International Journal of Mechanical, Industrial and Aerospace Sciences
ISSN: 2517-9950
Vol:5, No:9, 2011

Thermodynamic Analysis of R507A-R23 Cascade Refrigeration System

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***Abstract*—**The present work deals with thermodynamic analysis of cascade refrigeration system using ozone friendly refrigerants pair R507A and R23. R507A is azeotropic mixture composed of HFC refrigerants R125/R143a (50%/50% wt.). R23 is a single component HFC refrigerant used as replacement to CFC refrigerant R13 in low temperature applications. These refrigerants have zero ozone depletion potential and are non-flammable and as R507A an azeotropic mixture there is no problem of temperature glide. This study thermodynamically analyzed R507A-R23 cascade refrigeration system to optimize the design and operating parameters of the system. The design and operating parameters include: Condensing, evaporating, subcooling and superheating temperatures in the high temperature circuit, temperature difference in the cascade heat exchanger, Condensing, evaporating, subcooling and superheating temperatures in the low temperature circuit.

***Keywords*—**COP, R507A, R23, cascade refrigeration system

I. Introduction

IN low temperature refrigeration application including rapid freezing, storage of frozen food, liquefaction of petroleum vapour, manufacturing of dry ice etc. where the required evaporating temperature between -40 ^{0}C and -130 ^{0}C, the cascade system is often a convenient option, provided two suitable working fluids are chosen. A commonly used refrigerants pair in the past has been R12, R502 in high temperature cycle and R13 in low temperature cycle of cascade refrigeration system. These refrigerants have been phased out since 1996 in the developed countries, and should be totally phased out by 2010 in developing countries as per Montreal Protocol and its amendments from the United Nations Environment Programme (UNEP) [1, 2]. It becomes impendence to look for new refrigerants to substitute for R12, R-502 and R13.

Dinitrogen monoxide (N_2O) was reported to be used as a cascade refrigerant for achieving temperatures around -80 ^{0}C [3]. Carbon dioxide (CO_2) is also used in the low-temperature stage in two-stage cascade refrigeration systems [4], for which the lowest refrigeration temperature is limited above -55 ^{0}C because of its high triple-point temperature. Mixtures especially exhibiting azeotropic phase equilibrium behaviors show good potentials in cascade refrigeration systems. Phase equilibrium measurements of two binary systems of R170+ R23 [5] and R170+R116 [6] show positive azeotropic vapor–liquid phase equilibrium and therefore can be used as alternative refrigerants in the low-temperature stage of the two-stage cascade refrigeration system. Carbon dioxide and nitrous oxide binary mixture [7] is also used in low temperature cycle with R404a in high temperature cycle and experimental results were compared with R23 in low temperature cycle and R404a in high temperature cycle. Baolian Niu and Yufeng Zhang [8] used a new binary mixture of R744 and R290 as an alternative natural refrigerant to R13 and Experimental studies for this mixture on a cascade refrigeration system shows COP and refrigeration capacity of this binary mixture were higher than those of R13. Molenaar [9] compared the performance of three types cascade refrigerated test chambers. He concluded that R22 and R23 were acceptable substitutes for R502 and R13 in cascade refrigeration systems for test chamber application. Keumnam Cho, et al. [10] also investigated the effect of experimental parameters of a cascade system using R23 as alternative refrigerant. R508b is a binary azeotropic mixture comprised of R23 (46%) and R116 (54%) on a mass basis. It has many remarkable properties, such as non-flammable, non-toxic, etc. Di Nicola et al. [11] proposed blends of carbon dioxide and HFCs as working fluids for the low-temperature circuit in cascade refrigerating systems. Results showed that the R744 blends were an attractive option for the low-temperature circuit of cascade systems operating at temperatures approaching 200 K. Wadell [12] studied a two stage R134a/R508b cascade refrigeration system providing refrigerant (R508b) to the evaporator at flow rates of (0.833-1.167)$\times 10^{-3}$ kgs^{-1} and saturation temperatures of 187.15-194.15 K.

In present study thermodynamic analysis of cascade refrigeration system has been carried out using ozone friendly refrigerants pair R507A and R23. A thermodynamic analysis of cascade system has been carried to optimize the design and operating parameters of the system.

II. Cascade Refrigeration System

A schematic layout and T-S plot of two stage cascade system using two refrigerants as shown in Fig.1. The compressed refrigerant vapour from the lower stage is condensed in a heat exchanger, usually called cascade

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condenser, which is also the evaporator of the next higher stage refrigerant.

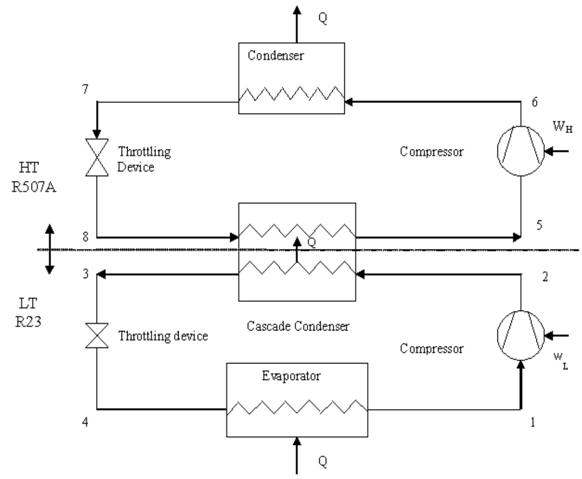

Fig.1 Schematic diagram of cascade system

The only useful refrigerating effect is produced in the evaporator of low temperature cascade system. The high temperature and low temperature cycle uses a refrigerant with high boiling temperature and low boiling temperature respectively.

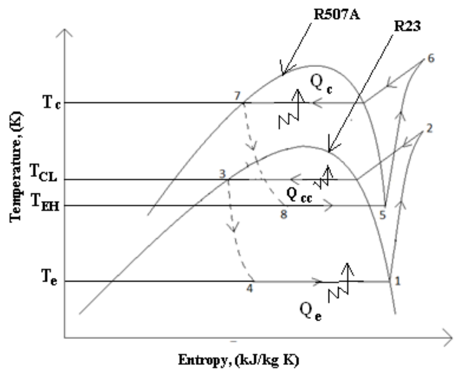

Fig.2 T-S diagram of cascade system

III. Thermodynamic Analysis

The cycle is modeled modularly incorporating each individual process of the cycle. Steady flow energy equation and mass balance equation has been employed. The following assumptions has been considered to simplify the calculation,

1. Heat transfer in cascade heat exchanger with the ambient is negligible.
2. The compressor process in the compressor is adiabatic and irreversible.
3. The expansion process is isenthalpic
4. Pressure drop in the connecting pipes and heat exchangers are negligible.
5. Heat transfer process in heat exchanger is isobaric.

IV. Results And Discussion

Computational model has been developed in engineering equation solver to evaluate the performance of the system. To find the effect of particular parameter on the performance system other parameters of system have been kept constant. The parameters assumed for the computation of results are furnished below.

1. Low temperature circuit evaporating temperature, $T_{E,LT}$ = -82 ^{0}C.
2. High temperature circuit condensing temperature, $T_{C,HT}$ = 35 ^{0}C.
3. Low temperature circuit condensing temperature, $T_{C,LT}$ = -36 ^{0}C.
4. Temperature difference in cascade condenser, $(\Delta T)_{CC}$ = 2.5 ^{0}C.
5. Degree of superheating , $(\Delta T)_{sup}$ = 0 ^{0}C in both HT and LT circuit.
6. Degree of subcooling , $(\Delta T)_{sub}$ = 0 ^{0}C in both HT and LT circuit.
7. Isentropic efficiency, η_{isen}, = 0.8 in both HT and LT compressor.
8. Effectiveness of cascade heat exchanger (ε_{CC}) = 1.
9. Mass flow rate of refrigerant is assumed 0.2 kg/min in LT circuit.

The parameters have been varied for the computations of results are furnished below.

1. The low temperature circuit evaporator temperature $T_{E,LT}$ is varied from -80 ^{0}C to -50 ^{0}C.
2. The high temperature circuit condensing temperature $(T_{C,HT})$ is varied from 30 ^{0}C to 55 ^{0}C.
3. The cascade condenser temperature difference ΔT_{CC} is varied from 2.5 ^{0}C to 12.5 ^{0}C.
4. Low temperature circuit condensing temperature $T_{C,LT}$ is varied from -50 ^{0}C to 0 ^{0}C.
5. Degree of subcooling $(\Delta T)_{sub}$ and Degree of superheating $(\Delta T)_{sup}$ is varied from 0 ^{0}C to 18^{0}C.

Fig.3 depicts the variation of COP, RE and WD with low temperature circuit evaporator temperature $(T_{E,LT})$ which has been varied from the -80 ^{0}C to -50 ^{0}C. The other operating parameters have been kept constant as described above. The overall COP of the system increased from 0.7851 to 1.232 as there is decrease in pressure ratio across low temperature circuit with increase in evaporator temperature. This results in reduced total compressor work by system and increased system refrigerating effect.

The COP of system deteriorated from 0.9274 to 0.5486 when high temperature circuit condenser temperature $(T_{C,HT})$ has been increased from 25 ^{0}C to 50 ^{0}C keeping other parameters constant (Fig.4). Rise in condenser temperature increases compressor work of HT circuit due to increase in pressure ratio across HT circuit and hence total work done by system increases while COP decreases. Theoretically there is no effect of $T_{C,HT}$ on refrigerating effect of system.

International Journal of Mechanical, Industrial and Aerospace Sciences
ISSN: 2517-9950
Vol:5, No:9, 2011

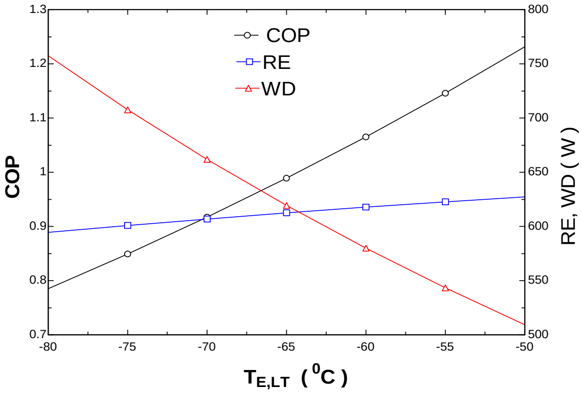

Fig.3 Effect of low temperature stage evaporator temperature on COP, RE and WD

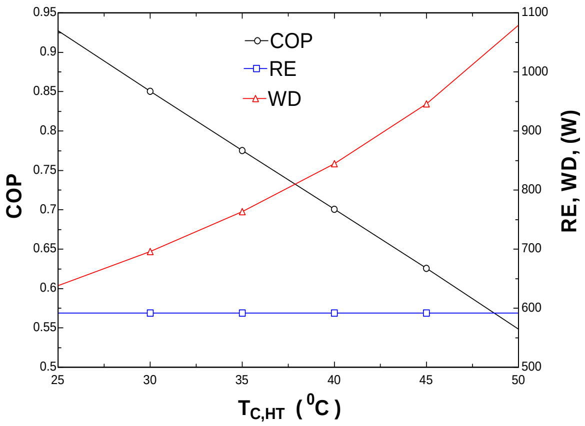

Fig.4 Effect of high temperature (HT) stage condensing temperature on COP, RE and WD

Fig.5 displays the effect of LT circuit condensing temperature $T_{C,LT}$ on COP of HT and LT circuit. The COP of HT circuit increased with $T_{C,LT}$ whereas that the COP of LT circuit decreased with $T_{C,LT}$. Hence the optimal value of $T_{C,LT}$ and corresponding maximum COP of system exist. From fig.5 reveals that maximum COP is 0.7805 at $T_{C,LT}$ = -25 ^{0}C. The RE and WD both decreases with $T_{C,LT}$, so one has to obtain the value corresponding to maximum COP.

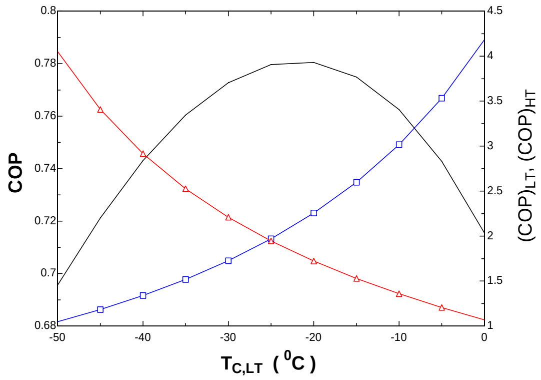

Fig.5 Effect of low temperature (LT) stage condensing temperature on COP

The system COP reduced by about 17.10 % when temperature difference in cascade condenser has been increased from 2.5 ^{0}C to 12.5 ^{0}C as shown in Fig.6. This is intuitive as increase in temperature difference causes heat transfer to occur through finite temperature difference resulting in external irreversibility in the system thereby decreasing the system performance. Theoretically there is no effect of $(\Delta T)_{CC}$ on refrigerating effect of system but work of compressor increases with increase in $(\Delta T)_{CC}$.

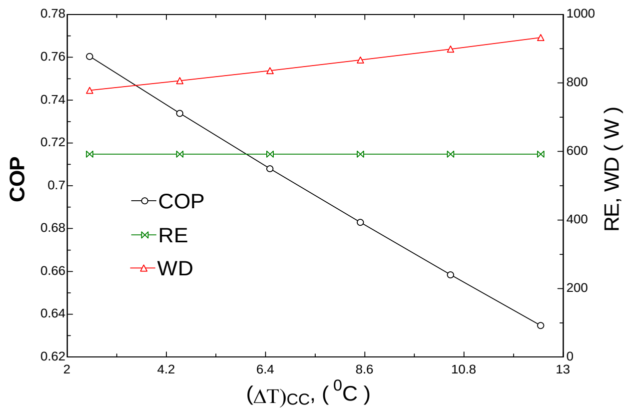

Fig.6 Effect of temperature difference in cascade condenser on COP, RE and WD

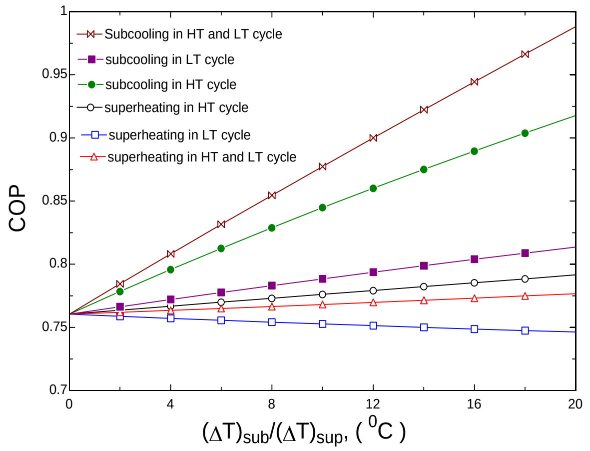

Fig.7 Effect of degree of subcooling and superheating on COP

The effect of having different and the same degree of subcooling and superheat in both cycles of the R507A-R23 cascade system was separately and jointly analyzed keeping the other operating parameters constant (Fig.7). Reference COP is 0.7604 at 0 degree of subcooling and superheating in the both circuit. It was observed that the rise in the COP of the system by higher amount in the case of subcooling in the both circuit than the subcooling separately in each circuit keeping superheating in the both circuit at 0 ^{0}C. The COP of system increased by the 26.3% when subcooling in both circuit varied from 0 to 18 ^{0}C. Fig.6 also shows subcooling in HT circuit results in more COP than subcooling in LT circuit. It is also observed that the COP of the system rises with superheating in

HT circuit while falls with superheating in LT circuit. Similarly there is marginally rising in the COP of the system in the case of jointly superheating the high and low temperature circuit. It is assumed here that superheating of refrigerant in HT circuit improves COP only if superheating takes place in cascade condenser due to absorption of heat from LT circuit condenser similarly superheating of LT circuit refrigerant takes place in evaporator due to heat transfer with surrounding fluid.

Fig.8 depicts the change in the maximum COP of the system at different values of superheating and subcooling as the condensing temperature of LT circuit ($T_{C,LT}$) is varied between -50 ^{0}C and 0 ^{0}C. COP_{MAX} increased marginally with an increase in superheat while increased considerably with increase in subcooling. It increased from 0.7796 at 0 degree of subcooling and superheating to 0.8488 at 6 degree of subcooling in both circuit and to 0.7819 at 6 degree of superheating in both circuit.

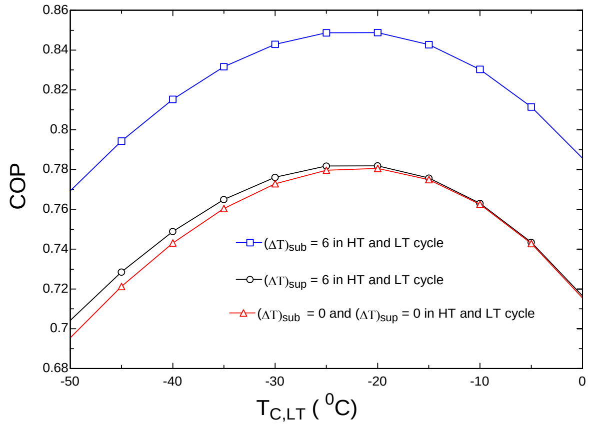

Fig.8 Effect of low temperature circuit condensing temperature on COP at different degree of subcooling and superheating

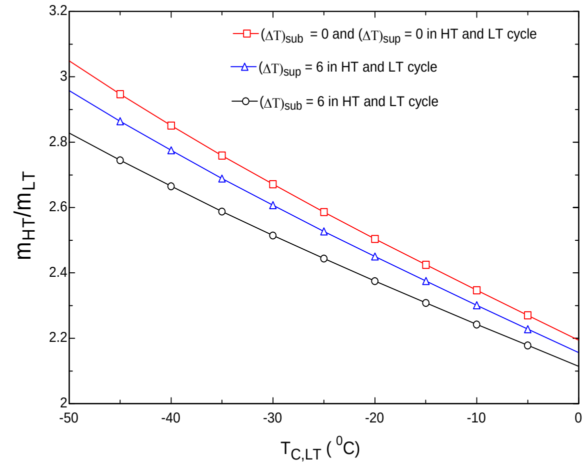

Fig.9 Effect of low temperature stage condensing temperature on mass flow ratio at different degree of subcooling and superheating

The mass flow ratio (m_{HT}/m_{LT}) decreased with rise in $T_{C,LT}$ at different degree of subcooling and superheating (Fig.9). Difference in value of mass flow rate reduces for different degree of subcooling and superheating at higher $T_{C,LT}$. Ratio of mass flow (m_{HT}/m_{LT}) reduced by more amount in case of subcooling from 0 ^{0}C to 6 ^{0}C than the superheating from 0 ^{0}C to 6 ^{0}C for fixed value of $T_{C,LT}$.

Fig.10 depicts variation in COP with $T_{C,LT}$ for different isentropic efficiency of both HT and LT compressors. Maximum COP shifts upward proportionally with an increase in isentropic efficiency.

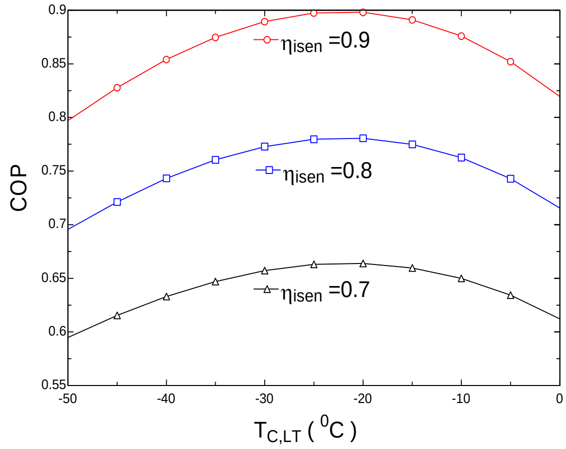

Fig.10 Effect of low temperature stage condensing temperature on COP at different isentropic efficiency of compressors

V. Conclusions

Thermodynamic analysis of R507A-R23 cascade refrigeration system have been presented to optimize the design and operating parameters of the system and following inferences are drawn.

1. The COP of the system increased from 0.7851 to 1.232 as low temperature circuit evaporator temperature ($T_{E,LT}$) is varied from -80 ^{0}C to -50 ^{0}C keeping others parameters constant.
2. The COP of system deteriorated from 0.9274 to 0.5486 when high temperature circuit condenser temperature ($T_{C,HT}$) is increased from 25 ^{0}C to 50 ^{0}C keeping other parameters constant.
3. The COP of high temperature circuit increased whereas the COP of low temperature circuit decreased when ($T_{C,LT}$) is varied from -50 ^{0}C to 0 ^{0}C. Hence the optimal value of ($T_{C,LT}$) where COP found maximum is -25 ^{0}C and corresponding maximum COP of system found 0.7805.
4. The COP of system reduced by about 17.10 % when temperature difference in cascade condenser was increased from 2.5 ^{0}C to 12.5 ^{0}C.
5. The COP rise of the system by higher amount in the case of subcooling in the both circuit than the subcooling separately in each circuit. COP of the system rises with superheating in HT circuit while falls with superheating in LT circuit assuming superheating of HT circuit refrigerant takes place in cascade condenser and superheating of LT circuit refrigerant takes place in evaporator due to heat transfer with surrounding

ISSN: 2517-9950
Vol:5, No:9, 2011

fluid. Similarly there is marginally rise in the COP of the system in the case of jointly superheating in both HT and LT circuit.

6. Maximum COP increased marginally with an increase in degree of superheat while increased considerably with increase in subcooling.

7. The mass flow ratio (m_{HT}/m_{LT}) decreased with rise in ($T_{C,LT}$) for different degree of subcooling and superheating.

8. Maximum COP shifts upward proportionally with an increase in isentropic efficiency of compressors.

NOMENCLATURE

COP	Coefficient of performance
H	Enthalphy (kJ/kg)
HFC	Hydrofluorocarbons
m	mass flow rate (kg/min)
ODP	Ozone depletion potential
Q	heat transfer (kJ)
RE	Refrigerating effect (W)
T	Temperature (^{0}C)
S	Entropy (kJ/kg K)
WD	Work done (W)

Greek symbol

η	efficiency
ε	effectiveness

Subscript

C	condenser
CC	Cascade condenser
E	evaporator
HT	high temperature
H	high
ise	isentropic
LT	low temperature
L	low
max	maximum
sub	subcooling
sup	superheating

REFERENCES

[1] Montreal protocol on substances that deplete the ozone layer. United Nations Environment programme (UNEP), 1987.
[2] UNEP Assessment report of the technology and economic assessment panel, UNEP Ozone Secretariat, Nairobi, Kenya.,2007
[3] Kruse,H.,Ru¨ ssmann,H. The natural fluid nitrous oxide - an option as substitute for low temperature synthetic refrigerants.International Journal of Refrigeration, 29 (5), 2006, 799–806.
[4] Nicola G.D., Giuliani G., Polonara F., Stryjek, R. Blends of carbon dioxide and HFCs as working fluids for the low temperature circuit in cascade refrigerating systems. International Journal of Refrigeration, vol.28, 2005, pp.130–140.
[5] Zhang, Y., Gong, M.Q., Zhu, H.B., Wu, J.F Vapor–liquid equilibrium data for the ethane + trifluoromethane system at temperatures from 188.31 K to 243.76 K. Journal of Chemical and Engineering Data, vol. 51 (4), 2006, pp. 1411–1414.
[6] Zhang, Y., Gong, M.Q., Zhu, H.B., Wu, J.F. Vapor–liquid equilibrium measurements and correlations for an azeotropic system of ethane + hexafluoroethane, Fluid Phase Equilibria vol. 240 (1), 2006, pp. 73–78.
[7] DiNicola, G., Giuliani, G., Polonara, F., Santori, G., Stryjek, R. Cascade Cycles Operating with CO2 + N2O Binary Systems as Low Temperature Fluid: Experimental Results, International Congress of Refrigeration, 2007, Beijing.
[8] Baolian Niu, Yufeng Zhang, Experimental study of the refrigeration cycle performance for the R744/R290 mixtures, International Journal of Refrigeration, vol. 30, 2007, pp.37-42.
[9] George L. Molenaar, Use of R-22/23 in lieu of R502/13 in a cascade refrigeration system, in: Proceedings of ASMENTC01 35th National Heat Transfer Conference, Anaheim, California (2001), pp. P243-P246.
[10] Keumnam Cho, Jonghoon Park, Honggi Cho, Performance of the cascade system using alternative refrigerants, in: Proceeding of ASMENTC01 35th National Heat Transfer Conference, Anaheim, California (2001), pp. P595-P600.
[11] Giovanni Di Nicola, Giuliano Giuliani, Fabio Polonara, Roman Stryjek, Blends of carbon dioxide and HFCs as working fluids for the low-temperature circuit in cascade refrigerating systems, Int J Refrigeration 28 (2005) 130-140.
[12] Robert Paul Wadell, Design of compact evaporators for ultra low temperature thermal management of microprocessors, Master thesis, Georgia Institute of Technology, 2005.