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ENERGY AND EXERGY ANALYSIS OF AN AUTOMOBILE AIR CONDITIONING SYSTEM USING REFRIGERANT R134a

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ABSTRACT

This paper deals with the energy and exergy analysis of automobile air conditioning (AAC) system to apply the performance of an AAC system using R134a ($C_2H_2F_4$ / Tetrafloretan) as the refrigerant. Exergy and analysis of seperate components of AC system have been carried out under various operating conditioning. Exergy and energy analysis show that the performance of the system degrades with increasing compressor speed. The coefficient of performance (COP) for R134a system is lower than that for diffirent compressor speed for the same cooling capacity. Furthermore, the COP increases with increasing evaporator load and decreases with increasing compressor speed and condensing temperature. The rate of exergy destruction in each compenent of the AAC system have been found. Exergy destruction increase with increasing compressor speed in the compressor, condenser, evaporator and TXV. The component with the greatest increase in exergy destruction as a result of compressor speed is the compressor itself. The rate of exergy destruction decrease compressor, condenser, evaporator and TXV, respectively.

Keywords: Refrigeration, Automobile air conditioning, Exergy, R134a.

R134A SOĞUTUCU AKIŞKANI KULLANAN OTOMOBİL KLİMALARINDA ENERJİ VE EKSERJİ ANALİZİ

ÖZET

Bu makalede, soğutucu akışkan olarak R134a'nın kullanıldığı bir otomobil klima sisteminde enerji ve ekserji analizi ile uğraşılmıştır. Otomobil klima sisteminin herbir bileşeni için farklı çalışma şartlarında ekserji analizi ortaya konmuştur. Ekserji ve enerji analizleri, artan kompresör devri ile sistem performansının azaldığını göstermiştir. R134a'lı sistemin Soğutma Tesir Katsayısı (STK), aynı soğutma yükü için artan kompresör hızı ile düşmektedir. Ayrıca STK, artan evaporatör yükü ile artmakta ve artan kompresör devri ve yoğuşma sıcaklığı ile azalmaktadır. Otomobil klima sisteminde herbir çevrim elemanında yapılan ekserji yıkımı bulunmuştur. Ekserji yıkımı, artan kompresör devri ile kompresör, kondenser , evaporatör ve TXV'de artmaktadır. Ekserji yıkımındaki en büyük artış, kompresör devrinin bir sonucu olarak komprsörün kendisinde olmaktadır. Exergy yıkımı, sırasıyla kompresör, kondenser , evaporatör ve TXV olmak üzere azalmaktadır.

Anahtar kelimeler: Soğutma, Otomobil kliması, Ekserji, R134a

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1. INTRODUCTION

In this study, an automobile air conditioning (AAC) system charged with R134a refrigerant was operated at various compressor speeds and thermal loads, and a performance analysis for R134a refrigerant was conducted. For this aim, an experimental AAC system consisting of a compressor, an evaporator, a condenser and a thermostatic expansion valve was set up. The compressor was driven by a three-phase electric motor, which was energized using an inverter. Various thermal loads in the range of 1500 and 3060 W were applied to the system by means of electric heaters. The experiments were conducted at the condensing temperatures of 50 ^{o}C and 60 ^{o}C for each thermal load, and at the compressor speeds of 600, 800, 1000, 1200, 1400 rpm for each thermal load-condensing temperature combination. The refrigerant and air temperatures, refrigerant pressures, compressor speed, air velocity passing through the evaporator and thermal load were measured. Using experimental data, an energy and exergy analysis was applied to the system.

The literature concerning experimental performance of AAC systems is very limited due to the fact that AAC is a competitive and technologically oriented industry. Kiatsiriroat and Euakit [1] determined theoretical and experimental performance of an AAC system using R22/R124a/R152 refrigerant mixture.

Ghodbane simulated [2] the performance of AAC systems using some hydrocarbon refrigerants, namely HC152a, HC270, HC290 and HC600a, in terms of COP and compressor discharge temperature. They determined that the HC152a and HC270 systems outperformed the HFC134a system by 11% and 15%, respectively, while HC290 showed only a marginal improvement and R600a yielded lower COPs than HFC134a. However, due to their potential flammability, the use of hydrocarbon refrigerants in AAC systems is considered unsafe unless some extra design precautions are taken.

Bhatti [3] dealt with potential augmentation of HFC134a AAC systems to lower their global warming impact. To this end, he investigated the effects of increasing the compressor isentropic efficiency, increasing the condenser effectiveness, decreasing the air side pressure drop in the evaporator, increasing the condenser air flow and decreasing the cooling load on the COP of the system. Based on the comparisons of the experimental results obtained from a base line HFC134a system and from a realistically enhanced HFC134a system, he remarked that the enhanced system could be the most practical solution to deal with the global warming caused by AAC systems.

Jung et al.[4] performed a computer analysis of some refrigerant mixtures containing HCFC22, HFC134a, HCFC142b, RE170, HC290 and HC600a as possible supplementary/retrofit alternatives for CFC12 in existing AAC systems. They used the results of this analysis for initial screening of possible refrigerant candidates. Then, they determined the experimental performances of the alternative refrigerant mixtures proposed through computer analysis, finding that an HFC134a/RE170 mixture was the best long term candidate to supplement CFC12. However, they did not report any comparison between their theoretical and experimental results.

Lee and Yoo[5] analysed each component of an HFC134a AAC system and developed a simulation model for the whole system by combining the performance analysis programs of the separate components. Their program for the evaporator performance was based on experimental results, and the program for the condenser performance assumed no subcooling at the condenser outlet. They found that the agreement between the results of the simulation model for the whole system and the experimental results was within 7%.

Ratts and Brown [6] experimentally analysed the effect of the HFC134a refrigerant charge level on the performance of an AAC system. To this aim, they determined the individual component losses in an AAC system as a function of refrigerant charge using the second law. They found that the compressor and condenser were the components causing the largest percentage of total losses, while the evaporator and expansion device losses accounted for a smaller percentage of the total losses.

Al-Rabghi and Niyaz [7]retrofitted a CFC12 AAC system to use HFC134a and determined the experimental coefficient of performance (COP) for the system as a function of compressor speed in each refrigerant case. They found that the AAC system using CFC12 had a better COP by 23% than the system using HFC134a.

Brown et al.[8] investigated the performance merits of CO_2 and HFC134a AAC systems using semi-theoretical cycle models. In addition to the standard refrigeration circuit components, namely the compressor, condenser, expansion device and evaporator, their CO_2 system was equipped with a liquid line/suction line heat exchanger. They determined that HFC134a has a better COP than CO_2 with the COP disparity being dependent on the compressor speed and ambient temperature.

Jabardo et al.[9] developed a steady state simulation model for an AAC system consisting of a variable capacity compressor, a micro-channel parallel flow condenser, a thermostatic expansion valve and a plate fin tube evaporator. They tested the validity of the model on an experimental unit. They observed that the deviations between the simulated and experimental results for the cooling capacity, COP and refrigerant mass flow rate as a function of compressor speed were within 5%. However, for the same performance parameters, as a function of the evaporator return air temperature, the deviations of the simulated results with respect to the experimental ones were as high as 18%.

Joudi et al.[10] simulated the performance of an ideal AAC system working with several refrigerants to determine the most suitable alternative refrigerant for CFC12. Their model predicted that the mixture HC290/HC600a was an optimum substitute for CFC12. Afterwards, they compared various performance parameters of an experimental AAC system using CFC12 and the mixture HC290/HC600a as the working fluids. They observed that the compressor power consumption in the HC290/HC600a case was slightly higher than that in the CFC12 case for the same cooling capacity. However, they did not report any comparison between the simulated and experimental results.

Kaynakli and Horuz [11] analysed the experimental performance of an HFC134a AAC system to find optimum operating conditions. They presented some performance parameters such as cooling capacity, compressor power, total power consumption, refrigerant mass flow rate and COP as a function of condensing temperature, evaporator return air temperature, ambient temperature and compressor speed.

Esen [12] analysed the theoretically and experimentally performance of R12 and R134a refrigerants and compressor speed on the performance of automobile air conditioning system. In the theoretical part of the study, a computer simulation program was developed to compare the performances of automobile air conditioning systems using R12 and R134a refrigerants. In the experimental part of the study, an experimental automobile air conditioning system was set up for performance comparison of R12 and R134a.

Tian and Li [13] developed a mathematical model for an HFC134a AAC system with a variable capacity compressor to simulate its steady state performance. Their model determined the effects of compressor speed, ambient temperature and evaporator air flow rate on the evaporating pressure, condensing pressure, cooling capacity and indicated compressor power. They validated the model results on an experimental unit, finding that the deviations between the simulated and measured parameters were within 11%.

Hosoz and Direk [14] dealt with the performance characteristics of an HFC134a AAC system with the feature of operating as an air to air heat pump. For this aim, they developed an experimental system and tested it in both air conditioning and heat pump modes, varying the compressor speed and air temperatures at the inlets of the outdoor and indoor coils. They evaluated the performance of the integrated system in terms of cooling and heating capacities, COP, compressor discharge temperature and the rate of exergy destroyed in each component of the system. They determined that the heat pump operation usually yielded a higher COP and a lower rate of exergy destruction per unit capacity compared to the air conditioning operation, although it provided inadequate heating.

Hosoz and Ertunc [15] analysed the ANN approach has been used for investigating the performance of an AAC system employing an HFC134a vapour compression refrigeration circuit. Utilizing steady state data obtained from the experimental AAC system, an ANN model for the system has been developed. With the use of this model, various performance parameters of the system, namely the compressor power, heat rejection rate in the condenser, refrigerant mass flow rate, compressor discharge temperature and COP, have been predicted and compared with actual ones.

2. TEST EQUIPMENT AND EXPERIMENTAL PROCEDURE

The experimental automobile air conditioning system made up of original components from a compact size passenger vehicle air conditioner is shown in Fig 1. Refrigeration circuit of the system consists of a five cylinder swash plate type compressor, a parallel-flow micro-channel type condenser, a liquid receiver/drier, an internally equilised thermostatic expansion valve and a laminated type evaporator. The AAC system used polyalkylene glycol type compressor oil and was charged with 740 g of R134a.

The compressor was belt-driven by a three-phase 4kW electric motor with anominal speed of 2850 rpm. The motor was energized through a frequency inverter to test the system at various compressor speeds. Normally, cooling capacity of an AAC system is controlled by a thermostat which ceases current flow through the coil of electromagnetic clutch to disengage the compressor shaft from the rotating pulley when desired return air temperature is achieved. Contrary to this common practice, the experimental plant has no thermostat in the control circuit of the clutch in order to test the system in steady state operation with no interruption. Evaporator and condenser fan motors as well as clutch coil were energized by means of direct-current power sources. each capable of providing variable output voltages. This feature allows obtaining a broad range of condensing temperatures regardless of air temperature at the condenser inlet.

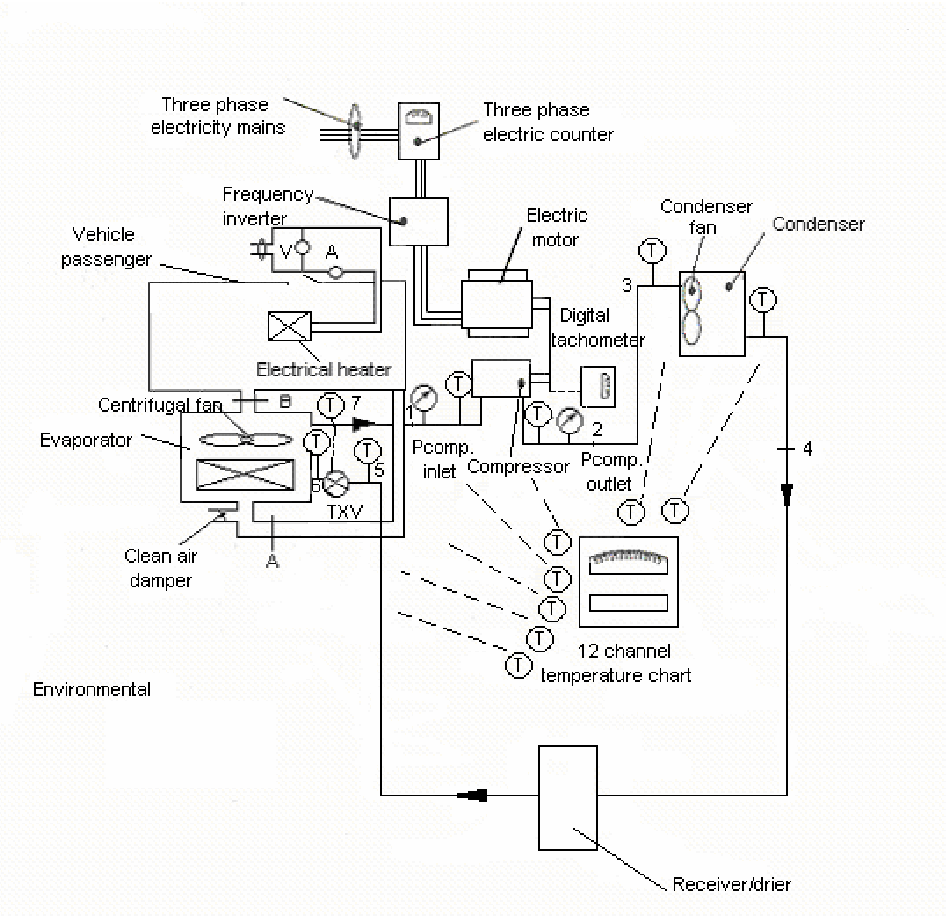

Figure 1. Sketch of the experimental setup and instrumentation.

The evaporator was maintained in its original plastic casing and inserted in an air duct upstream of the simulated passenger compartment. This compartment has a volume of approximately 1.5 m^3 and contains electrical heaters to give the thermal load to the system. The heaters can be controlled between 1500 and 2850 kW with intervals of 450.

The air stream was circulated in a closed circuit consisting of the evaporator, the compartment and the return air duct by means of centrifugal fan.All lines in the refrigeration and air circuits of the system, including the passenger compartment, were thermally insulated by either polyurethane foam or elastomeric insulator.

Fig.1. also indicates the locations where mechanical and electrical measurements were performed. All temperature measurements were accomplished by type K thermocouples. Thermocouples for refrigerant temperature were in direct contact with the outside surface of the aluminium tube sections of the hoses. Both dry bulb and wet bulb temperatures of the air stream at the inlet and outlet of the evaporator were measured. Air temperatures at the outlet of the evaporator were detected at three different locations, and they were averaged to find mean values. Compressor suction and discharge pressures were measured using Bourdon tubes gauges. It was assumed that evaporating and condensing pressures were equal to the measured ones.

Air mass flow rate circulating through the evaporator and the passenger compartment was determined by measuring average air velocity in the return duct by a turbine type anemometer, finding density of the air at the inlet of the evaporator with the dimensions of the return duct. The compressor speed was measured by a photoelectric tachometer. The heat input to the passenger compartment was found by measuring the voltage across the heaters and the current draw.

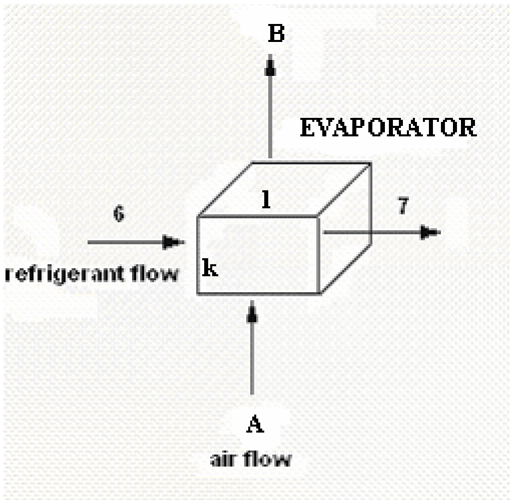

Figure 2. Refrigerant and air flow in evaporator.

The AAC system was tested at constant condensing temperatures of 50 and 60^{o}C. These temperatures were achieved by varying the heat input to the compartment to the desired value, adjusting the speed of the compressor by means of a potentiometer connected to the inverter, and changing the speed of the condenser fan by varying the output voltage of the direct-current power source.

For each condensing temperature, the thermal load in the compartment was kept at 1500 and 2850 W while compressor speed was varied between 600 and 1400 rpm with intervals of 200 rpm. Because the experimental system does not use a thermostat, the speeds over 1400 rpm cause extremely low evaporating temperatures that are not the case in realistic operations. Therefore, the upper limit of the compressor speed was selected as 1400 rpm. Characteristics of the instrumentation is shown in Table 1.

Table 1. Characteristics of the instrumentation.

Parameters	Measuring Instruments	Measuring Space	Accuracy
Temperature	K Type Thermocouple	-50/100 °C	0.3 °C
Pressure	Bourdon Type Manometer	-1/10, 0/30 bar	0.1/0.5 bar
Compressor speed	Digital Tachometer	10/100000 rpm	%2

Table 1. Characteristics of the instrumentation. (Devamı)

Parameters	Measuring Instruments	Measuring Space	Accuracy
Voltage	Analog Voltmeter	0/400 V	%1
Current	Analog Ampermeter	0/20 A	%1
Air Speed	Analog Anemometer	0.1/15 m/s	%3

3. PERFORMANCE ANALYSIS

Because the experimental air conditioning system operates with 100% recirculated air and has a good thermal insulation, the heat input given to the passenger compartment by means of electrical heaters must be removed by the refrigerant passing through the evaporator. Therefore, referring to Fig.1 and Fig. 2 evaporator load, i.e. cooling capacity of the system, can be evaluated for both the heaters and air sides, and expressed by

$$\dot{Q}_e = VI \cong \dot{m}_a\left[\left(h_a + wh_g\right)_A - \left(h_a + wh_g\right)_B - \left(w_A - w_B\right)h_f\right] = \dot{m}_a\left(h_A - h_B\right) = \dot{m}_r\left(h_7 - h_6\right) \qquad (1)$$

As seen in Eq. (1), heaters side calculations utilize the results of voltage and current measurements. Air side calculations, on the other hand, are based on mass flow rate and specific enthalpies of the moist air at the inlet and outlet of the evaporator, and enthalpy of the condensate leaving the evaporator. During the test, capacity deviations between two sides were usually within 5%, and therefore, only heaters side results were used as the evaporator load. Thermodynamic properties of the air and R134a were evaluated using o software package for refrigeration [16].
After finding the evaporator load, the refrigerant mass flow rate can be determined from:

$$\dot{m}_r = \frac{\dot{Q}_e}{h_7 - h_6} \qquad (2)$$

Condenser heat rejection rate,

$$\dot{Q}_{cond} = \dot{m}_r\left(h_3 - h_4\right) \qquad (3)$$

Assuming that the compressor is adiabatic, compressor power can be written as:

$$\dot{W}_{comp} = \dot{m}_r\left(h_2 - h_1\right) \qquad (4)$$

The energetic performance of the system is found by evaluating its coefficient of performance, defined as the ratio between cooling capacity and compressor power:

$$COP = \dot{Q}_e / \dot{W}_{comp} \qquad (5)$$

Neclecting the heat transfer with the environment, the rate of exergy destruction in the compressor can be evaluated as:

$$\dot{E}_{d,comp} = \dot{m}_r T_0\left(s_2 - s_1\right) \qquad (6)$$

The rate of exergy destruction in the condenser including destructions in the discharge and liquid lines due to heat loss to the ambient air can be expressed as:

$$\dot{E}_{d,cond} = \dot{m}_r T_0 \left[\left(s_5 - s_2 \right) - \frac{\left(h_5 - h_2 \right)}{T_0} \right] \tag{7}$$

Assuming that expansion process is adiabatic, the rate of exergy destruction in the expansion valve can be determined from:

$$\dot{E}_{d,txv} = \dot{m}_r T_0 \left(s_6 - s_5 \right) \tag{8}$$

The rates of exergy destruction in the evaporator and suction line are obtained from the following equations:

$$\dot{E}_{d,e} = \dot{m}_r T_0 \left[\left(s_7 - s_6 \right) - \frac{\left(h_7 - h_6 \right)}{T_A} \right] \tag{9}$$

$$\dot{E}_{d,sl} = \dot{m}_r T_0 \left[\left(s_1 - s_7 \right) - \frac{\left(h_1 - h_7 \right)}{T_0} \right] \tag{10}$$

Finally, the rate of total exergy destruction in the ferrigeration circuit of the system can be determined from;

$$\dot{E}_{d,tot} = \dot{E}_{d,comp} + \dot{E}_{d,cond} + \dot{E}_{d,txv} + \dot{E}_{d,evap} + \dot{E}_{d,sl} \tag{11}$$

4. EXPERIMENTAL INDEFINETNESS AND ERROR ANALYSIS

In order to show reliability of experimental results, it has great importance to point out indefinetness of experimental set up. Experimental indefinetness and error analysis are obtained by Eq. (12)

$$\Gamma_{indefinetness} = \frac{\delta\Gamma}{\Gamma} \tag{12}$$

Hence, indefinetness of performance parameters are proved in whole system;

Indefinetness analysis of air mass flow rate:

Air mass flow rate is obtained by:

$$\dot{m}_a = \overline{V}_a A_a \rho_a \tag{13}$$

Consequently, calculation of indefinetness of air mass flow rate has been calculated by Eq. (14)

$$\delta \dot{m}_a = \pm \sqrt{\left(\frac{\partial \dot{m}_a}{\partial V_a} \delta \overline{V}_a \right)^2 + \left(\frac{\partial \dot{m}_a}{\partial A_a} \delta A_a \right)^2 + \left(\frac{\partial \dot{m}_a}{\partial \rho_a} \delta \rho_a \right)^2} \tag{14}$$

Dimension of the canal is;

$$A_a = kl \tag{15}$$

Explained by Eq. (15)

Then,

$$\delta A_a = \pm\sqrt{\left(\frac{\partial A_a}{\partial k}\delta k\right)^2 + \left(\frac{\partial A_a}{\partial l}\delta l\right)^2} \tag{16}$$

As a result, $A_a = kl = 0.0625m^2$, $\delta k = 0.001$ m, $\delta l = 0.001$ m and $\delta A_a = 3.53.10^{-4} m^2$.

According to errors of temperature and humidity measurement $\delta\rho$ =0.002 kg/m^3.

Air velocity is:

$$\bar{V}_a = 0.6427 + 0.1423V_{battery} + 0.0005V_{battery} \tag{17}$$

obtained by Eq.(17)
According to battery voltage, values of air velocity is calculated as;

$\bar{V}_a = 2.358 m/s$.

According to error of voltage and air velocity measurement,

$\delta\bar{V}_a$ is obtained as $0.0266 m/s$.

As a result, $\delta\dot{m}_a = \pm 0.001028 \quad kg/s$, $\dot{m}_{a\max} = \dot{m}_a + \delta\dot{m}_a$, $\dot{m}_{a\max} = 0.1722 \quad kg/s$.

Indefinetness analysis of refrigerant mass flow rate:

Equation (1) can be expressed by;

$$\dot{m}_r = \frac{\dot{m}_a\left(h_A - h_B\right)}{\left(h_7 - h_6\right)} \tag{18}$$

Eq.(18). Hence, $\dot{m}_r$ is calculated as $\dot{m}_r = 0.02115 \ kg/s$.

As a result, $\delta\dot{m}_r$ is shown as,

$$\delta\dot{m}_r = \pm\sqrt{\left(\frac{\partial\dot{m}_r}{\partial\dot{m}_h}\delta\dot{m}_h\right)^2 + \left(\frac{\partial\dot{m}_r}{\partial h_A}\delta h_A\right)^2 + \left(\frac{\partial\dot{m}_r}{\partial h_B}\delta h_B\right)^2 + \left(\frac{\partial\dot{m}_r}{\partial h_7}\delta h_1\right)^2 + \left(\frac{\partial\dot{m}_r}{\partial h_4}\delta h_4\right)^2} \tag{19}$$

According to Eq. (19), $\delta\dot{m}_r$ is.Consequently, Q_{cond}, W_{comp} and *COP* has been calculated from values of $\dot{m}_r$.

Indefinetness analysis of condenser heat rejection rate:

According to Eq.(3), δQ_{cond} is obtained as Eq.(20).

$$\delta Q_{cond} = \pm\sqrt{\left(\frac{\partial Q_{cond}}{\partial \dot{m}_r}\delta \dot{m}_r\right)^2 + \left(\frac{\partial Q_{cond}}{\partial h_2}\delta h_2\right)^2 + \left(\frac{\partial Q_{cond}}{\partial h_3}\delta h_3\right)^2} \tag{20}$$

As a result, $\delta Q_{cond} = \pm 0.0221 \quad kW$.

Indefinetness analysis of compressor power:

According to Eq. (4), δW_{comp} is obtained as Eq. (21).

$$\delta W_{comp} = \pm\sqrt{\left(\frac{\partial W_{comp}}{\partial m_r}\delta \dot{m}_r\right)^2 + \left(\frac{\partial W_{comp}}{\partial h_2}\delta h_2\right)^2 + \left(\frac{\partial W_{comp}}{\partial h_1}\delta h_1\right)^2} \tag{21}$$

As a result $\delta W_{comp} = \pm 0.0104 kW$.

Indefinetness analysis of refrigeration capacity:

According to Eq. (1), δQ_e is obtained as Eq. (22).

$$\delta Q_e = \pm\sqrt{\left(\frac{\partial Q_e}{\partial V}\delta \overline{V}\right)^2 + \left(\frac{\partial Q_e}{\partial I}\delta I\right)^2} \tag{22}$$

As a result, $\delta Q_e = \pm 0.00902 \, kW$.

Indefinetness analysis of coefficient of performance:

According to Eq. (5), δCOP is obtained as Eq. (23).

$$\delta COP = \pm\sqrt{\left(\frac{\partial COP}{\partial Q_e}\delta Q_e\right)^2 + \left(\frac{\partial COP}{\partial W_{comp}}\delta W_{comp}\right)^2} \tag{23}$$

As a result,

$\delta COP = \pm\, 0.02575$.

Results of indefinetness values in the Equations of (14), (16), (19), (20), (21), (22) and (23) has been given Table 2.

Table 2. Values of indefinetness analysis.

Calculated of ındefinetness	Values
Air flow rate	±0.1722 kg/m^3
Air canal dimension	±0.000353 m^2
Refrigerant flow rate	±0.0001 kg/m^3
Condenser heat rejection rate	±0.0221 kW
Compressor power	±0.0104 kW
Refrigeration	±0.00902 kW
Coefficient performance	±0.0217

5. RESULTS AND DISCUSSION

In this study, the effect of R134a refrigerant and compressor speed on the performance of automobile air conditioning systems was experimentally investigated. In the experimental study, a experimental AC system was set up to find the performances of automobile air conditioning systems using R134a refrigerant.[18,19]

Figure 3 the compression ratio as a function of the compressor speed for R134a conditioning systems at 1500 and 2400W cooling loads. Compression ratio increases with increases with increasing compressor speed and condensing temperature. Compression rate of the system with R134a system for 60°C condenser temperature is higher than R134 system for 50°C condenser temperature for same cooling capacity.

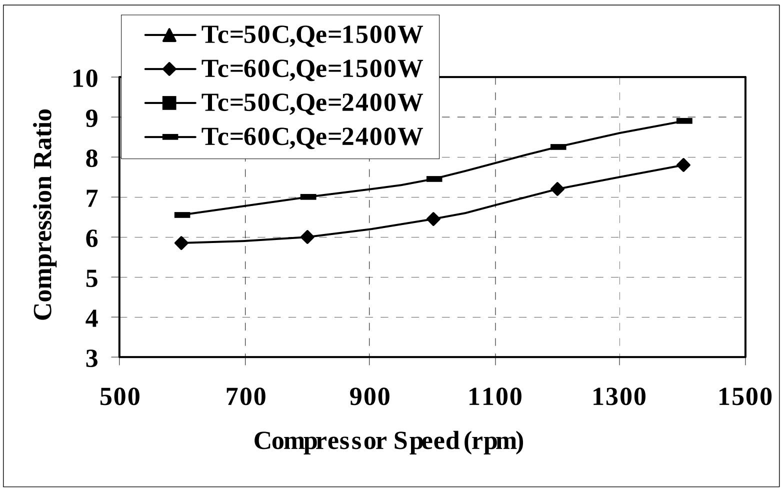

Figure 3. Compression ratio as a function of the compressor speed.

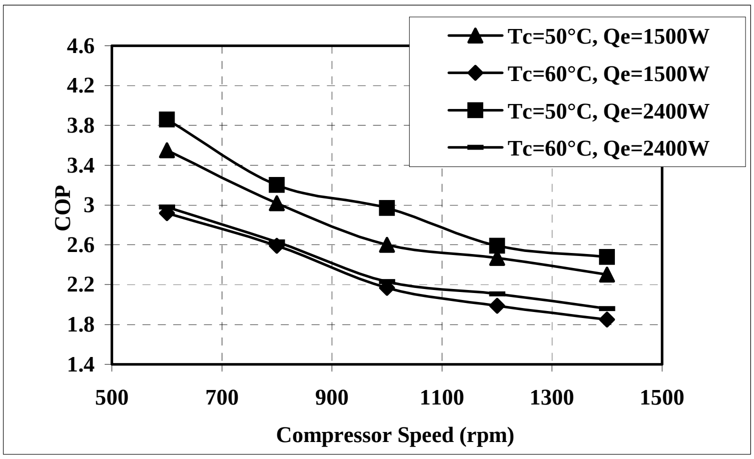

Figure 4. COP as a function of the compressor speed.

Figure 4 shows the COP as a function of compressor speed for R134a air conditioning systems at 1500 and 2400W cooling load. COP of the system with R134a system for 60°C condenser temperature is %28-30 lower than R134a system for 50°C condenser temperature at 2400 cooling load and 1000 rpm. The COP increases with increasing evaporator load and decreases with increasing compressor speed and condensing temperature

Figure 5 the mass flow rate as a function of he compressor speed for R134a air conditioning systems at 1500 and 2400W cooling load. Mass flow rate of the systems with R134a system for 60°C condenser temperature is %4-5 higher than R134a system for 50°C condenser temperature at 1500W cooling and 600 rpm. And also, compressor power reduces with compressor speed.

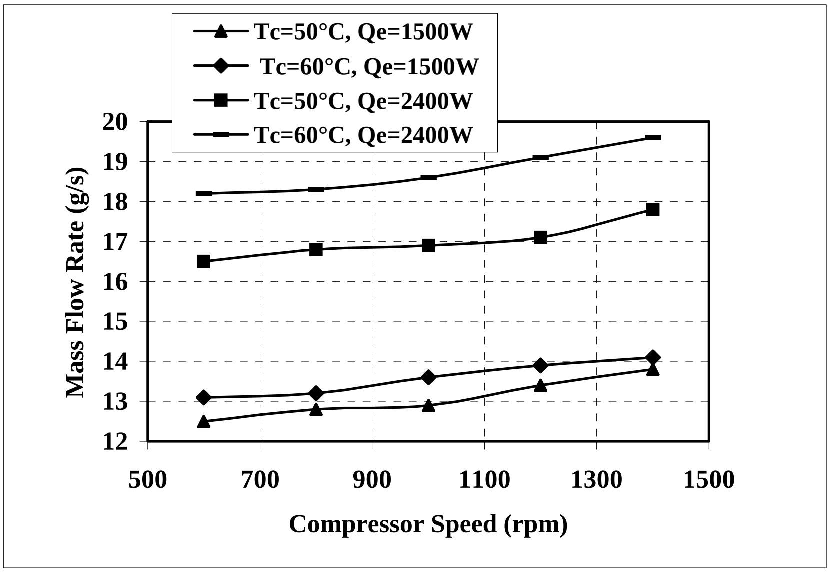

Figure 5. Refrigerant mass flow rate as a function of the compressor speed.

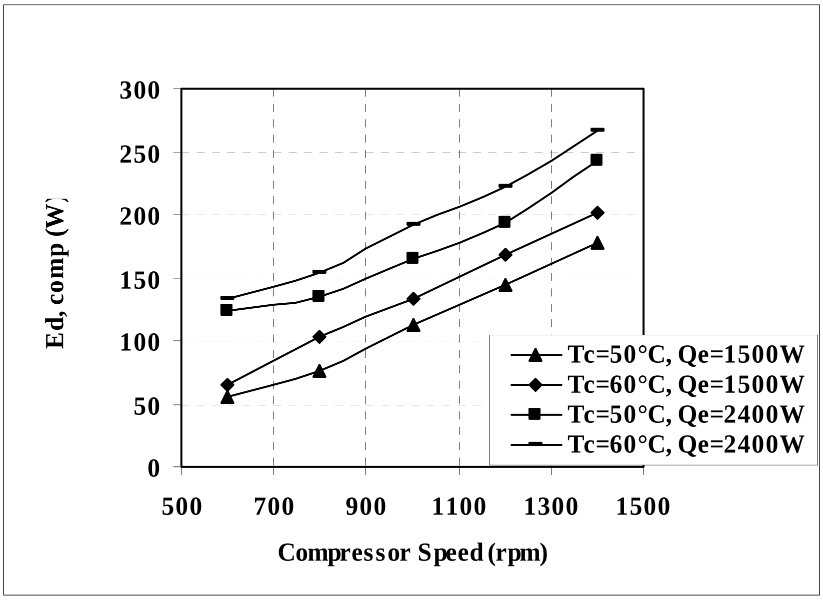

Figure 6. Rate of exergy destroyed in the compressor.

Figure 6 indicates the rate of exergy destroyed in the compressor. Exergy destruction in the adiabatic compressor is caused by gas friction, mechanical friction of the moving parts and internal heat transfer. More specifically, exergy destruction in the compressor is due to the warming of the refrigerant at the inlet of the cylinder, the imperfection of the stuffing between the piston and the cylinder and losses in the compressor valves [17]. The extent of irreversibility in the compressor, which originates from the sources cited above, is a function of evaporator load, condensing temperature and compressor speed. It is seen that the higher the condensing temperature and evaporator load, the higher the rate of the exergy destroyed in the compressor. Morever, the exergy destruction in the compressor increases with compressor speed.

Figure 7 depicts the rate of exergy destruction in the condenser and liquid line as a function of compressor speed. It is assumed that there is not any pressure drop in the refrigeration circuit compenents and refrigerant lines except the expansion valve. Therefore, exergy destruction in the condenser and liquid line is due to irreversibilities associated with heat transfer between the refrigerant and air streams. Results show that, similar to the tendencies of $\dot{E}_{d,comp}$ curves, the rate of exergy destroyed in the condenser and liquid line increases with increasing condensing temperature, evaporator load and compressor speed.

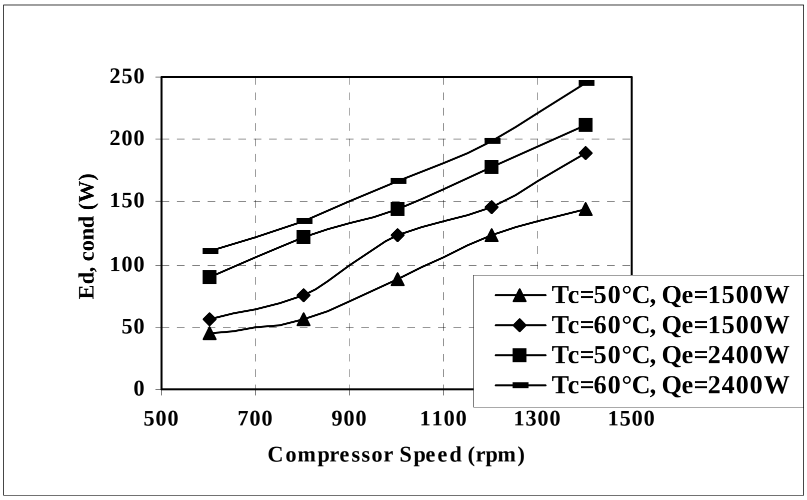

Figure 7. Rate of exergy destroyed in the condenser.

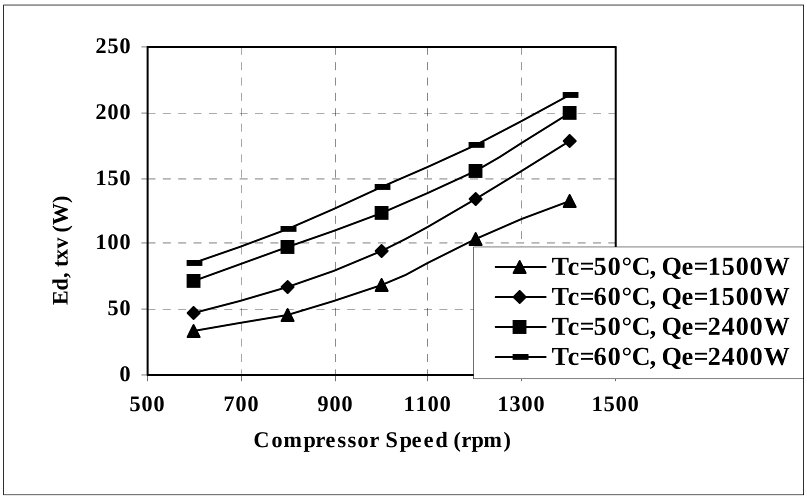

Figure 8. Rate of exergy destroyed in the thermostatic expansion valve.

Figure 8 exhibits the rate of exergy destruction in the expansion valve as a function of compressor speed. Assuming no heat transfer to or from the environment, the rate of exergy destruction in the expansion valve is due to refrigerant friction accompanying the expansion across the valve. It is seen that the higher the condensing temperature, evaporator load and compressor speed, the higher the rate of exergy destruction in the expansion.

The total rate of exergy destruction in the evaporator and suction line as a function of compressor speed is shown in Figure 9. Since it is assumed that there is no pressure drop in the heat exchangers and refrigerant lines, exergy destruction in the evaporator is due to heat transfer between the air and refrigerant streams. Results show that condensing temperature does not affect exergy destruction in the evaporator as strongly as it affects destructions in other compenents whereas evaporator load and compressor speed arethe two factors determining exergy destruction in evaporator.

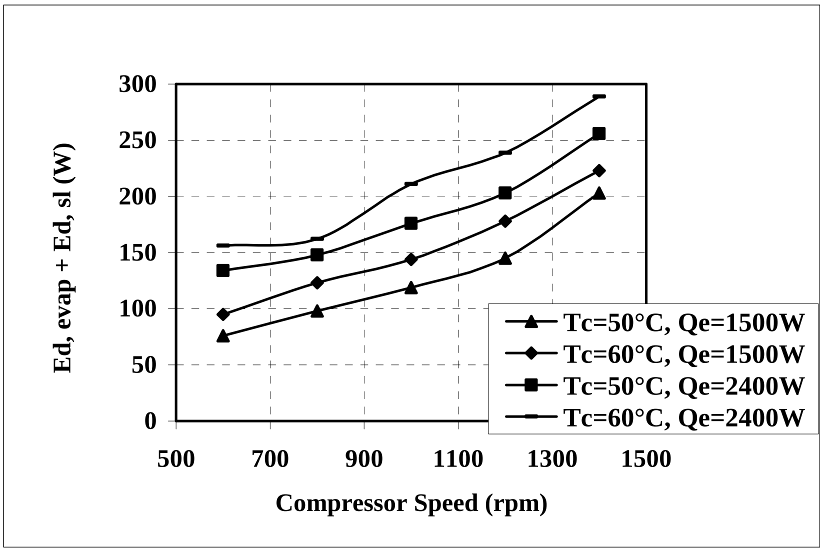

Figure 9. Rate of exergy destroyed in the evaporator and suction line.

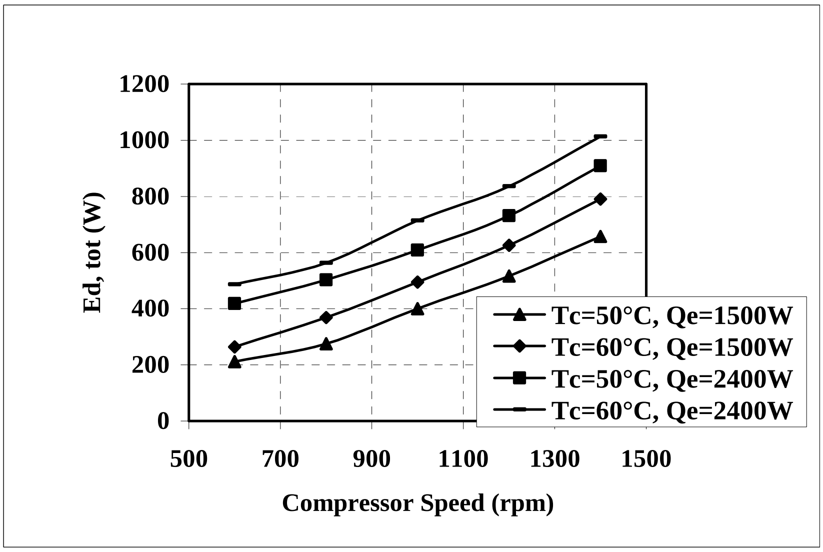

Figure 10. Rate of exergy destroyed in the entire refrigeration circuit.

The rate of total exergy destroyed in the entire refrigeration circuit as a function of compressor speed is presented in Figure 10. Similar to the trends in previous graphs, the higher the rate of total exergy destruction in the refrigeration circuit compenents of the ACC system. Furtermore, as observed before, total exergy destruction increases with compressor speed. By inspecting Figure 4-9, it is seen that the highest contribution to the total exergy destruction comes from the condenser while evaporator, expansion valve and compressor also make significant contribution in decreasing order. Therefore, further studies must be concentrated on the heat exchangers, especially on the condenser to improveenergetic performance of the system. For example, the condenser must be redesigned to have have a larger heat transfer area and a higher air flow rate. Since the rate of total exergy destruction in the system is linked COP, an attempt to decrase exergy destruction will eventually increase COP for the system.

The ratio of the rate of total exergy destruction in the entire refrigeration circuit to the evaporator load is depicted in Figure 11. Results show that this ratio increasing condensing temparaure and compressor speed while evaporator load has almost no effect on it. By inspecting COP curves presented in Fig 4, it is seen that the COP for the system decreases with increasing compressor speed wheares the rate of total exergy destruction per unit evaporator load increases with it. Therefore, the ACC system can be operated more efficiently at lower compressor speeds and evaporator loads.

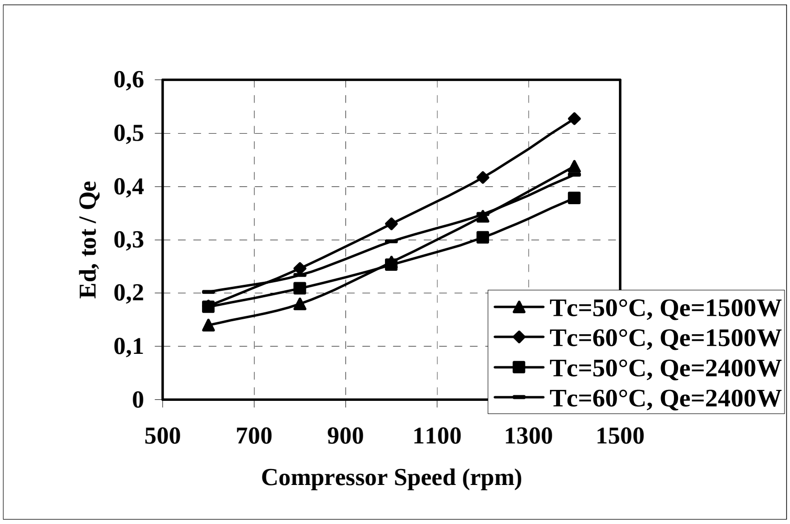

Figure 11. Rate of exergy destroyed in the entire refrigaration circuit to the evaporator load.

5. CONCLUSIONS

In the experimental part of the study, an experimental AAC system was set up for performance analysis of R134a. The major components of the AAC system are a compressor, an evaporator, a condenser and a thermostatic expansion valve (TXV). The compressor was belt-driven by a three-phase electric motor and compressor speed was controlled by changing the frequency of the supply voltage with an inverter. Cooling loads of 1500, 1950, 2400, 2850 Watts were given to the system by means of electric heaters. Experiments were conducted at several condensing temperatures using both refrigerants. The automobile air conditioning system was run at compressor speeds of 600, 800, 1000, 1200, 1400 rpm for each cooling load. Refrigerant temperatures and air dry and wet bulb temperatures were measured at the selected points on the experimental setup using K-type thermocouples. The compressor and electric motor speeds were measured by a digital tachometer. Refrigerant pressures were measured with Bourdon tube manometers. In addition, air speeds at the condenser and evaporator as well as power input to the electric motor of the compressor were measured. Experimental data was obtained at various evaporating temperatures for each condensing temperature and compressor speed. Using experimental data, an energy and exergy analysis was applied to the system. Experimental results indicate that R134a refrigerant show quite perfect thermal performance. COPs for R134a system are acceptable a value for an identical cooling capacity**.** Another important result is that R134a has a low mass flow rate for cooling capacity. Therefore R134a charge is quite low. Experimental results were presented in graphs.

An exergy analysis of an R134a ACC system has been performed using data acquired in steady-state test runs of an experimental system. It is possible to draw the following conclusions from this analysis.

- The COP for the ACC system increases with increasing evaporator load and with decreasing compressor speed and condensing temperature.
- The rate of exergy destruction in each compenent of the refrigeration circuit increases with increasing evaporator load and compressor speed while it decreases with decreasing condensing temperature.
- The ratio of the rate of total exergy destruction to the evaprator load increases with increasing compressor speed and condensing temperature whereas the evaporator load does not have an appreciable effect on it.
- The components making the highest contribution to the total exergy destruction in the refrigeration circuit is compressor. This means that further studies must be concentrated on compressor to improve energy performance of the system.

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REFERENCES

1. T. Kiatsiriroat, T. Euakit, “Performance analyses of an automobile air conditioning system with R/22/R124/R152A Refrigerant.”, Applied Thermal Engineering, 17(11): 1085–1097, 1997.
2. M. Ghodbane, “An investigation of R152a and hydrocarbon refrigerants in mobile air conditioning”, SAE Technical Papers, Paper No. 1999-01-0874. Society of Automotive Engineers, 1999.
3. M.S. Bhatti, Enhancement of R-134a automotive air conditioning system. SAE Technical Papers, Society of Automotive Engineers, Paper No. 1999-01-0870, 1999.
4. D. Jung, B. Park and H. Lee,”Evaluation of supplementary/retrofit refrigerants for automobile air-conditioners charged with CFC12”, Int J. Refrigeration, 22 ,pp. 558–568, 1999.
5. G.H. Lee and J.Y. Yoo, “Performance analysis and simulation of automobile air conditioning system”, Int J Refrigeration, 23 ,pp. 243–254, 2000.
6. E.B. Ratts and J.S. Brown, “An experimental analysis of the effect of refrigerant charge level on an automotive refrigeration system”, Int J Therm Sci, 39,pp. 592–604, 2000.
7. O.M. Al-Rabghi and A.A. Niyaz, “Retrofitting R-12 car air conditioner with R-134a refrigerant”, Int J Energy Res 24 , pp. 467–474, 2000.
8. J.S. Brown, S.F. Yana-Motta and P.A. Domanski, “Comparative analysis of an automotive air conditioning system operating with CO_2 and R134a”, International Journal Refrigeration, 25, pp. 19–32, 2002.
9. J.M.S. Jabardo, W.G. Mamani and M.R. Ianella, “ Modelling and experimental evaluation of an automotive air conditioning system with a variable capacity compressor”, Int J Refrigeration, 25 , pp. 1157–1173, 2003.
10. K.A. Joudi, A.S. Mohammed and M.K. Aljanabi, “Experimental and computer performance study of an automotive air conditioning system with alternative refrigerants”, Energy Conversion Management, 44, pp. 2959–2976, 2003.
11. O. Kaynakli and I. Horuz, “An experimental analysis of automotive air conditioning system”, Int Commun Heat Mass Transfer, 30, pp. 273–284, 2003.
12. D.O. Esen, “Experimental analysis of the effect of R12 and R134a refrigerants and compressor speed on the performance of the refrigeration cycle of automobile air conditioning”, PhD Thesis, University of Kocaeli, Turkey,2005.
13. C. Tian and X. Li, “Numerical simulation on performance band of automotive air conditioning system with variable displacement compressor”, Energy Conversion Management, 46 , pp. 2718–2738, 2005.
14. M. Hosoz and M.Direk, “Performance evaluation of an integrated automotive air conditioning and heat pump system”, Energy Conversion and Management, 47, 5, pp. 545-559, 2006.
15. M. Hosoz and H.M. Ertunc, “Artificial neural network analysis of an automotive air conditioning system” Energy Conversion and Management, 47, Issues 11-12, pp. 1574-1587, 2006.
16. “A collection of simulation tools for refrigeration”, CoolPack, 2004. www.et.dtu.dk/CoolPack
17. C. Aprea, F.D. Rossi, A. Greco and Renno C., “Refrigeration plant exergetic analysis varying the compressor capacity”, International Jounal Energy Research, 27: 653–669, 2003.
18. D.O Esen and M. Hosoz The effect of compressor speed on the performance of automobile air conditioning systems”, 1st International Vocational and Technical Education Technologies Congress MTET’05, İstanbul, September 5–7, 2005.

19. D.O. Esen and M. Hosoz, Experimental analysis of an automobile air conditioning system", 1st Cappadocia International Mechanical Engineering Symposium (CMES'04), Cappadocia/Urgup, September 5–7, 2004.
20. Moran M.J. and Shapiro H.N. Fundamentals of engineering thermodynamics. New York: John&Sons.; 2000.