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THERMODYNAMIC ANALYSIS OF BI-EVAPORATOR EJECTOR REFRIGERATION CYCLE USING R744 AS NATURAL REFRIGERANT

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ABSTRACT

Researchers have been focused on refrigerant R744 because it is a natural refrigerant and its Global Warming Potential value is 1. However, coefficient of performance (COP) of refrigeration systems that use R744 as refrigerant has lower values than refrigeration systems that use CFC and HCFC refrigerants as refrigerant. Previous studies showed that COP of the refrigeration system with R744 can be increased by using two-phase ejector. In this study, additional evaporators are used instead of the separator. Thus, the refrigeration system can be constructed more accurately. Theoretical analysis of the bi-evaporator refrigeration system with two-phase ejector is carried out by using energy and momentum equations. COP and entrainment ratio are obtained for various operating conditions. Results show that COP improvement can be achieved about 21% for proper design in Mediterranean climate zone.

INTRODUCTION

The natural refrigerant R744 has no ozone depletion potential (ODP) and low global warming potential (GWP). The environment-friendly refrigerant also has excellent advantages such as absence of toxicity and inflammability, presence of high volumetric capacity, low price and good heat transfer properties [1]. Despite these advantages, critical temperature of carbon dioxide is generally lower than heat rejection temperature in air conditioning applications, so transcritical vapor compression cycle is applicable for R744 in air conditioning systems. Transcritical R744 system has very large pressure difference between heat rejection pressure and evaporating pressure when compared to the other conventional refrigeration systems working with CFC and HCFC refrigerants. Large pressure differences cause higher throttling losses, so COP of the transcritical R744 refrigeration system is much lower than the conventional refrigeration systems.

In recent years, increasing environmental concerns about global warming has led to quicken the studies of R744 refrigeration systems. In order to overcome low COP, throttling loss could be reduced. So, researchers have studied on replacing two-phase ejector with the expansion valve.

The use of two-phase ejector in vapor compression refrigeration system was first introduced by Kornhauser [2] through a numerical analysis using R12 as a refrigerant and reported 21% improvement in COP. Thereafter, various ejector based refrigeration systems have been studied with different working substances.

Transcritical R744 ejector refrigeration cycles take place in literature more than any other refrigerants. Since transcritical cycle has larger throttling losses, R744 transcritical system offer more potential for increase on COP. Li and Groll [3], Deng et al. [4], Nakagawa et al. [5] and Ahammed et al. [6] have investigated the replacement the two phase ejector instead of expansion valve in transcritical R744 refrigeration cycle. Their studies show that the COP is increased 22% over conventional system. In this study, bi-evaporator ejector system is analyzed, additional evaporator is used instead of the separator. Oshitani et al. [7] firstly introduced the two evaporator ejector systems, Lawrence and Elbel [8] investigated the second law analysis and performance characteristics of this ejector cycle, Baumaraf et al. [9] studied and compared the performances of cycle with R134a and R1234yf as the refrigerants. Ünal and Yılmaz [10] performed the thermodynamic analysis of the system enhanced with two phase ejector and two evaporators with R134a refrigerant.7

In this study, thermodynamic analysis of the bi-evaporator refrigeration system with two-phase ejector is carried out for different operating conditions. Change of COP and entrainment ratio of conventional and ejector refrigeration cycle are compared and showed graphically.

BI-EVAPORATOR EJECTOR REFRIGERATION SYSTEM

Standard transcritical ejector refrigeration system components are compressor, gas cooler, ejector, evaporator and liquid separator. The bi-evaporator ejector system has an additional evaporator which takes place instead of the separator. The bi-evaporator ejector system demonstrated in Fig. 1 and pressure-enthalpy (P-h) diagram is shown in Fig. 2. The refrigerant which comes from the primary evaporator enters the compressor. Then the refrigerant is compressed to a desired level for the gas cooler pressure in transcritical stage. The refrigerant is cooled in gas cooler, it leaves the gas cooler still in transcritical zone. The refrigerant is divided into to be sent to the ejector and expansion valve at the gas cooler exit. The refrigerant, which leaves the expansion valve, enters the secondary evaporator, and then it enters the ejector as the secondary fluid. Primary and secondary fluids are mixed in the mixing section. After the mixing process, the refrigerant enters diffuser section of the ejector. And then it goes to the primary evaporator. The refrigerant which comes from the primary evaporator enters the compressor, thus the cooling cycle is completed.

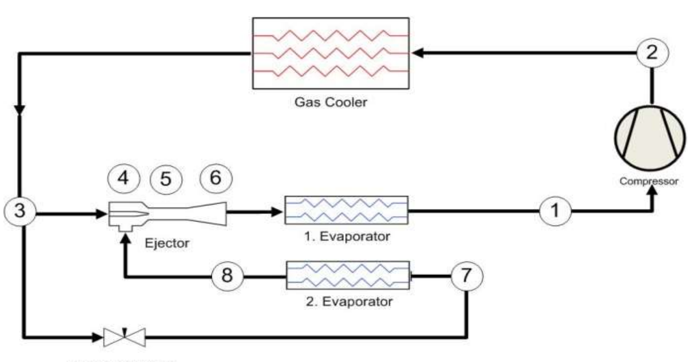

Figure 1 Bi-evaporator ejector refrigeration system

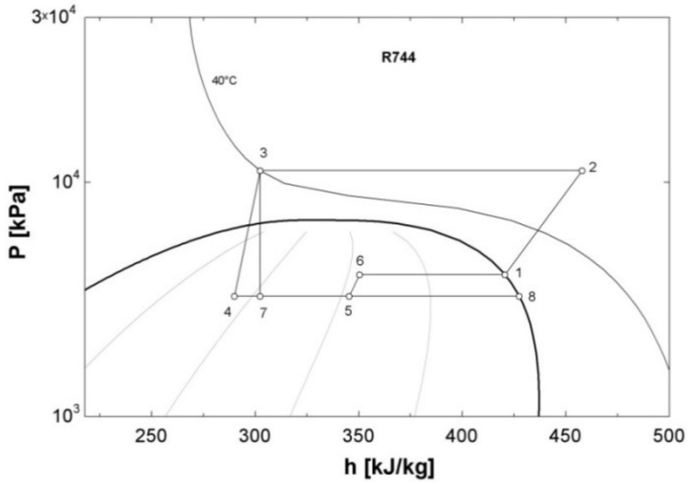

Figure 2 P-h diagram of the ejector refrigeration system

THERMODYNAMIC ANALYSIS

Thermodynamic analysis was carried out with continuity, energy and momentum equations and approach of ejector theory which are followed from Ünal's and Yılmaz [10] study. It is assumed that the mixing process occurs at the constant cross-sectional area and constant pressure. Also refrigerant velocities at the points (8) and (6) shown in Fig. 2 are taken into consideration in energy and momentum equations.

Thermodynamic analysis of the ejector refrigeration system is carried out by considering the following assumptions.

a) Gas cooler exit temperature, gas cooler pressure and evaporating temperatures are known.
b) Pressure losses of the system are neglected.
c) The throttling process in expansion valve isenthalpic.
d) Isentropic efficiencies of the nozzle and diffusor are known.
e) Efficiency of the mixing section of ejector is known.
f) The process in the mixing section takes place at constant pressure and constant cross-sectional area.

All the thermodynamic properties of point (1) can be determined if the primary evaporator temperature is known. In order to calculate the thermodynamic properties at the compressor exit, compressor isentropic efficiency expression can be used as given below:

$$\eta_c = \frac{h_{2s} - h_1}{h_2 - h_1} \quad (1)$$

Isentropic efficiency for compressor η_c has been calculated from the following correlation given by Robinson and Groll [11].

$$\eta_c=0{,}815+0{,}022\cdot P^{*}-0{,}0041\cdot\left(P^{*}\right)^{2}+0{,}0001\cdot\left(P^{*}\right)^{3} \qquad (2)$$

P^* is defined as follows:

$$P^{*}=\frac{P_1}{P_2} \qquad (3)$$

h_{2s} is determined by using Eq. (4) as a function of the P_{2s} and s_{2s}. As known, s_{2s} is equal to s_1.

$$h_{2s}=F(s_{2s}, P_{2s}) \qquad (4)$$

The enthalpy of the refrigerant at the compressor exit can be calculated from Eq. (1) by using the compressor isentropic efficiency given in Eq. (2). Since the gas cooler exit temperature is known, thermodynamic properties at the gas cooler exit are calculated.

An ejector comprises three main sections that are nozzle, mixing and diffusor. Schematic view of the ejector is given in Fig. 3. Points (3), (4), (5) and (6) indicate the nozzle inlet, diffuser inlet and diffuser exit, respectively.

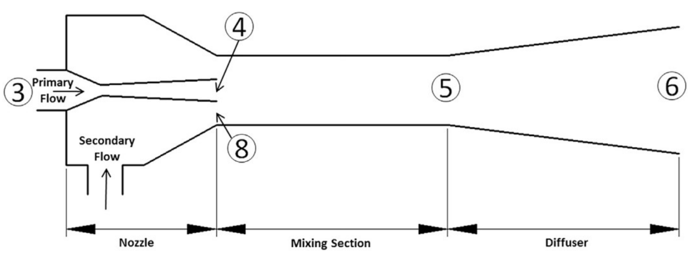

Figure 3 Schematic view of the ejector

Thermodynamic properties at the nozzle exit can be calculated by using the energy equation between points (3) and (4) given in Eq. (5) and nozzle isentropic efficiency given in Eq. (6). Due to conservation of mass principle, it should be considered that $\dot{m}_3=\dot{m}_4$ and velocity of the refrigerant at the nozzle inlet is neglected in Eq. (5).

$$h_3=h_4+\frac{V_4^2}{2} \qquad (5)$$

$$\eta_n=\frac{h_3\text{-}h_4}{h_3\text{-}h_{4s}} \qquad (6)$$

Thermodynamic properties at the diffuser inlet can be calculated by using energy and momentum equations in Eq. (8) and Eq. (9) as follows:

$$\dot{m}_4+\dot{m}_8=\dot{m}_5 \qquad (7)$$

$$\left(h_4+\frac{V_4^2}{2}\right)+\omega\left(h_8+\frac{V_8^2}{2}\right)=(1+\omega)\left(h_5+\frac{V_5^2}{2}\right) \qquad (8)$$

$$P_4A_4+\eta_{mix}\rho_4A_4V_4^2+P_8(A_8)+\eta_{mix}\rho_8(A_8)V_8^2=P_5A_5+\rho_5A_5V \qquad (9)$$

Mixing efficiency denotes the frictional losses in the mixing section and it is defined as follows: ω is the entrainment ratio, which expresses the ratio of mass flow rates of primary and secondary fluids that enter the ejector, has been defined by Eq. (10):

$$\omega=\dot{m}_8/\dot{m}_3 \qquad (10)$$

For the determination of thermodynamic properties of the refrigerant at the diffuser exit, energy equation and diffuser isentropic efficiency can be used as given Eq. (11) and Eq. (12), respectively. The minimum refrigerant velocity for the oil return was recommended as 5-7 ms^{-1} in the compressor suction line [12]. There is an evaporator between the compressor and diffuser outlet of the investigated two phase ejector cooling system in this work. So, velocity of the refrigerant at the diffuser outlet is considered as $V_6=15\ ms^{-1}$ for the sake of safe oil return.

$$h_5+\frac{V_5^2}{2}=h_6+\frac{V_6^2}{2} \qquad (11)$$

$$\eta_d=\frac{h_{6,is}\text{-}h_5}{h_6\text{-}h_5} \qquad (12)$$

The refrigerant enters the expansion valve at the point (3) and gets out from point (7). The expansion process in the expansion valve is constant enthalpy process.

Primary mass flow rate ($\dot{m}_3$) can be calculated from the cooling capacity of the system which is given in Eq. (13):

$$\dot{Q}=\dot{m}_3[\omega(h_8\text{-}h_7)+(1+\omega)(h_1\text{-}h_6)] \qquad (13)$$

The coefficient of performance of the system can be calculated by the equation below:

$$COP=\frac{\dot{Q}_{e1}+\dot{Q}_{e2}}{W}=\frac{\omega(h_8-h_7)+(1+\omega)(h_1-h_6)}{(1+\omega)(h_2-h_1)} \quad (14)$$

COP of the two-phase ejector refrigeration system given in Eq. (14) is compared to the COP of the conventional refrigeration system (COP_{std}), and the COP increase rate (COP^*) is determined as follows:

$$COP^*=\frac{COP-COP_{std}}{COP_{std}} \quad (15)$$

The coefficient of performance of the conventional refrigeration system can be calculated by the following equation:

$$COP_{std}=\frac{h_8-h_7}{h_2-h_8} \quad (16)$$

RESULTS AND DISCUSSION

The variation of coefficient of performance (COP) and entrainment ratio of the system with two-phase ejector cycle are investigated and presented at different operating conditions. For the design condition of the system, the ambient temperature can be taken as 35 ℃ for the Mediterranean climate zone [13]. Thus, gas cooler exit temperature should be set 42 ℃ according to the ambient temperature which is mentioned above. For practical applications, evaporator temperatures were taken as 12 ℃ and 5 ℃ as the primary and secondary evaporators, respectively [10]. In the calculations, nozzle, mixing and diffuser section efficiencies were taken as 0.85, 0.8 and 0.85 respectively [14].

The variation of the conventional refrigeration system and ejector refrigeration system's COP with gas cooler exit temperature is shown in Fig. 4. It is seen that they decrease with gas cooler exit temperature. As shown in Fig. 5, COP increase rate decreases from 23% to 18% and entrainment ratio increases, consequently, entrainment ratio and COP increase rate are inversely proportional for these conditions.

The variation of the COP values of two systems with gas cooler pressure is shown in Fig. 6. In consideration of CO_2 thermal property, COP values increase. When reached to the optimum level, it starts decreasing. However, ejector cycle always has better performance than conventional cycle. Fig. 7 also shows the COP increasing ratio alters between 19% and 23% and in this figure it can be seen that maximum COP* is obtained at gas cooler pressure as 9800 kPa.

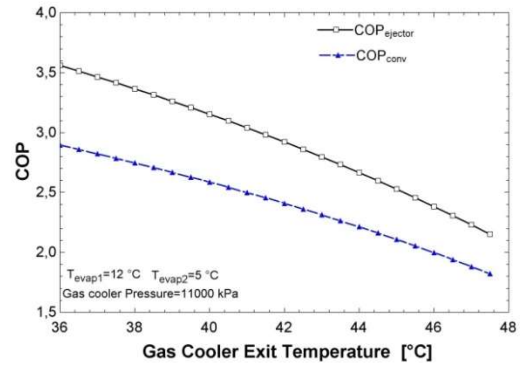

Figure 4 Variation of COP values with gas cooler exit temperature.

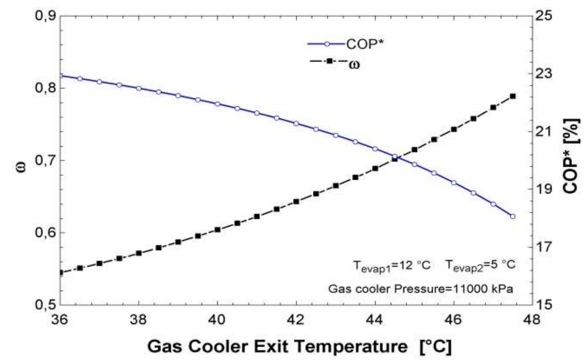

Figure 5 Variation of entrainment ratio and COP increase rate with gas cooler exit temperature.

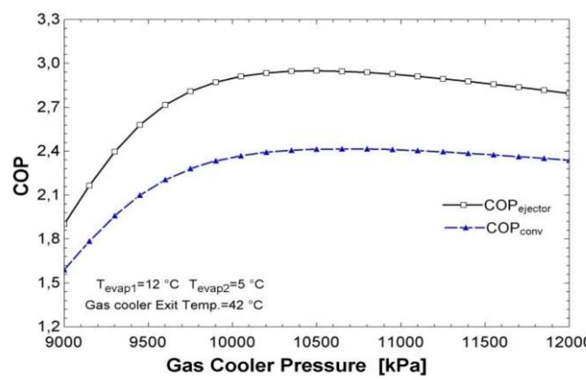

Figure 6 Variation of COP values with gas cooler pressure.

The effect of the second evaporator temperature on COP and entrainment ratio are given in Fig. 8 and Fig. 9. COP of the conventional system increases with second evaporator temperature, but COP of the ejector system decreases slightly. Since cooling load is accepted as constant, COPs depend on the compressor work only. When second evaporator temperature gets high, its saturation pressure increases. In conventional systems, as the second evaporator pressure increases, compression ratio decreases, therefore COP increases. Second evaporator temperature does not affect compression ratio in ejector refrigeration system. Because the total refrigerant mass flow rate rises, COP decreases slightly. Fig. 9 illustrates that pressure lift inversely proportional to the entrainment ratio in the system.

The variation of the conventional refrigeration system and ejector refrigeration system's COP with first evaporator temperature is shown in Fig. 10. In conventional refrigeration system, there is no first evaporator, so does not effect on conventional COP. In ejector refrigeration system, COP depends on the first evaporator pressure and temperature.

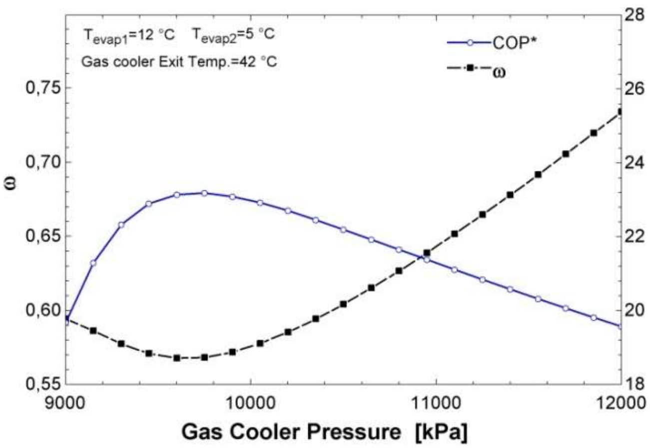

Figure 7 Variation of entrainment ratio and COP increase rate with gas cooler pressure.

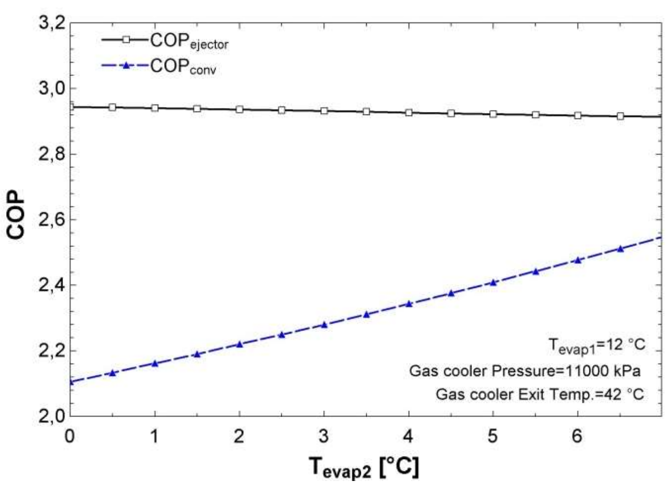

Figure 8 The effect of second evaporator temperature on COP.

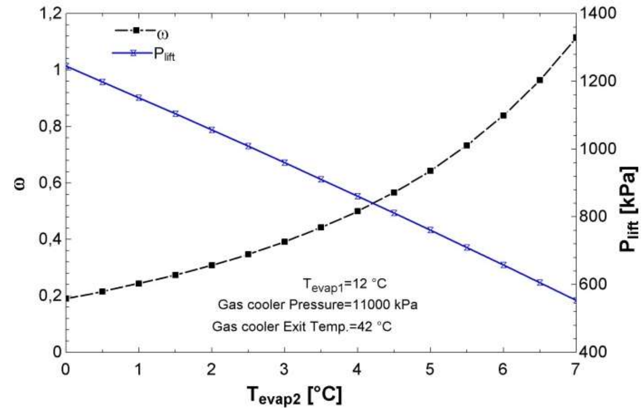

Figure 9 The effect of second evaporator temperature on entrainment ratio.

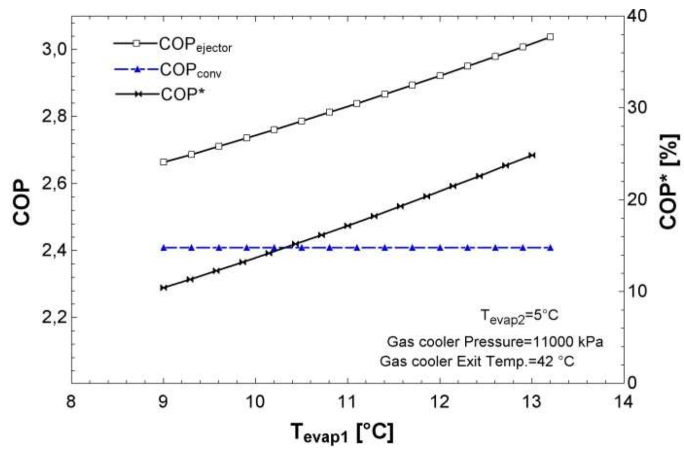

Figure 10 The effect of first evaporator temperature on COP values.

CONCLUSION

This paper provides the thermodynamic analysis of the bi-evaporator ejector refrigeration cycle using R744 as natural refrigerant. COP of this system is compared with the conventional refrigeration system. Variations of COP and entrainment ratio are analyzed depending on some design parameters. Results showed that COP can be increased 21% with using the two phase ejector in refrigeration system according to the conventional system. Also it is found that COP is inversely proportional to the entrainment ratio.

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NOMENCLATURE

COP	coefficient of performance [-]
COP_{std}	coefficient of performance of the conventional system [-]
COP^*	increase in COP [%]
h	enthalpy [J kg^{-1}]
$\dot{m}$	mass flow rate [kg s^{-1}]
P	pressure [N m^2]
$\dot{Q}$	cooling capacity [W]
s	entropy [J kg^{-1}K]
T	temperature [℃]
V	velocity [ms^{-1}]
η_c	Isentropic efficiency of compressor [-]
η_n	Isentropic efficiency of nozzle [-]
η_d	Isentropic efficiency of diffuser [-]
η_m	Mixing section efficiency [-]
ω	entrainment ratio [-]

REFERENCES

[1] G. Lorentzen, Revival of carbon dioxide as a refrigerant, *Int. J. Refrig.* **I7 (5)** (1994), 292-301.

[2] Kornhauser AA, The use of an ejector as a refrigerant expander, International Refrigeration and Air-conditioning Conference, 1990; 82:10-19.

[3] Li, D., Groll, E.A., Transcritical CO_2 refrigeration cycle with ejector expansion device. *Int. J. Refrigeration* **28** (2005), 766-773

[4] Deng, J., Jiang, P., Lu, T., Lu, W., Particular characteristics of transcritical CO2 refrigeration cycle with an ejector. *Appl. Therm. Eng*. **27** (2007), 381-388.

[5] Nakagawa, M., Marasigan, A.R., Matsukawa, T., Kurashina, A.,. Experimental investigation on the effect of mixing length on the performance of two-phase ejector for CO2 refrigeration cycle with and without heat exchanger. *Int. J. Refrigeration* **34** (2011), 1604-1613.

[6] Ahammed E., Bhattacharyya S., Ramgopal M.. Thermodynamic design and simulation of a CO2 based transcritical vapour compression refrigeration system with an ejector. *Int. J. Refrigeration* **45** (2014), 177-188.

[7] Oshitani, et al., 2005. Vapor compression cycle having ejector. US Patent 0268644 A1.

[8] Lawrence, N., Elbel, S.,. Experimental investigation of a twophase ejector cycle suitable for use with low-pressure refrigerants R134a and R1234yf. *Int. J. Refrigeration* **38** (2014), 310-322.

[9] Boumaraf L., Haberschill P., Lallemand A. Investigation of a novel ejector expansion refrigeration system using the working fluid R134a and its potential substitute R1234yf. *Int. J. Refrigeration* **45** (2014), 148-159.

[10] Ünal Ş., Yılmaz T. Thermodynamic analysis of the two-phase ejector air-conditioning system for buses. *Appl. Therm. Eng.* **79** (2015), 108-116.

[11] Robinson, D.M., Groll, E.A.,. Efficiencies of transcritical CO2 cycles with and without an expansion turbine. *Int. J. Refrigeration* **21 (7)** (1998), 577-589

[12] A.R. Trott, T. Welch, Refrigeration and Air-conditioning, third ed., Butter-worth-Heinemann, Oxford, UK, 2000.

[13] Climatic Design Information, ASHRAE Handbook, fundamentals, Atlanta, USA, 2001.

[14] F. Liu, E.A. Groll, Study of ejector efficiencies in refrigeration cycles, *Appl.Therm. Eng.* **52** (2013), 360-370.