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Irreversibility Analysis of a Minibus Air-Conditioner for Different Condensation Pressures

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Abstract

In this study, a minibus air conditioning (MAC) set using R134a was tested and evaluated in the Adana High Vocational School, Çukurova University. The condensation pressure was gradually increased from 650 kPa to 770 kPa. The indoor and outdoor temperatures were kept constant at 20 ºC and 22 ºC, respectively. The cooling capacity of the MAC is 6000 kcal/h at 35 ambient temperature. The MAC consists of four main elements; a compressor, a condenser, an expansion valve and an evaporator. The purpose of this study is to demonstrate how the irreversibility analysis is performed. For this aim, experiments were carried out for different condensation pressures at constant ambient temperature in order to determine the rates of the exergy transfer and the entropy generation within the all components and the MAC system. In addition to this, the rational exergy efficiency of the all components and the whole system were calculated. Increasing condensation pressure caused 8.3% increase in both the rate of entropy generation and the irreversibility rate for the whole system. Besides, the rational exergy efficiency of the whole system was approximately 24%.

Keywords: Exergy, Exergy destruction, Entropy generation, Irreversibility, Minibus air-conditioner

Farklı Yoğuşma Basınçları için Minibüs Klimasının Tersinmezlik Analizi

Öz

Bu çalışmada, R134a kullanılan bir minibüs klima seti Çukurova Üniversitesi Adana Meslek Yüksekokulunda test edildi ve değerlendirilmiştir. Yoğuşma basıncı kademeli olarak 650 kPa'dan 770 kPa'ya yükseltilmiştir. İç ve dış hava sıcaklıkları sırasıyla 20 ºC ve 22 ºC sıcaklıklarında sabit tutulmuştur. Minibüs klimasının 35 çevre sıcaklığındaki soğutma kapasitesi 6000 kcal/h'tır. Minibüs kliması dört ana elemandan meydana gelir; kompresör, yoğuşturucu, genleşme elemanı ve buharlaştırıcı. Bu çalışmanın amacı tersinmezlik analizinin nasıl yapıldığını göstermektir. Bu amaçla, minibüs klimasının tüm bileşenlerdeki ve minibüs klimasındaki ekserji transferi ve entropi üretim hızını saptamak amacıyla farklı buharlaşma basınçlarında ve sabit çevre sıcaklıklarında deneyler yapılmıştır. Buna ek olarak, tüm bileşenlerin ve tüm sistemin ekserji verimi hesaplanmıştır. Yoğuşma basıncının artması, hem sistemin

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entropi üretim hızının hem de tersinmezlik hızının %8,3 artmasına sebep olmuştur. Ayrıca tüm sistemin rasyonel ekserji verimi yaklaşık olarak %24'tür.

Anahtar Kelimeler: Ekserji, Ekserji yıkımı, Entropi üretimi, Tersinmezlik, Minibüs kliması

1. INTRODUCTION

In order to meet the short distance transportation needs of people in the city, it prefers public transport such as buses and minibuses. The need for public transport is increasing day by day and therefore the number of these vehicles is increasing rapidly. With growing numbers of vehicles, energy efficiency becomes more important in public transportation [1]. People spend a lot of time in buses or minibuses that have an important place in public transportation. For this reason, the comfort conditions in the vehicle must be made suitable. Minibus air conditioners are used in order to adapt the ambient air to the comfort conditions in the minibus [2].

Air conditioner located in in the public transport vehicles ensure cooling and dehumidifying of the humid and hot air in order to provide thermal comfort and an acceptable indoor air quality for passengers [3]. Efficient use of these air conditioners is extremely important in terms of energy consumption.

The first law of thermodynamics (or energy analysis) is a very common method in evaluating the performance of thermal systems [4]. However, examination of exergy analysis or irreversibility analysis reveals more realistic results than energy analysis. Exergy analysis finds out the magnitudes and locations of inefficiencies of the cycles [5]. This leads us to improve the efficiency of the system.

Liang and Kuehn [6] analyzed a water to water source heat pump system using R22 on the law of energy and mass conservations in order to evaluate the irreversibilities of the condenser, evaporator and reciprocating compressor. They concluded that an increment in the heat transfer coefficient for the condenser and evaporator units would significantly reduce the irreversibility, despite an increase in pressure drop.

Sahin et al. [7] applied an exergy analysis to three different organic Rankine cycle models using different refrigerants at different condenser inlet pressure, turbine inlet temperature and turbine inlet pressure. They determined the exergy of the system and the irreversibility values of the components per unit mass for R600a, R290 and R152a fluids. They also demonstrated the effect of different fluids on the irreversibility values of the components by using MATLAB and EES software for exergy analysis of organic Rankine cycle.

Yataganbaba et al. [8] carried out irreversibility analysis to a two-evaporator vapor compression refrigeration cycle using R407C, R410A, R404A and R134a in order to investigate the effect of the evaporation and condensation temperatures on the exergy destruction of the system components. They emphasized that the exergy destruction rate of the components was affected by the changes in temperature of the evaporation and condensation.

Tosun et al. [9] performed an exergy analysis to improve the bus air conditioning system design. They evaluated exergy destructions and exergy efficiency of components and entire system for different air mixing rates and seasons. They determined the exergy destructions within the evaporator, the condenser and the expansion valve as 2.78, 2.61 and 0.99 kW, respectively and calculated the maximum exergy destruction value as 6.96 kW within the compressor unit.

In the present study, a MAC system was tested and evaluated exergetically in terms of its performance. The condensation pressure was gradually increased from 650 kPa to 770 kPa while the evaporation pressure increased from 210 kPa to 330 kPa. The indoor and outdoor temperatures were kept constant at 20 °C and 22 °C, respectively. Thus, an irreversibility analysis was applied for different condensation and evaporation pressures in order to determine the rates of the exergy transfer and the entropy generation within the all components and the MAC system.

2. SYSTEM DESCRIPTION

The MAC system was installed in Çukurova University Adana High Vocational School. Schematic representation of the experimental set is shown in Figure 1.

Figure 1. The MAC set

The experimental set operates on a mechanical vapor compression cycle. This cycle is most commonly used cycle for refrigeration and air-conditioner applications. The vapor compression refrigeration cycle consists four main elements as a compressor, a condenser, an expansion valve and an evaporator (Figure 2).

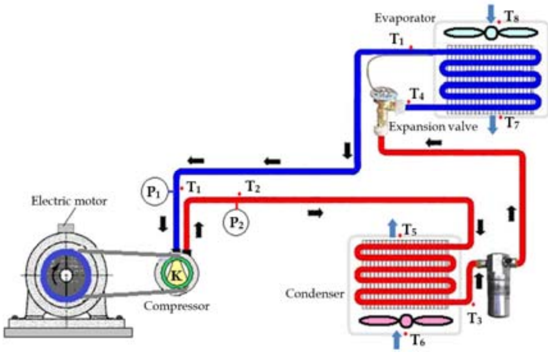

Figure 2. Schematic representation of MAC set

Refrigerant is compressed to condenser in a form of superheated vapor by compressor that driven by electric motor. The temperature of the refrigerant in superheated vapor state is higher than the ambient temperature. Refrigerant is cooled by heat rejection to the surroundings in condenser at high pressure and temperatures. Refrigerant leaves at high pressure as compressed liquid, then it is expanded to the evaporation pressure by passing through a throttle valve. The temperature of the refrigerant is lower than the temperature of the interior air of the minibus after expansion process. The refrigerant enters the evaporator as low quality saturated mixture and it evaporates by heat removal from interior air of the minibus. At the end of the evaporation process, the refrigerant is in superheated vapor phase and reenters the compressor, completing the cycle.

Technical specifications of the MAC were shown in Table 1.

Table 1. Technical specifications of the MAC set

Definition	**Properties**
Refrigerant	R134a
Amount of refrigerant	1.5 ∓ 0.05 kg
Cooling capacity	6000 kcal/h
Heating capacity	8650 kcal/h
Evaporator fan flow rate	1200 m³/h
Condenser fan flow rate	3100 m³/h
Type of expansion valve	Externally equalized TXV

3. THEORETICAL ANALYSIS

In order to investigate the irreversibility and the exergy efficiency of each components of the MAC, exergy and entropy balances were applied. The following assumptions were made in the calculations:

- All processes are steady state.
- The change of kinetic and potential energies of the entire system is zero.
- There are no chemical, magnetic and nuclear reactions.
- Heat transfer takes place only in the condenser and the evaporator units.
- The compression process on the compressor is adiabatic.
- There are no heat and work transfer in the expansion valve.
- The directions of the work done by the system and heat transfer and mass entry to the system are positive.
- The air is an ideal gas and has a constant specific heat.
- The atmospheric pressure is 101.3 kPa.

In the exergy balance, the difference between inlet and outlet exergy transfer rate by heat, work and mass equals to rate of the exergy destruction (Equation 1) [8, 10-12].

$$\dot{X}_{heat,in}+\dot{X}_{mass,in}-\dot{X}_{work,out}=\dot{X}_{dest} \tag{1}$$

The exergy transfer rate by heat, work and mass can be defined as follow (Equation 2-4):

$$\dot{X}_{heat,in}=\left(1-{}^{T_0}/_{T}\right)\dot{Q}_{in} \tag{2}$$

$$\dot{X}_{work,out}=\dot{W}_{out} \tag{3}$$

$$\dot{X}_{mass}=\dot{m}(\varepsilon_{in}-\varepsilon_{out}) \tag{4}$$

The general exergy destruction rate can be obtained by substituting equations (2), (3) and (4) into the equation (1) (Equation 5).

$$\dot{X}_{dest}=\left(1-{}^{T_0}/_{T}\right)\dot{Q}-\dot{W}+\dot{m}(\varepsilon_{in}-\varepsilon_{out}) \tag{5}$$

The flow exergy is evaluated as (Equation 2) [8, 10-12]:

$$\varepsilon=(h-T_0s)-(h_0-T_0s_0) \tag{6}$$

Alternatively, the total flow exergy of moist air can be also calculated by the following (Equation 7) [4, 8].

$$\begin{aligned}\varepsilon_{air}=&\left(c_{p,a}+\omega c_{p,v}\right)T_0\left(\frac{T}{T_0}-1-\ln\frac{T}{T_0}\right)+(1+\tilde{\omega})R_aT_0\ln\frac{P}{P_0}\\&+R_aT_0\left[(1+\tilde{\omega})\ln\frac{1+\tilde{\omega}_0}{1+\tilde{\omega}}+\tilde{\omega}\ln\frac{\tilde{\omega}}{\tilde{\omega}_0}\right]\end{aligned} \tag{7}$$

where $\tilde{\omega}=1.608\omega$

In the MAC system, an electrical work is transferred in the compressor unit in order to increase the pressure of the refrigerant. The amount of the rate of work is (Equation 8)

$$\dot{W}_{comp}=\dot{m}_{ref}(h_2-h_1) \tag{8}$$

The amount of heat transfer in the condenser unit is described from the first law of thermodynamics for both refrigerant and air sides as follows (Equation 9):

$$\dot{Q}_{cond}=\dot{m}_{ref}(h_2-h_3)=\left(\dot{m}c_p\right)_{air}(T_5-T_6) \tag{9}$$

For refrigerant and air sides, the amount of heat transfer in the evaporator unit is defined as (Equation 10):

$$\dot{Q}_{evap}=\dot{m}_{ref}(h_1-h_4)=\left(\dot{m}c_p\right)_{air}(T_8-T_7) \tag{10}$$

In the entropy balance, the difference between outlet and inlet entropy transfer rate by heat and mass equals to rate of the entropy generation [10, 11]. The entropy generation rate during any process is written as (Equation 11) [10-13]

$$\dot{S}_{gen}=\sum\dot{m}_{out}s_{out}-\sum\dot{m}_{in}s_{in}-\sum\frac{\dot{Q}_{in}}{T} \tag{11}$$

Table 2. The equations of the exergy destruction, the entropy generation, and the exergy efficiency for each components of MAC

Component	The Irreversibility Rate from The Exergy Destruction $\dot{I}=\dot{X}_{dest}$ (kW)	The Irreversibility Rate from the Entropy Generation $\dot{I}=T_o\dot{S}_{gen}$ (kW)	The Rational Exergy Efficiency ψ (%)
Compressor	$\dot{m}_{ref}(\varepsilon_1-\varepsilon_2)+\dot{W}_{comp}$	$T_o\dot{m}_{ref}(s_2-s_1)$	$\frac{\varepsilon_2-\varepsilon_1}{h_2-h_1}$
Condenser	$\dot{m}_{ref}(\varepsilon_2-\varepsilon_3)+\dot{m}_{air}(\varepsilon_6-\varepsilon_5)$ or $\dot{m}_{ref}(\varepsilon_2-\varepsilon_3)-\left(1-{T_o}/{T_{cond}}\right)\dot{Q}_{cond}$	$T_o\left[\dot{m}_{ref}(s_3-s_2)+(\dot{m}c_p)_{air}\ln(T_5/T_6)\right]$ or $T_o\dot{m}_{ref}\left[(s_3-s_2)+{(h_2-h_3)}/{T_{cond}}\right]$	$\left(1-\frac{T_o}{T_{cond}}\right)\cdot\frac{h_2-h_3}{\varepsilon_2-\varepsilon_3}$
Expansion valve	$\dot{m}_{ref}(\varepsilon_3-\varepsilon_4)$	$T_o\dot{m}_{ref}(s_4-s_3)$	$\frac{\varepsilon_4}{\varepsilon_3}$
Evaporator	$\dot{m}_{ref}(\varepsilon_4-\varepsilon_1)+\dot{m}_{air}(\varepsilon_8-\varepsilon_7)$ or $\dot{m}_{ref}(\varepsilon_4-\varepsilon_1)+\left(1-{T_o}/{T_{evap}}\right)\dot{Q}_{evap}$	$T_o\left[\dot{m}_{ref}(s_1-s_4)+(\dot{m}c_p)_{air}\ln(T_7/T_8)\right]$ or $T_o\dot{m}_{ref}\left[(s_1-s_4)-{(h_1-h_4)}/{T_{evap}}\right]$	$\left(1-\frac{T_o}{T_{evap}}\right)\cdot\frac{h_4-h_1}{\varepsilon_4-\varepsilon_1}$
MAC system	$\dot{m}_{air}[(\varepsilon_6-\varepsilon_5)+(\varepsilon_8-\varepsilon_7)]+\dot{W}_{comp}$ or $\left(1-\frac{T_o}{T_{evap}}\right)\dot{Q}_{evap}-\left(1-\frac{T_o}{T_{cond}}\right)\dot{Q}_{cond}+\dot{W}_{comp}$	$T_o(\dot{m}c_p)_{air}[\ln(T_5/T_6)+\ln(T_7/T_8)]$ or $T_o\dot{m}_{ref}\left[{(h_2-h_3)}/{T_{cond}}-{(h_1-h_4)}/{T_{evap}}\right]$	$\left(1-\frac{T_o}{T_{evap}}\right)\cdot\frac{h_4-h_1}{h_2-h_1}$

The specific entropy difference of an ideal gas under the constant specific heat is achieved as follow (Equation 12) [9]

$$s_{out}-s_{in}=c_{p,air}\ln\frac{T_{out}}{T_{in}}-R_{air}\ln\frac{P_{out}}{P_{in}} \qquad (12)$$

The irreversibility rate and the exergy destruction rate are the similar and it is also obtained by multiplying the entropy generation rate with the temperature of the dead state (Equation 13).

$$\dot{I}=\dot{X}_{dest}=T_0\dot{S}_{gen} \qquad (13)$$

The rational exergy is the ratio of the desired exergy output to the consumed exergy input (Equation 14) [10, 15].

$$\psi=\frac{\dot{X}_{des,out}}{\dot{X}_{cons,in}} \qquad (14)$$

The formulations of the irreversibility analysis and exergy efficiency are summarized in Table 2. In the calculations, the ideal gas constant of dry air and the specific heat of moist air are taken as R_a=0.287 kJ/kgK and $c_{p,air}$=1.016 kJ/kgK, respectively [16]. The dead state properties for the refrigerant and the atmospheric air are shown in Table 3.

Table 3. The dead state properties

Properties	Air	Refrigerant
Pressure, P_o	101.3 kPa	101.3 kPa
Temperature, T_o	18 °C	18 °C
Specific enthalpy, h_o	34.74 kJ/kg	320.0 kJ/kg
Specific entropy, s_o	0.132 kJ/kgK	1.08 kJ/kgK

4. RESULTS AND DISCUSSIONS

The condensation pressure was gradually increased from 650 kPa to 770 kPa while the evaporation pressure increased from 210 kPa to 330 kPa. For the MAC system represented in Figure 2, the experimental and calculated results for the condensation pressure of 700 kPa are summarized in Table 4.

Table 4. Experimental results of the MAC system

State	P (kPa)	T (□)	h (kJ/kg)	s (kJ/kgK)	ε (kJ/kg)	$\dot{X}$ (kW)
1	315.0	2.5	302.0	0.933	24.74	1.626
2	700.0	58.4	347.0	1.020	44.47	2.923
3	700.0	25.5	137.1	0.327	36.34	2.388
4	315.0	1.95	137.1	0.332	34.90	2.293
5	101.3	29.2	51.37	0.184	1.490	1.179
6	101.3	21.3	43.30	0.158	0.990	0.784
7	101.3	12.1	23.91	0.089	1.689	1.044
8	101.3	20.8	24.06	0.086	2.713	1.676
0 (ref)	101.3	18.0	320.0	1.080	–	–
0 (air)	101.3	18.0	34.7	0.132	–	–

The effect of the condensation pressure on the specific flow exergy and the specific entropy of the cycle locations are indicated in Figures 3 and 4. As can be seen in Figure 3, the maximum flow exergy was obtained at the point 2 which is the inlet of the condenser unit (or exit of the compressor) and the minimum flow exergy was obtained at the point 1 which is the exit of the evaporator unit (or inlet of the compressor). It means that, the exergy flow is maximum where the temperature of the refrigerant is highest and the exergy flow is minimum where the temperature of the refrigerant is lowest. As the condensation pressure increased from 650 kPa to 770 kPa, the specific flow exergies of the points 1, 2, 3 and 4 increased by 24.3%, 9.22%, 0.26% and 1.63%, respectively.

Figure 4 shows, the minimum specific entropy was obtained at the point 3 and 4 which are the inlet and exit of the expansion valve, respectively (or exit of the condenser and inlet of the evaporator) and the maximum specific entropy was obtained at the point 2 which is the inlet of the condenser unit (or exit of the compressor). When the condensation pressure increased, the specific entropy of the point 1 decreased by 0.6%, on the other hand the specific entropy of the points 3 and 4 increased by 6.9% and 6.2%, respectively. The specific entropy of the point 2 remained almost constant.

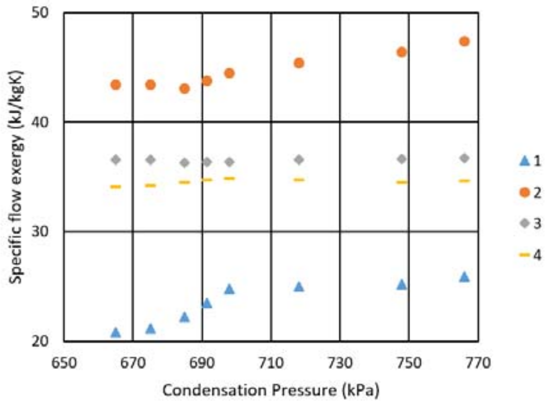

Figure 3. Variation of the specific flow exergy with different condensation pressures

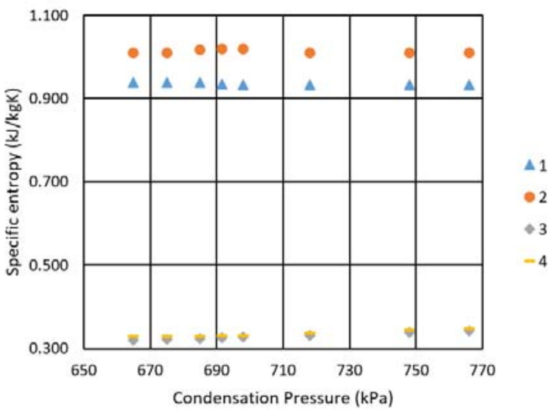

Figure 4. Variation of the specific entropy with different condensation pressures

The rate of the entropy generation within the compressor, the condenser, the expansion valve, the evaporator and the MAC system at the condensation pressure of 700 kPa are 0.00570, 0.00048, 0.00033, 0.00012 and 0.00663 kW/K, respectively. And the change of the entropy generation rate is illustrated in Figure 5. The entropy generation rate within the compressor unit and whole system are much higher than the other components. The rate of the entropy generation is the lowest within the evaporator. The rate of the entropy generation within the compressor, the condenser and the MAC system increased by 11.5%, 45.6% and 8.28%, respectively, when the condensation pressure increased. However, the rate of the entropy generation within the expansion valve and the evaporator decreased by 16.0% and 38.6%.

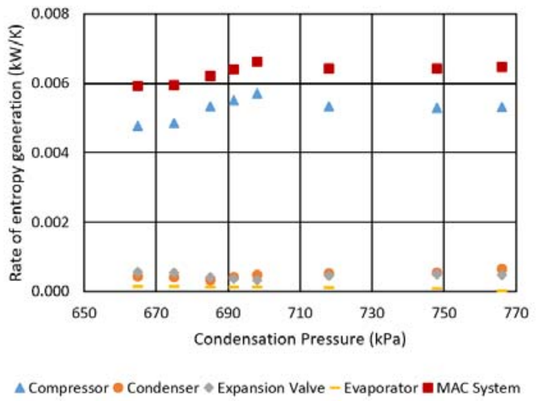

Figure 5. Variation of the entropy generation rate with different condensation pressures

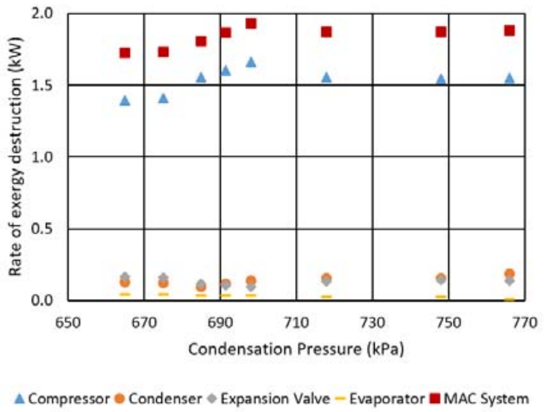

Figure 6. Variation of the irreversibility rate or the exergy destruction rate with different condensation pressures

Figure 6 indicates the effect of the condensation pressure on the irreversibility rate, in other words, the effect on the rate of the exergy destruction. The irreversibility rates of the compressor, the condenser, the expansion valve, the evaporator and the MAC system at the condensation pressure of 700 kPa are 1.6609, 0.1387, 0.0947, 0.0351 and 1.9294 kW, respectively. As the condensation pressure increased from 650 to 770 kPa, the irreversibility rate of the compressor, the condenser and the MAC system increased by 11.5%, 45.6%

and 8.28%, respectively, when the condensation pressure increased. Besides, the irreversibility rate of the expansion valve and the evaporator decreased by 16.0% and 38.6%. Figures 5 and 6 emphasize that in the components of the refrigeration cycle, the trend of the entropy generation rate and the trend of the exergy destruction rate are the same.

The variation of the rational exergy efficiency of the all components and whole system with the condensation pressure is shown in Figure 7. The rational exergy efficiency of the compressor, the condenser, the expansion valve, the evaporator and the MAC system at the condensation pressure of 700 kPa are 43.84%, 74.05%, 96.04%, 94.74% and 21.38%, respectively.

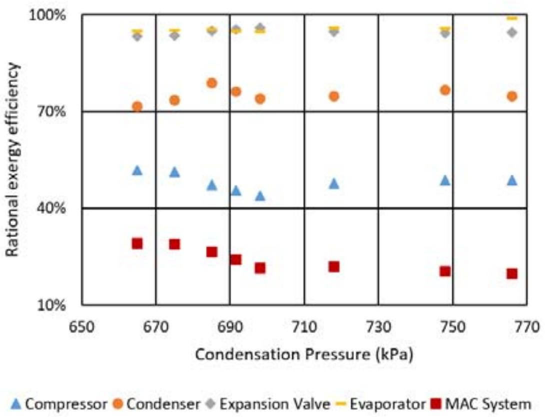

Figure 7. Variation of the rational exergy efficiency with different condensation pressures

Table 5. Experimental results of the MAC system

	Rate of energy transfer (kW)	Rate of entropy generation (kW/K)	Rate of exergy destruction (kW)	Exergy efficiency (%)
Compressor	2.96	0.00570	1.661	43.8%
Condenser	13.79	0.00048	0.139	74.0%
Exp. valve	0.00	0.00033	0.095	96.0%
Evaporator	10.84	0.00012	0.035	94.7%
MAC system	-	0.00663	1.929	21.4%

The calculation results of the rate of energy transfer, entropy generation and exergy destruction or irreversibility and exergy efficiency for the condensation pressure of 700 kPa are summarized in Table 5.

5. CONCLUSIONS

The condensation pressure was increased about 18.5% under the same indoor and outdoor conditions. The effect of the condenser pressure on the rate of irreversibility and on the exergy efficiency is investigated.

The following main concluding remarks may be listed from the main results of this study:

a) The irreversibility analysis was explained in detail and the formulas to be used in the irreversibility analysis for the vapor compression refrigeration cycle were summarized in tabular form (in Table 2).
b) The maximum flow exergy is obtained at the compressor output which is about 44.65 kJ/kg.
c) The irreversibility rate of the whole system was approximately 1.8381 kW. The lowest irreversibility rate was about 0.0321 kW which was obtained in the evaporator.
d) The rate of the entropy generation within the compressor and the MAC system increased by 11.5% and 8.28%, respectively, when the condensation pressure increased.
e) The rational exergy efficiency of the evaporator and the expansion valve were the higher compared to other components and the efficiencies were approximately %96 and %95, respectively. The lowest rational exergy efficiency was about 48% which was obtained in the compressor.
f) The rational exergy efficiency of the whole system was approximately 24%.
g) For a further study, performing exergoeconomic (thermoeconomic) and enhanced (advanced) exergy analyses is recommended.

NOMENCLATURE

c	: Specific heat (kJ/kgK)
$\dot{E}$	: Energy rate (kW)
h	: Specific enthalpy (kJ/kg)
$\dot{I}$	: Irreversibility rate (kW)
$\dot{m}$	: Mass flow rate (kg/s)
P	: Absolute pressure (kPa)
$\dot{Q}$	: Heat transfer rate (kW)
R	: Ideal gas constant (kJ/kgK)
s	: Specific entropy (kJ/kgK)
$\dot{S}$	: Entropy transfer rate (kW/K)
T	: Temperature (°C or K)
$\dot{W}$	: Work transfer rate or power (kW)
$\dot{X}$	: Exergy rate (kW)

Greek symbols

ω	: Specific humidity (kg water vapor/kg dry air)
ϕ	: Relative humidity (%)
ε	: Flow exergy (kJ/kg)
ψ	: Rational exergy efficiency (%)

Subscripts

0	: Dead state
1, 2, ...	: Cycle locations
a	: Dry air
air	: Moist (atmospheric) air
comp	: Compressor
cond	: Condenser or condensation
cons	: Consumed
des	: Desired
dest	: Destroyed
evap	: Evaporator or evaporation
expan	: Expansion valve
gen	: Generation
in	: Inlet
out	: Outlet
p	: Constant pressure
v	: Vapor

Abbreviations

COP	: Coefficient of performance
MAC	: Minibus air conditioning

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