in the selection of the gas:
- High latent heat of evaporation to benefit from phase change,
- Low boiling point in atmospheric conditions,
- To be harmless to the environment and human health,
- High critical temperature and pressure,
- No risk of chemical reaction depending on the system to be used,
- Not to cause corrosion,
- Easy supply and low cost.

Chlorofluorocarbon (CFC) dichlorodifluoromethane (CCl_2F_2), known as R12 from the ASHRAE classification, is a refrigerant widely used in refrigeration systems and air-conditioning after its discovery in the 1930s [33]. In addition, the fact that it is the gas closest to the properties of the R12 gas, which has been banned today, has also been influential in the choice made. Because R12 gas, due to its properties, has been used as a very effective gas in the field of cooling. As a result of the research made by the General Motors company, it was decided that the most useful one among the refrigerants developed was R12, which was named Freon. Mass production was started for R12 gas in 1931 [34]. In the Montreal Protocol, R134a gas has been proposed as an alternative refrigerant to R12 [35].

Table 2. Properties of R134a and R12 refrigerants [31,32,36,37].

Refrigerant	Refrigerant Group	Chemical Formula	Security Class	GWP	ODP	Critical temperature (°C)	Critical Pressure (Bar)	Boiling Point (°C)
R134a	HFC	CH_2FCF_3	A1	1300	0	101.1	40.59	-26.07
R12	CFC	CCl_2F_2	A1	10200	1	111.97	41.4	-29.75

2.2. Installation of the experimental mechanism

It aimed to calculate the convection heat transfer coefficient of the R134a refrigerant in the establishment of the experimental setup. The schematic view of the experimental setup is given in Fig. 1 and its installed state is given in Fig. 2. This setup aims to create a simulator of the cylinder structure in an internal combustion engine. While creating this structure, the copper cylinder represents the cylinder in the internal combustion engine. The engine oil filled into the cylinder was increased to different temperature values and used to reach the desired temperature values of the cylinder wall. With the spiral structure formed by wrapping a copper pipe around the cylinder, the heat taken from the cylinder is transferred to the radiator with the help of the same copper pipe. The refrigerant cooled in the radiator is provided to reach the entrance of the spiral structure around the cylinder with the copper return pipe. In this way, it has been aimed that the refrigerant heated around the cylinder will circulate and heat transfer within the copper pipe. Temperature values were recorded with the Graphtec GL240 data logger device with the help of T-type thermocouples placed in certain parts of the established experimental system. In addition, the pressure and temperature values of the gas circulating in the system during the operation were determined with the help of the Testo 550 digital manifold measuring device.

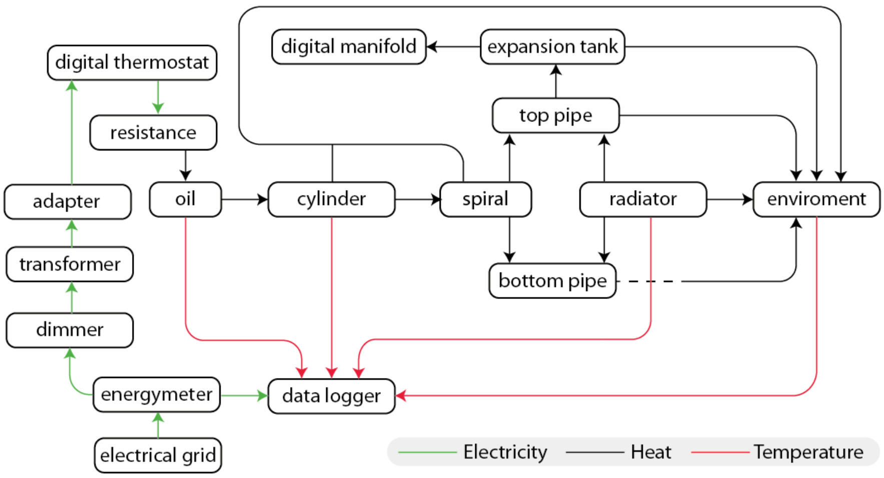

Figure 1. Experimental setup schematic.

After the system installation shown in Fig. 2, the air in the system was discharged with the help of a vacuum device to ensure the circulation of the gas refrigerant in the experimental setup. After this process, five different amounts of refrigerant (400, 450, 500, 550, and 600 g) were filled into the experiment system with the help of a digital scale.

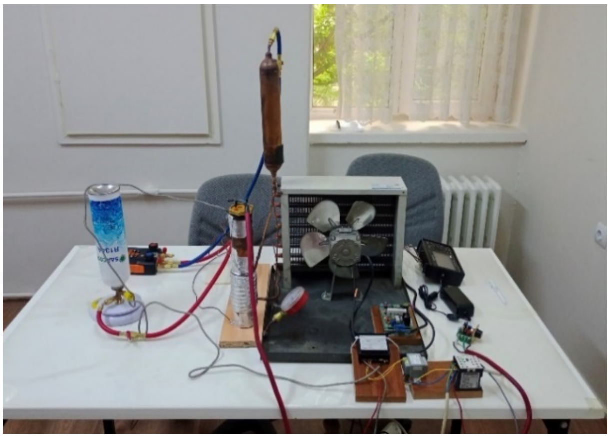

Figure 2. The experimental setup.

Thermocouple connections were made to the positions shown in Fig. 3 to see the temperature changes during the system's operation. The recorded temperature values were used in coordination with Excel and Refprop programs. Thermocouple positions and liquid-gas distribution in the system are visualized with solid models created using the Catia program.

2.3. Experiment process

Different amounts of refrigerant were filled into the system, and the oil temperature was fixed at the predetermined temperature value (50, 60, 70, 80, 90, 100, 110, and 120 °C) at each filling. The aim here is to compare the test results obtained according to different refrigerant amounts and cylinder temperatures to determine the most efficient state of the system and to see its operability. Table 3 shows the experimentally studied refrigerant quantities and the studies carried out depending on the fixed cylinder temperatures.

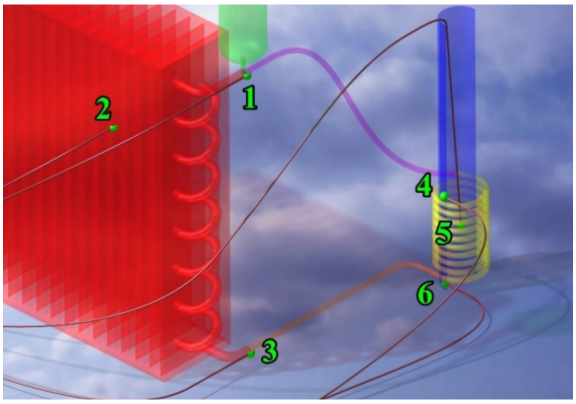

Figure 3. Thermocouple locations. (1. Radiator inlet, 2. Radiator fins, 3. Radiator outlet, 4. Spiral top end, 5. Oil, 6. Spiral bottom end).

Table 3. The number of experiments and variables.

Number of Experiments	**Refrigerant Amount** (g)	**Oil Temperature** (°C)
8	400	50, 60, 70, 80, 90, 100, 110, 120
8	450	50, 60, 70, 80, 90, 100, 110, 120
8	500	50, 60, 70, 80, 90, 100, 110, 120
8	550	50, 60, 70, 80, 90, 100, 110, 120
8	600	50, 60, 70, 80, 90, 100, 110, 120

Experiments were repeated three times for each refrigerant amount seen in Table 3. During the experiments, the ambient temperature was set at an average of 20 °C. For each experiment, besides the temperature and pressure values of the gas, the temperature values obtained from the thermocouples placed at eight different points were recorded. Two thermocouples for oil temperature adjustment in the cylinder and one thermocouple for measuring the ambient temperature were positioned in the experimental environment. The remaining 5 thermocouples were connected to the radiator inlet, radiator outlet, radiator fins, and spiral top and bottom parts.

2.4. Calculation method

The temperature values obtained with the help of the thermocouples shown in Fig. 3 and the pressure values measured through the digital manifold have been used for the calculations. The calculation method is based on the principle that the total heat loss from the system to the experimental environment and the heat stored in the system is equal to the heat drawn from the spiral part. The convection heat transfer coefficient of the refrigerant was calculated based on the heat drawn from the system. The following equations were used to calculate the liquid-gas ratios of the liquid and gas phase refrigerant in the system [38]:

$$\vartheta ln\frac{P_{sat}}{P_{cr}} = -7.686556\theta + 2.311791\theta^{3/2} - 2.039554\theta^2 - 3.583758\theta^4 \tag{1}$$

Where P_{sat} (MPa) is the pressure of the saturated vapor (Eq. 1), P_{cr} (MPa) is the critical point pressure of the refrigerant (Eq. 1), $\vartheta = T_{sat}/T_{cr}$ is the ratio of the gas saturation temperature to the critical point temperature, $\theta = 1 - \vartheta$ is the ratio of the difference between the critical and gas $P_{cr} = -7.6$ saturation temperatures to the critical point temperature. The value of saturated liquid density (ρ')is calculated from Eq. 2 [38]:

$$\rho' = 518.2 + 884.13\theta^{1/3} + 485.84\theta^{2/3} + 193.29\theta^{10/3} \tag{2}$$

The density value of the saturated steam can be calculated with the help of Eq. 3 [38]:

$$ln\frac{\rho"}{\rho_0} = -2.837294\theta^{1/3} - 7.875988\theta^{2/3} + 4.478586\theta^{1/2} - 14.140125\theta^{9/4} - 52,361297\theta^{11/2} \tag{3}$$

Where ρ'' (kg/m^3) is the density of saturated steam, and ρ_0 is the reference density determined for this equation as 516.86 kg/m^3. The liquid-gas volumes of the refrigerant can be calculated [34] with the help of Eq.4:

$$x = \frac{v_T - v_{sat,l}}{v_{sat,g} - v_{sat,l}} = \frac{m_v}{m_T} \tag{4}$$

Where x degree of dryness, v_T (total specific volume) is the ratio of the total volume of the refrigerant in the system to the total mass, $v_{sat,l}$ *is* the specific volume of the saturated liquid, $v_{sat,g}$ is the specific volume of the saturated steam, m_v is the vapor mass and m_T is the total mass. Eq. 5 is used to calculate the convection heat transfer coefficient:

$$h_{\text{R134a}} = \frac{Q_T}{A_{is}(T_{sat,avg} - T_{sat})} = \frac{Q_{rad} + Q_{p,t} + Q_{p,b} + Q_s + Q_{et} + Q_R}{A_{is}(T_{sat,avg} - T_{sat})} \tag{5}$$

Where h_{R134a} (W/m^{2}K) is the heat transfer coefficient of the R134a refrigerant, Q_T (W) is the total amount of heat drawn from the system, A_{is} (m^2) is the area from which the heat is drawn, that is, the inner surface area of the spiral, $T_{sat,avg}$ (K) is the average cylinder wall temperature obtained by averaging the temperature values measured from the thermocouple connections at the top and bottom of the spiral. Q_{rad}, $Q_{p,t}$, $Q_{p,b}$, Q_s, and Q_{et} (W) are the heat transferred to the surrounding environment from the radiator, top pipe, bottom pipe, spiral, and expansion tank regions shown in Fig. 3, respectively, and Q_R (W) is the heat stored in the refrigerant. For the regions shown in Fig. 3 was taken an average surface temperature value. Table 4 shows that the surface temperatures of these regions are calculated by averaging the temperatures of which thermocouple locations.

Table 4. Surface temperatures of the regions.

Region	Temperature Averaging Locations
Radiator ($T_{rad,avg}$)	Radiator inlet ($T_{rad,in}$), outlet ($T_{rad,out}$) and fin ($T_{rad,f}$) temperatures
Expansion Tank ($T_{et,avg}$)	Spiral top ($T_{s,up}$) and radiator inlet temperatures
Top Pipe ($T_{tp,avg}$)	Spiral top and radiator inlet temperatures
Bottom Pipe ($T_{bp,avg}$)	Spiral bottom ($T_{s,bp}$) and radiator outlet temperatures
Spiral ($T_{s,avg}$)	Spiral top and spiral bottom temperatures

The following equations (Eq. 6 and 7) are used to calculate the heat transfer from the system to the environment.

$$Q_{con} = hA_s(T_s - T_R) \tag{6}$$

Where Q_{con} (W) is the amount of heat transferred by convection, h (W/m^{2}K) is the convection heat transfer coefficient, A_s (m^2) is the area of the surface where the heat convection takes place, T_s and T_R (K) are the temperatures of the surface and the refrigerant providing the convection, respectively [39]:

$$h = \frac{kNu}{L} \tag{7}$$

Where h (W/m^{2}K) is the convection heat transfer coefficient, k (W/mK) is the thermal conductivity, L (m) is the vertical length for the plates, while the diameter for the horizontal pipes is, and Nu is the Nusselt number. The k and Nu values vary depending on the properties of the refrigerant where natural convection takes place and the surface temperature [40]. The thermal conductivity for the heat transferred to the surrounding air is calculated from Eq. 8 [41]:

$$k_h = 0.02624\left(\frac{T_f}{300}\right)^{0.8646} \tag{8}$$

Where k_h (W/mK) is the thermal conductivity for air, and T_f (K) is the film temperature calculated by averaging the ambient temperature and surface temperature [41]. Two different equations are used to calculate the Nusselt number, Eq. 9 for vertical plates and Eq. 10 for horizontal pipes [40]:

$$Nu = \left(0.825 + \frac{0.387Ra^{1/6}}{\left[1+\left(0.492/\mathrm{Pr}\right)^{9/16}\right]^{8/27}}\right)^2 \tag{9}$$

$$Nu = \left(0.600 + \frac{0.387Ra^{1/6}}{\left[1+\left(0.559/\mathrm{Pr}\right)^{9/16}\right]^{8/27}}\right)^2 \tag{10}$$

Where Pr is the Prandtl number, and Ra is the Rayleigh number. The Rayleigh number can be multiplied by the Prandtl and Grashof numbers [40]. Eq. 11 can be used to calculate the Prandtl number for air [41]:

$$\mathrm{Pr}_{air} = 0.68 + 4.69\times10^{-7}(T_f - 540)^2 \tag{11}$$

Eq. 12 is used to calculate the Grashof number [40]:

$$\mathrm{Gr}_L = \frac{g\beta(T_s - T_R)L_c^3}{\nu^2} \tag{12}$$

Where g gravitational acceleration (m/s^2), $\beta = 1/T_f$ volumetric expansion coefficient (1/K), T_s surface temperature (K), T_R refrigerant temperature (K), ν kinematic viscosity of the refrigerant (m^2/s) [40]. The equation used in the calculation of the kinematic viscosity for air (ν_h) is as shown in Eq. 13 [41]:

$$\nu_h = \frac{1.458\times10^{-6}T_f^{1.5}}{\rho_h(T_f + 110.4)} \tag{13}$$

Where ρ_h (kg/m^3) is the density of air and is calculated as shown in Eq. 14 [41]:

$$\rho_h = \frac{P}{R_A T_f} \tag{14}$$

Where P (Pa) is the air pressure, $R_A = 287.05$ J/kgK is the gas constant for air [41]. After all these calculations, the amount of heat transferred to the surrounding environment by convection has been calculated. The following equations are used to calculate the heat stored in the system (Q_{sys}):

$$\dot{Q}_{sys} = \dot{m}_g.C_{p,g}(\Delta T) \tag{15}$$

Where $\dot{m}_{\mathrm{g}}$ (kg/s) is the mass flow rate of the gas, $C_{p.\mathrm{g}}$ (J/kgK) is the specific heat of the gas, and ΔT is the temperature difference between the two points of the gas [18]. The mass flow rate can be calculated using Eq. 16 [34]:

$$\dot{m} = \rho \mathrm{v} A_c \tag{16}$$

Where ρ (kg/m^3) is the density, A_c (m^2) is the cross-sectional area where the flow occurs, v (m/s) is the flow velocity [34]. The following Eq. 17 is used to determine the flow rate [42]:

$$\mathrm{v}=\sqrt{2(h_{g,f} - h_{g,i})} \tag{17}$$

Where $h_{\mathrm{g},f}$ (J/kgK) and $h_{\mathrm{g},i}$ (J/kgK) are the enthalpy values of the gas in the final and initial states [42]. It is known that $\left(C_{p.\mathrm{g}}(\Delta T)_{\mathrm{g}}\right)$ value in Eq. 15 is also equal to $\left(h_{\mathrm{g},f} - h_{\mathrm{g},i}\right)$ value [34]. This enthalpy difference has been used while making the calculations. The first value, the saturated vapor enthalpy, is the value when the liquid evaporates, and the last situation value, the enthalpy of the gas at temperature and pressure conditions read from the digital manifold during the experiment, has been taken as the basis.

The Eq. 18 used in the calculation of the enthalpy value of the R134a refrigerant is as follows [38]:

$$h_g = R_g T_g[1 + \tau(\Phi_\tau^{\mathrm{o}} + \Phi_\tau^r) + \delta\Phi_\delta^r] \tag{18}$$

Where h_{g} (J/kgK) is the enthalpy of the gas, $R_{\mathrm{g}} = 81{,}488856$ J/kgK is the gas constant for R134a, T_{g} (K) is the temperature of the gas, $\tau = T_{cr}/T_{\mathrm{g}}$ is the ratio of the critical point temperature to the gas temperature, $\delta = \rho_{\mathrm{g}}/\rho_{cr}$ is the ratio of gas density to critical point density, and ϕ_τ^{o}, ϕ_τ^{r}, and ϕ_δ^{r} are derivatives of general free energy equations [38].

Calculation of ϕ_τ^{o}, ϕ_τ^{r}, and ϕ_δ^{r} values is as seen in equations 19, 20, and 21, respectively [38]:

$$\Phi_\tau^{\mathrm{o}} = a_2^{\mathrm{o}} + \frac{a_3^{\mathrm{o}}}{\tau} + \sum_{j=4}^{N^o} a_j^o t_j^o \tau^{t_j^o - 1} \tag{19}$$

$$\Phi_{\tau}^{r} = \sum_{i=1}^{N_0} a_i t_i \delta^{d_i} \tau^{t_i - 1} + \sum_{k=1}^{4} \left(\exp\left(-\delta^k\right) \sum_{i=N_{k-1}+1}^{N_k} a_i t_i \delta^{d_i} \tau^{t_i - 1} \right) \tag{20}$$

$$\Phi_{\delta}^{r} = \sum_{j=1}^{N_0} a_i d_i \delta^{d_i - 1} \tau^{t_i} + \sum_{k=1}^{4} \left(\exp\left(-\delta^k\right) \sum_{i=N_{k-1}+1}^{N_k} a_i (d_i - k\delta^k) \delta^{d_i - 1} \tau^{t_i} \right) \tag{21}$$

The equation constants used here are shown in Table 5 [38]:

Table 5. Equation constants for equations 19, 20, and 21.

i	a_i	t_i	d_i	N_i	a_i^o	t_i^o	i	a_i	t_i	d_i
0	-	-	-	8	-	-	11	-0.0074859070	5	2
1	0.0558681700	-0.5	2	11	-1.019535	-	12	0.0001017263	1	4
2	0.4982230000	0.0	1	17	9.047135	-	13	-0.5184567000	5	1
3	0.0245869800	0.0	3	20	-1.629789	-	14	-0.0869228800	5	4
4	0.0008570145	0.0	6	21	-9.723916	-0.50	15	0.2057144000	6	1
5	0.0004788584	1.5	6	-	-3.927170	-0.75	16	-0.0050004570	10	2
6	-1.8008080000	1.5	1	-	-	-	17	0.0004603262	10	4
7	0.2671641000	2.0	1	-	-	-	18	-0.0034978360	10	1
8	-0.0478165200	2.0	2	-	-	-	19	0.0069950380	18	5
9	0.0142398700	1.0	5	-	-	-	20	-0.0145218400	22	3
10	0.3324062000	3.0	2	-	-	-	21	-0.0001285458	50	10

3. Findings and Discussion

The pressure of the refrigerant in the gas phase in the system has increased depending on the amount of refrigerant filled into the system. Fig. 4 shows the gas pressures available in the system:

According to the experimentally obtained information, the pressure (Fig. 4) and temperature (Fig. 5) graphs of the gas in the system are quite similar to each other. The temperature values obtained here are very close to the saturation temperature of the refrigerant. In a way, this means that the gas absorbs a small amount of heat after evaporation. This may be due to the fact that the evaporated liquid moves very quickly after passing into the gas phase, so it does not have time to absorb more heat and the heat conduction is low in gases. The purpose of the installed system is also to benefit from the high heat transfer that occurs during evaporation. The variation of gas temperatures in the system is shown in the graphs in Fig. 5.

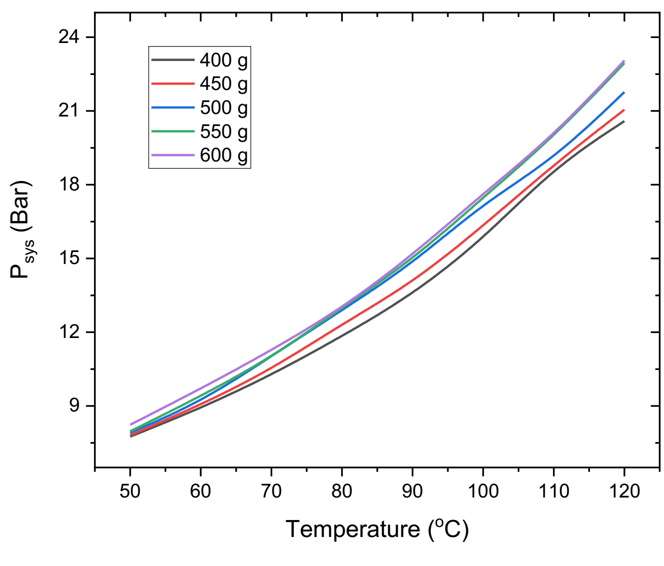

Figure 4. Comparison of the gas pressures in the system according to gas weights.

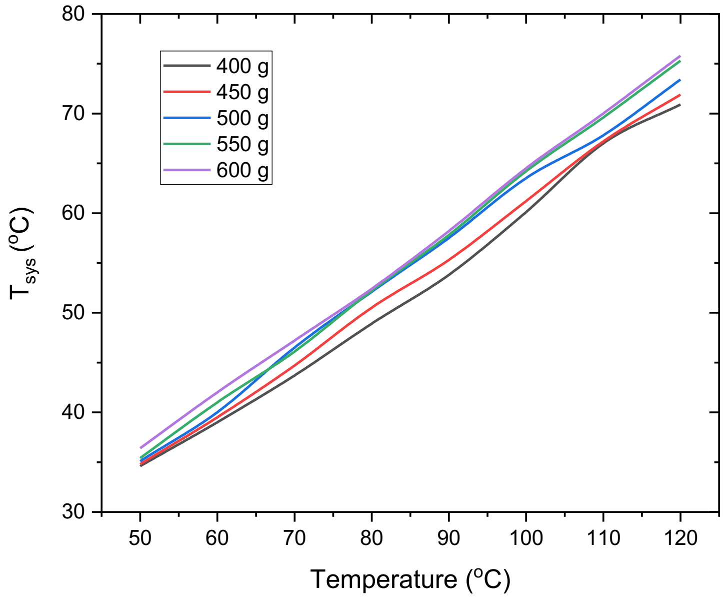

Figure 5. Comparison of the gas temperatures in the system according to gas weights.

As it is known, the temperature values in the graph seen in Fig. 6 were measured from the outer surfaces of the pipe through which the refrigerant flows. These surface temperatures provide input on heat transfer to the environment and approximate information about the refrigerant in the system. The temperature values of the radiator inlet and outlet regions are close to each other. The change in radiator inlet and outlet temperatures is as seen in the graph in Fig. 6.

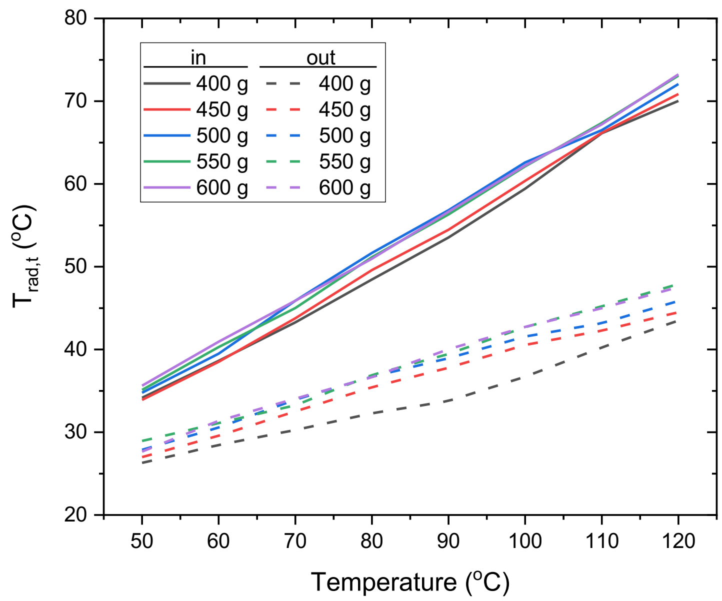

Figure 6. Comparison of radiator inlet and outlet temperatures according to gas weights.

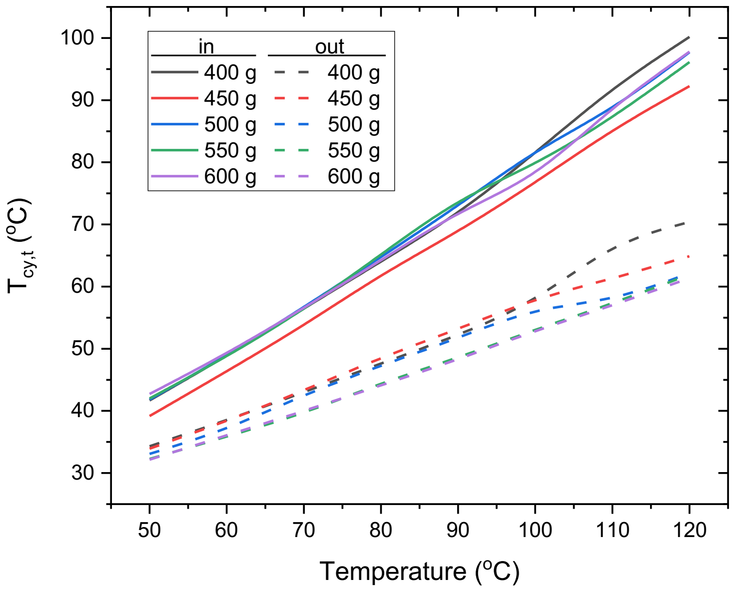

Figure 7. Comparison of cylinder top side outlet and bottom side inlet temperatures according to gas weights**.**

The oil in the cylinder has been heated continuously to a constant temperature to generate cylinder outer surface temperatures at the desired temperature value. For comparisons in different refrigerant masses, it has been seen in all experimental studies that oil temperatures and cylinder outer surface temperatures are heated almost equally. The effect of oil temperature on the temperature change in the cylinder can be seen in Fig. 7. Cylinder top temperatures have the highest temperature values after oil temperatures as a result of the system's operating principle. In the meantime, since the average of the cylinder top and bottom temperatures will be accepted as the cylinder surface temperature, it has an essential effect on the calculation of the heat transfer coefficient of the refrigerant, even if it is near to each other in many experiments.

Depending on the system's working principle, the liquid-gas ratios can give us information about heat transfer. Liquid-gas ratios create differences too small to be seen visually for experiments containing the same amount of refrigerant. For this reason, in Fig. 8, solid models created in the CAD environment and the change of liquid-gas ratios depending on the amount of refrigerant are shown.

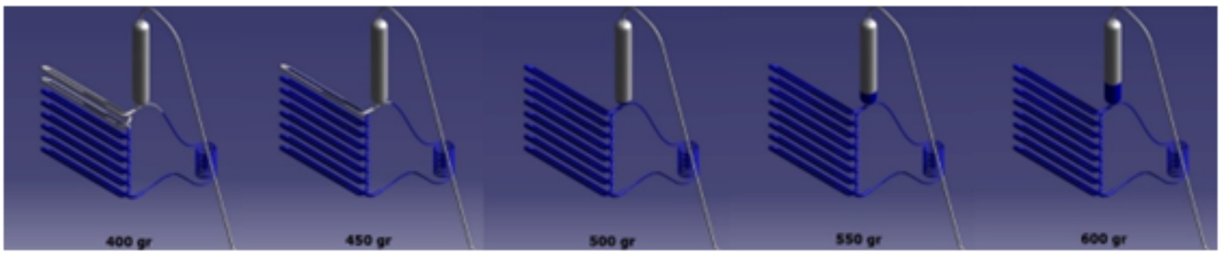

Figure 8. Liquid-gas ratios in the system.

Since no part in the system can cause forced convection, the velocity of the gas in the system calculated based on the test inputs is also of great importance. The refrigerant, which reaches these speeds by only taking the heat, reveals one of the system's advantages. The gas velocities in the system can be examined with the help of the graph shown in Fig. 9.

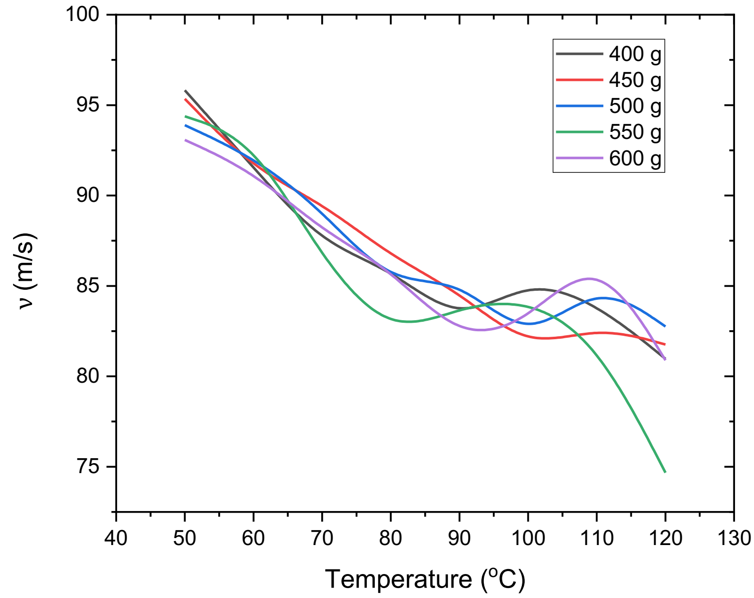

Figure 9. Comparison of the gas velocities according to gas weights.

The total amount of heat withdrawn from the system also has an important place in calculating the convection heat transfer coefficient. Since the surface area where the heat is drawn in the system is the inner surface area of the spiral region, this value will not change according to the experiments. For this reason, the most important parameters affecting the calculation result will be the total heat withdrawn from the system and the top and bottom cylinder temperatures. The total amount of heat drawn from the system obtained after the necessary calculations are made seen in the graph in Fig. 10.

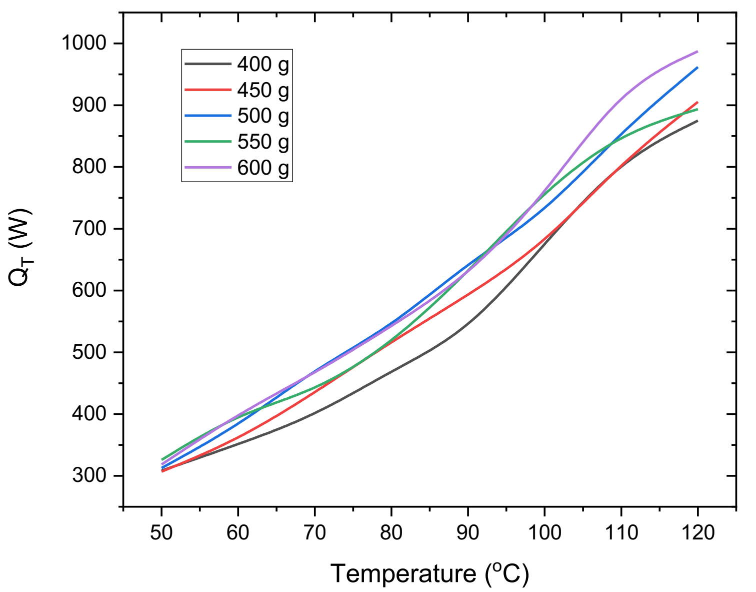

Figure 10. Total amount of heat withdrawn from the system.

The heat transfer coefficient, which will reveal the heat transfer performance of the refrigerant in the created system, is an important feature that gives factual information about whether the system can be used in terms of performance. Compared to other experiments at 600 g of refrigerant and 100°C oil temperature, a very high heat transfer coefficient was obtained. However, considering the engine's operating conditions, it is inevitable to get a good result at higher temperature values. For this reason, as seen from the graph in Fig. 11, the test result containing 550 g of refrigerant in a system with these characteristics is more satisfactory.

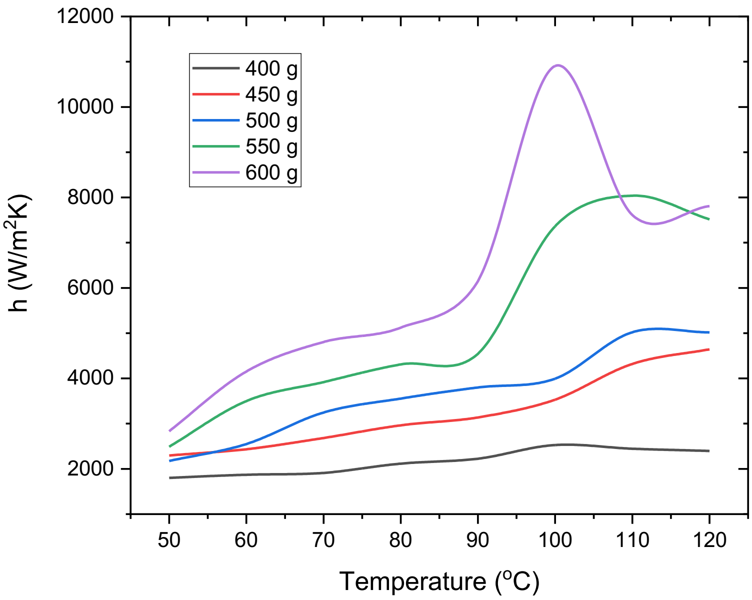

Figure 11. The convection heat transfer coefficients for R134a.

4. Conclusions and Recommendations

This study conducted an experimental investigation of using R134a refrigerant in the engine cooling system. The significant results obtained from the experiments carried out with different refrigerant amounts, and different temperature values can be listed as follows:

In this system, which includes many parameters, the highest heat transfer coefficient value was obtained in the experiment at 100°C containing 600 g of refrigerant. The most efficient result was obtained in the experiment at 120°C containing 550 g of refrigerant.

It is seen that the system is suitable for use in internal combustion engines. The calculated gas velocities show that the refrigerant is transferred naturally, and the heat transfer coefficients reach very high values. It is seen as a high-performance system in drawing the heat from the cylinder walls and transferring it to the radiator. Of course, it will be much more satisfying to set up the system on a real engine and do the necessary experiments.

In addition, experimental studies can be carried out by investigating whether different refrigerants can be used to cool internal combustion engines.

As a result of the successful use of the system in internal combustion engines, there will be some changes in the engine cooling system. The pump will be disabled, but copper pipes will replace the plastic pipes. The necessity of using a fan should be determined by conducting more extensive experiments.

In addition, the availability of gaseous refrigerants as a refrigerant in internal combustion engine cooling systems will allow the system structures to shrink. This will also allow the size of the engine to be reduced. Therefore, the decrease in vehicle weight will indirectly contribute to reducing vehicle production costs and also help to reduce fuel consumption. Reducing fuel consumption will also contribute to the formation of less harmful emissions in terms of both economic benefits and environmental health.

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