Theoretical Evaluation of Ejector- Based Heat Pump Cycles for cold climates

Ridha Ben Mansour 1 [a,], Mohamed Ouzzane [2 a] , Zine Aidoun [3 a]*
[a] CanmetENERGY, Natural Resources Canada, 1615 Lionel Boulet Varennes (QC) J3X 1S6 Canada

Abstract: In this study, different configurations of an ejector system assisted by solar energy or waste heat are proposed and investigated in order to increase the performance of residential air source heat pumps in cold climates. Besides the conventional heat pump which is used as a reference, three more configurations of compressor-ejector cycles were selected for this study: a hybrid ejector-compressor booster and two cascade compressor-ejector cycles. The modeling methodology developed to handle such systems is based on the principles of thermodynamics and fluid dynamics. This cycle model is used in combination with a validated ejector model and manufacturer's compressor performance maps. Conjugate effects of the thermal and mechanical compression components and sub-cycles interactions on the system performance are investigated. Performance analysis is assessed to compare these cycles. For fixed working conditions, substantial improvement in COP of the selected cycles over the conventional heat pump cycle is found.

Key Words: Ejector, heat pump, modeling, performance, low temperature

1 INTRODUCTION

Heat pumps, air conditioners and refrigerators are generally activated by electricity, generated in power stations burning fossil fuels, with a detrimental impact on the environment. An alternative approach with the potential of circumventing this negative effect may be based on hybrid ejection-compression systems. These can be activated by low quality thermal energy sources (renewable energy, industrial waste heat) and offer the opportunity to save electricity by rendering the system globally more efficient. A wide range of working fluids can be used, and in many cases, environmentally friendly refrigerants can be selected. Furthermore, compared to absorption systems or multi-stage mechanical compression systems, ejection-compression systems have gained more attention because of their relative simplicity, reliability and expected overall costs. Many authors have shown ejector systems may be used in different heat pump application such as heat upgrading, heating, cooling and refrigeration [Munday et al. (1977), Chen et al. (1994), Sun and Eames (1995) and Guo and Shen (2009), Chen et al., (2013)]. Although the most cited drawbacks for ejector cycle are the lack of knowledge on their performance over a wide range of operating conditions and their low energy efficiency when compared to mechanical vapour compression systems, the benefits are nonetheless obvious. Modeling-experiments integration represent therefore a valuable selection approach for the purpose of identifying the best technological options accounting for ejector and compressor characteristics, refrigerants properties and external conditions.

In this paper, different configurations of an ejector system assisted by solar energy or waste heat are proposed for study with the objective of achieving low temperature conditions heat pump operation. Modelling work performance analysis are performed to select promising configurations.

2 EJECTOR OPERATION AND PERFORMANCE

Supersonic ejectors are simple mechanical components (Fig. 1), which perform mixing and/or recompression of two fluid streams at different energy levels. Their operation principle is the same, regardless of the application. It is briefly described here but a more detailed account can be found in many journal publications [Ouzzane and Aidoun (2003)]. The primary stream having the highest total energy is motive (depicted A, on Fig. 1), the secondary stream with the lowest total energy (depicted B) is induced. Operation of such systems is quite simple: the motive stream (high pressure and temperature) flows through a convergent divergent nozzle to reach supersonic velocity. By an entrainment-induced effect, the secondary stream is drawn into the flow and accelerated. Mixing and recompression of the two streams then occurs in a mixing chamber, where complex interactions take place between the mixing layer and shocks. There is a mechanical energy transfer from the highest to the lowest energy level, with the mixing pressure lying between the motive and the induction pressures. Extensive experimental and theoretical work on ejectors in order to understand the fundamental mechanisms of the flow structure and the operational behaviour have been done [Keenan and Neuman (1946, 1950), Munday and Bagster (1977), Huang and al.(1998), Ouzzane and Aidoun (2003), Ablwaifa (2006) and Bartosiewicz et al.(2006)].
The focus of the present work is on the use of single phase gas ejectors powered by an external low quality heat source (waste, solar, ground heat) for performance enhancement of conventional heat pump cycles.

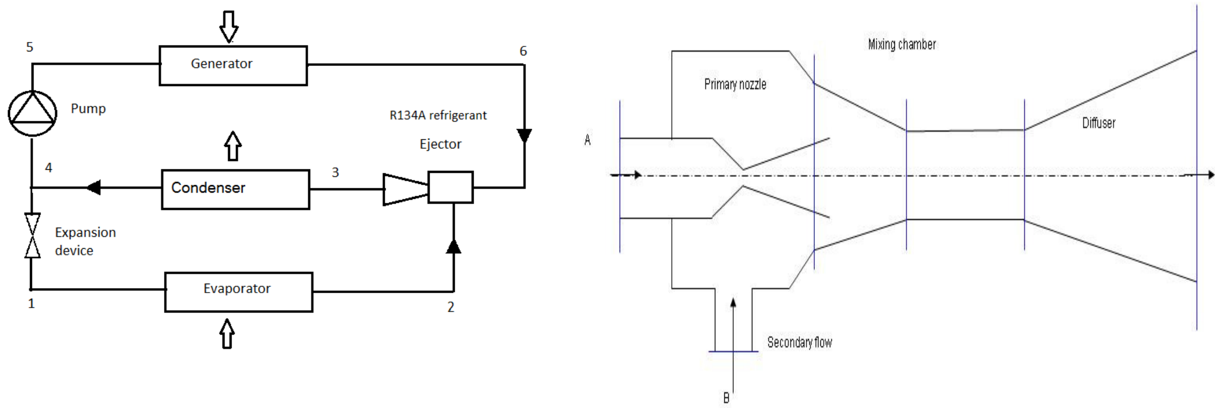

Figure 1: Schematic diagram of ejector geometry and cycle

2 EJECTOR CYCLES AND APPLICATIONS

Ejector heat pump systems have a large potential for using energy sources at low temperature. The simplest configuration of an ejector heat pump cycle is represented in Figure 1. In this application, the ejector replaces the electrically driven compressor, but uses heat rather than electricity to produce the compression effect. Compared with vapor compression systems, the ejector cooling systems have lower COP [Cardemil and Colle (2009)].

Combinations of ejector and compressor sub-cycles in order to form hybrid systems (solar collectors, absorption or mechanical compression) provide a considerable potential for cycle optimization and performance enhancement. As a result, ejector-compressor, ejector-absorption and ejector-solar collector cycles continue to receive increased attention in view of their positive impact on performance in exchange for relatively minor modifications to the overall system.

In the case of compressor-assisted ejector cycles, the work of Huang et al. (1998), Sun et al. (2009), Chen et al. (2013), Yan et al (2013) and those of Sokolov & Hershal (1990,1991,1993), are worth mentioning. Depending on the particular operating conditions and cycle configurations, reported COP improvements over conventional cycles ranged from 5 % to 40%. A good overview of the different potential applications of ejector-assisted cycles may be found in the article by Aidoun et al. (2011). Recently, Ben Mansour et al. (2014) have performed a study of the conjugate effects of ejector performance characteristics, the activation pressure-temperature conditions at the generator and the interaction with the compressor on refrigeration applications in hybrid compression-ejection cycles. Substantial performance improvement of the selected cycles over the conventional mechanical compression cycle was found in refrigeration applications and under cold climate conditions.

Our primary interest in this paper therefore, is the use of supersonic ejectors activated by an external heat source (solar or other) for optimising heat pump systems performance, through thermal and mechanical compression matching in the heating pumping context.

3 CYCLE DESCRIPTION AND MODELING

Ejector- assisted heat pump cycles modeling, based on thermodynamics and thermal hydraulics principles is used to study the combined effects of the energy transfer between several thermal and mechanical components, some of which are passive such as heat exchangers or expansion valves while others like the compressor and the ejector are respectively activated by mechanical and thermal sources of power. The interaction between all these components during the cycle operation is further complicated by the dynamic nature of the compressor and the static nature of the ejector, both being compression agents in the overall cycle and coupled directly or by means of an interface heat exchanger in order to compound their compression effect. The ejector model used in this study was inspired from Aidoun and Ouzzane (2003) who developed a 1-D design and simulation tool for ejectors. On the other hand, compressor models generally found in the refrigeration cycles literature are oversimplified, usually idealising important processes and neglecting others (compressions, internal leakage, the heat transfer, electro-mechanical losses). Furthermore, these models require very specific property data, only available from manufacturers. In this study, data for scroll compressors were used. Performance variables (power input, refrigerant, mass flow, motor current and compressor efficiency) are tabulated in terms of suction and discharge dew point temperatures. Inlet superheat to the compressor and subcooling leaving the condenser are held constant in the process of data generation for the performance maps. These maps facilitate system design in applications where different options are explored. Furthermore, they are useful in performance simulations.

For the purposes of the present study, the following cycles were considered in order to assess their potential for overall performance improvement:

3.1 Conventional heat pump cycle

Shown in Fig. 2, this is the simplest form of mechanical heat pump cycle. It differs from the basic ejector cycle shown in Fig. 1, by the ejector, the pump and the generator replacing the compressor. These cases are mentioned here as reference and for validation only, since they are abundantly described in the refrigeration literature.

3.2 Hybrid ejector-compression heat pump cycle

This system consists of a conventional ejector cycle and a booster, positioned between the ejector and the evaporator (Fig. 3). The booster is used to lift the refrigerant pressure in the evaporator to an intermediate level. The combined thermal-mechanical compression results in improved cycle performance.

3.3 Cascade systems

A cascade system consists of two independent circuits (conventional ejector cycle and mechanical compression cycle), thermally coupled by means of evaporator/condenser working at intermediate pressure conditions. Each circuit can have a different refrigerant, suitable for a specific working condition. Two configurations are assessed in this paper and they are represented in Figs 4 and 5, respectively.

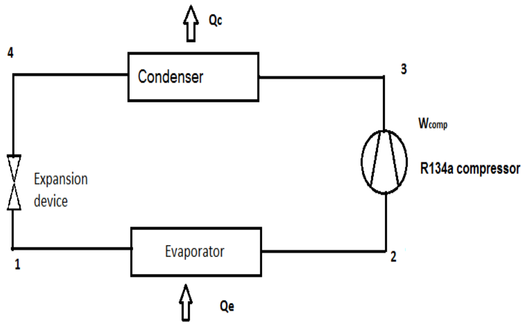

Figure 2: Schematic diagram of conventional heat pump cycle

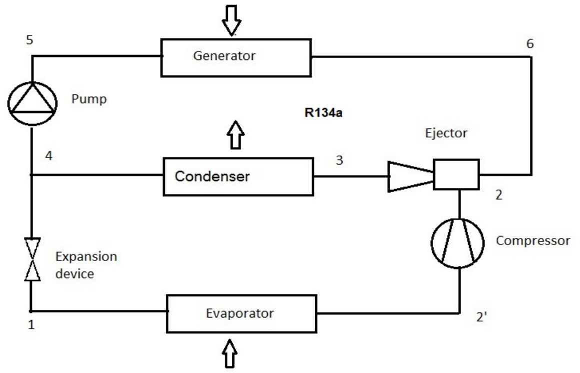

Figure 3: Schematic diagram of hybrid ejector-compression heat pump cycle

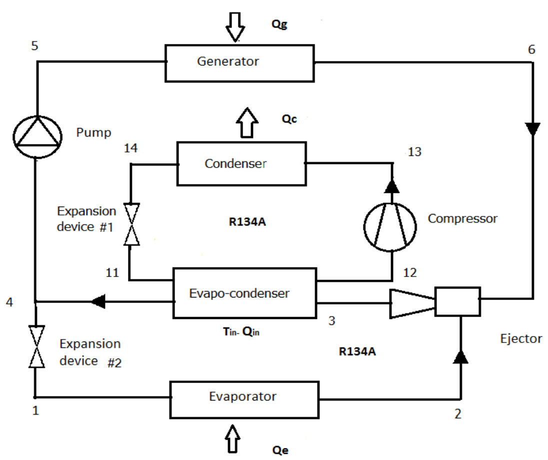

Figure 4: Schematic diagram of cascade system I

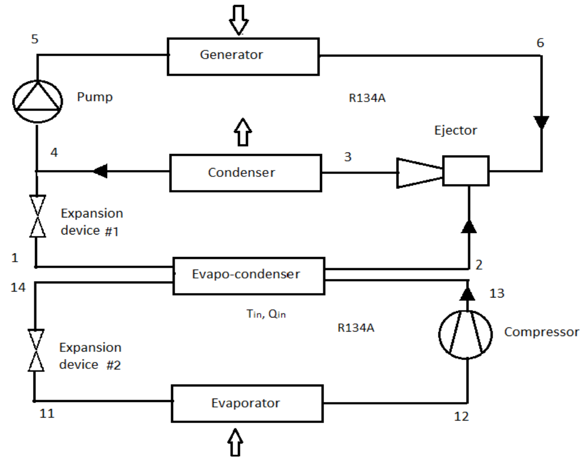

Figure 5: Schematic diagram of cascade system II

3.4 Cycle energy analysis

The main assumptions listed below are made for the simulation of all the cycles selected:

- The flow inside the ejector is one-dimensional and all the process is adiabatic.
- Mixing of primary and secondary fluids in the ejector mixing section is assumed to occur at constant pressure.
- To account for non ideal processes, the effect of frictional and mixing losses are taken into account by introducing some coefficients into isentropic relations
- Steady state conditions
- Moderate superheat at the evaporator and generator outlets
- Saturated condensate at the condenser outlet
- Expansion at constant enthalpy to the evaporator
- Pressure losses negligible

3.4.1 Cycle performance characteristics

Ejector performance is generally expressed in terms of the entrainment ratio (ω), the compression ratio (τ_{ej}) and the efficiency (η_{ej}) [(Dvorak and Vit(2005)] defined as:

$$\omega = \frac{\dot{m}_s}{\dot{m}_p} \tag{1}$$

$$\tau_{ej} = \frac{P_c}{P_e} \tag{2}$$

$$\eta_{ej} = \frac{\dot{m}_s}{\dot{m}_p} \frac{1-\left(\frac{P_e}{P_c}\right)^{\frac{\kappa-1}{\kappa}}}{\left(\frac{P_g}{P_c}\right)^{\frac{\kappa-1}{\kappa}}-1} \tag{3}$$

Where $\dot{m}$ and P are the mass flow rate and the pressure respectively; the subscripts e, c and g refer to evaporator, condenser and generator and κ is an adiabatic exponent.

In the case of a conventional heat pump cycle, the coefficient of performance (COP) is usually considered to account for electrical power consumption and defined as:

$$COP_I = \frac{Q_c}{W_{comp}} \tag{4}$$

For cycles integrating an ejector activated by an external heat source and a compressor, the COP_{el} is given by:

$$COP_{el} = \frac{Q_c}{W_{comp} + W_{pump}} \tag{5}$$

Where Q and W are the capacity the power and the subscripts comp and pump refer to compressor and pump.

It is worth mentioning that in the above definition, the driving thermal power (Q_g) is not account for and it is considered as free and abundant since it may be recovered from low quality heat sources (chemical and food processing plants or renewable energy).

In the present case, the mechanical coefficient of the performance (COP_{comp}) is adopted to evaluate the efficiency of mechanical compression unit.

$$COP_{comp} = \frac{Q_{c\_comp}}{W_{comp}} \tag{6}$$

Q_{c_comp} is the heating capacity provided by the mechanical compression unit.

3.4.2 **Modelling equations**

The ejector model used in this study is inspired from Ouzzane and Aidoun (2003) who developed a 1-D design and simulation tool for ejectors. Conservation equations for momentum, energy and mass are successively applied to a control volume in the primary and secondary nozzles, the mixing chamber, the constant section zone and the diffuser. These equations are supplemented by an entropy conservation condition and the use of NIST (Lemone et al. (2013) database and subroutines for the calculation of refrigerants properties Thus, for any control volume, the following equations are applied:

Momentum conservation: $$\sum (PA + \dot{m}V)_i = \sum (PA + \dot{m}V)_o \tag{7}$$

Energy conservation: $$\sum \dot{m}_i (h_i + \frac{V_i^2}{2}) = \sum \dot{m}_o (h_o + \frac{V_o^2}{2}) \tag{8}$$

Mass conservation: $$\sum \dot{m}_i = \sum \dot{m}_o \Leftrightarrow \sum (\rho VA)_i = \sum (\rho VA)_o \tag{9}$$

For the other components forming the considered systems, the basic equations are obtained from the conservation laws of energy and mass. For the sake of concision, balance equations for the cascade system II, are summarized in Table 1.

Table 1: Balance equations for the heating cascade system II Fig. 5)

Component	Mass balance	Energy balance	Eq.
Condenser	$\dot{m}_3 = \dot{m}_s + \dot{m}_p$	$Q_c = (\dot{m}_p + \dot{m}_s)(h_3 - h_4)$	(10.a)
Evaporator-condenser	$\dot{m}_{comp} = \dot{m}_{14}; \dot{m}_1 = \dot{m}_s$	$Q_{int} = \dot{m}_{comp}(h_{13} - h_{14}) = \dot{m}_s(h_2 - h_1)$	(10.b)
Evaporator	$\dot{m}_{comp} = \dot{m}_1$	$Q_e = \dot{m}_{comp}(h_{13} - h_{12})$	(10.c)
Expansion device #1	$\dot{m}_{14} = \dot{m}_{comp}$	$h_{14} = h_{11}$	(10.d)
Compressor	$\dot{m}_{comp} = \dot{m}_{13}$	$W_{comp} = \dot{m}_{comp}(h_{13} - h_{12})$	(10.e)
Generator	$\dot{m}_5 = \dot{m}_6 = \dot{m}_p$	$Q_g = \dot{m}_p(h_6 - h_5)$	(10.f)
Expansion device #2	$\dot{m}_4 = \dot{m}_1 = \dot{m}_s$	$h_4 = h_1$	(10.g)
Pump	$\dot{m}_5 = \dot{m}_p$	$W_{pump} = \dot{m}_p(h_5 - h_4)$	(10.h)
Ejector	$\dot{m}_3 = \dot{m}_s + \dot{m}_p$	$(\dot{m}_p + \dot{m}_s)h_3 = \dot{m}_p h_6 + \dot{m}_s h_2$	(10.i)

3.4.3 **Solution methodology and validation**

The studied cycles differ in complexity, depending on their configuration. Cascade systems I and II are thermally coupled through an interface heat exchanger. In the hybrid system, the compressor and the ejector are arranged in series and matched directly, the state of the refrigerant at the compressor outlet corresponding to that of the ejector inlet. The solution procedure is similar for all the cases and Cascade system I is taken as a typical example to detail the procedure. It is necessary to first select the compressor that can match the ejector capacity for the operating conditions in the cycle and impose a limit to the external heat source i.e. Q_g <65%.Q_c. For these conditions, the main characteristics of the potential compressor used in the different cycles are presented in table 2.

The flow chart of the main program is shown in Figure 6 and presents the common structure to these cycles. Inputs are the refrigerant type, the heat pump capacity and source and sink temperatures. Outputs of the analysis include the geometric parameters of the ejector and the overall system performance. A more detailed account of the ejector-assisted cycles model and the resolution procedure can be found in the work of Ben Mansour et al. (2014).

The model and the solution procedure are validated with the experimental results of an ejector cooling machine operating, with refrigerant R245fa and presented by Huang et al.(2011). Figure 6a-b represents the entrainment ratio (ω) and the performance coefficient $Q_e/(W_{pump}+Q_g)$ comparison of the present model with Huang et al.'s experimental data (2011). The predictions of the present model agree fairly well with the experimental data above, the largest discrepancy being less than 15 %.

Table 2: Characteristics of the scroll compressors for heating application (R134A, T_e = -20 ^OC and T_c = 50 ^OC)

COMPRESSOR	CYCLE	DISPLACEMENT VOLUME ($M^3.H^{-1}$)	W_{COMP} (KW)	Q_C (KW)
I	Conventional heat pump	28.8	4.5	10.15
II	Other systems	14.4	2.11	4.94

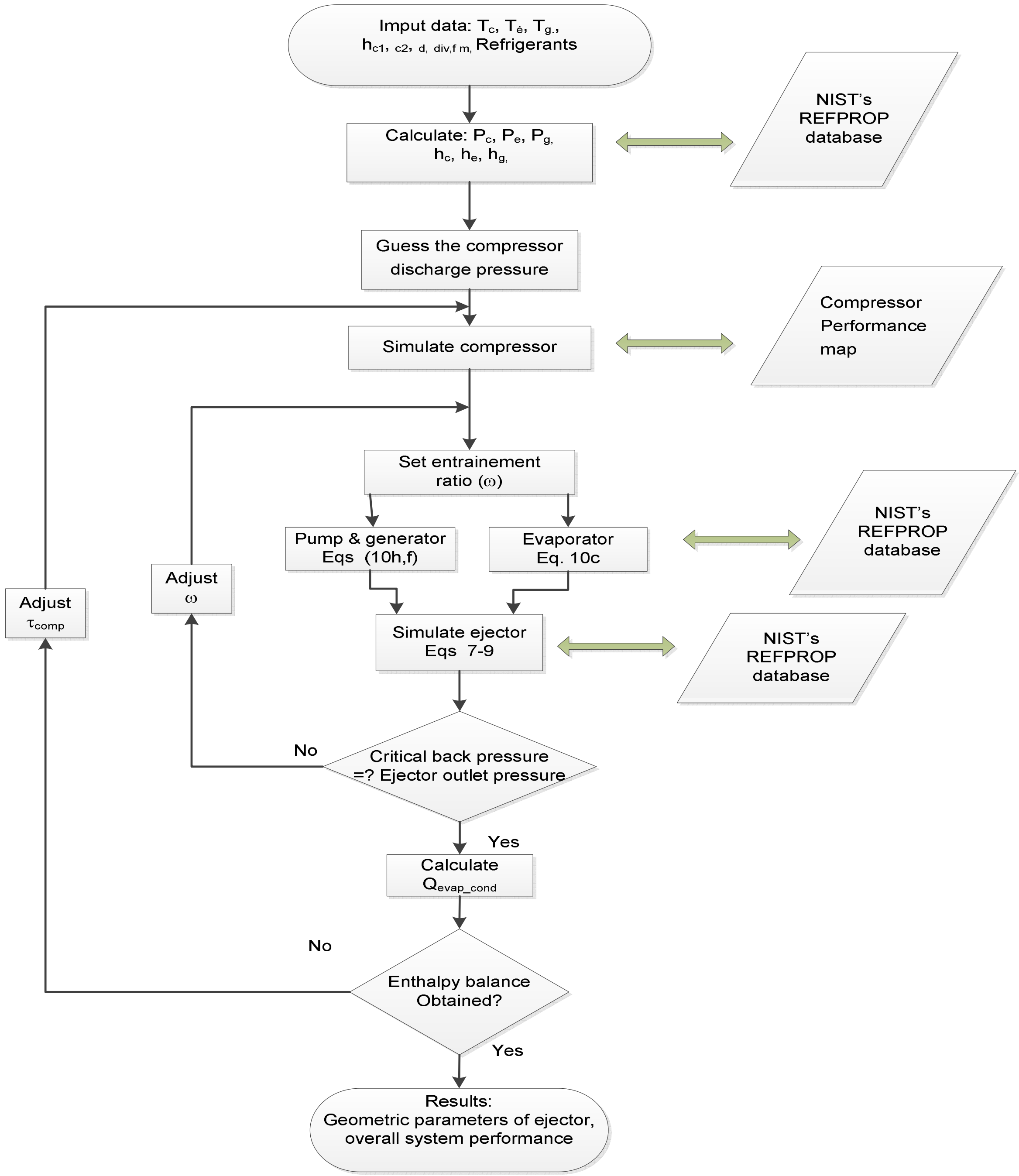

Figure 6 : Flow chart of the simulation program

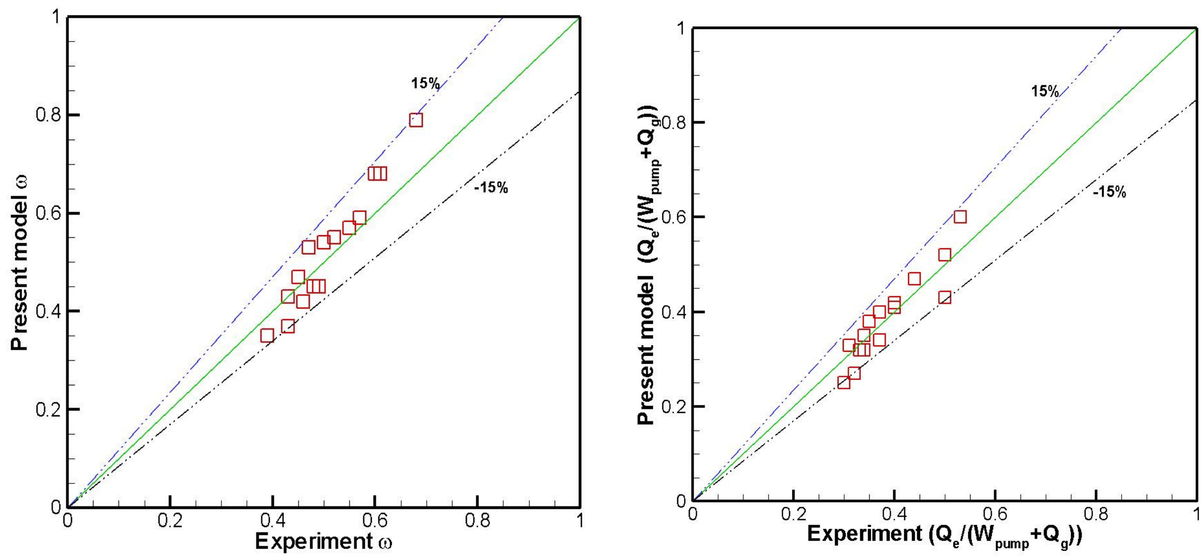

Figure 6 : Comparison of model predictions with Huang et al.'s experiments(2011) :
(a) comparison of entrainment ratio, and
(b) comparison of performance ratio ($Q_e/(W_{pump}+Q_g)$) (90≤ T_g≤110, 28.6≤ T_c≤38.7, 12≤ T_c≤20 and 3.4≤ Q_e≤5.2)

4 RESULTS AND DISCUSSION

The results of the present investigations are presented in Fig 7 and 8 and in Table 3. Several quantities determining the performance of the ejector cycles are investigated under the nominal conditions (Q_c =10.4 kW; T_c = 50 °C, T_e = -20 °C, T_g = 90 °C and ΔT_{int} =5 °C) using R134a as the working fluid for the ejector and the mechanical compression loops.
ΔT_{int} is the temperature difference in cascade heat exchanger (°C).

4.1 Cycles performance comparison

When the temperature lift (T_c-T_e) (i.e. cold climate) required from heating system is high, recourse is made to multi-staging. The COP improves but it comes with additional costs due to more components and complexity. A hybrid thermal-mechanical cycle integrating an ejector and a compressor may provide under some conditions, a far less complex, less costly alternative to multi-staging and far better performing machine than conventional compression. A comparative study of these alternatives will be performed herein, taking the established conventional mechanical system as a reference.

Cycle performance is assessed under the same operating conditions mentioned above. Because the operation physics of the mechanical compression cycle is dynamic as opposed to the cascade cycle which is partly dynamic due to the compressor and partly thermal and static due to the ejector, the resulting overall behaviour shows several differences and conflicting trends as pointed out earlier. For a representative and balanced comparison the performance of the cycles was calculated and then compared to that of the conventional heat pump for the same conditions and the same refrigerant (R134A for ejector and mechanical units). The results are summarised in Table 3 in terms of representative performance parameters commonly employed in heat pumping and refrigeration systems.

It can be seen from this table that he efficiency of the direct heating system is lower than that of other configurations because the heating capacity provided is more important than the energy consumed for all the selected systems.
In the hybrid system and in other cascade systems, a smaller compressor than for the conventional cycle is used, i.e. Swept volume decrease by 50% (Table 2). In addition, the

compression ratio (τ_{comp}) for the hybrid system decreases substantially, so that the compressor work is decreased nearly by 60%. Consequently, the electrical coefficient of performance (COP_{el}) for this system is improved by 136 % over the reference for the same heating capacity. It's important to remember that no explicit account was taken of the effect of the driving thermal power (Q_g) on the evaluation of COP_{el} (Eq.5)

The cascade system I shows the compression ratio (τ_{comp}) decreases substantially, nearly by 56 %. As a result, the electrical coefficient of performance is increased by 71% over the mechanical heat pump. On the other hand, the results show that for this cycle the evaporating capacity (Q_e) is lowest in comparison to the other cycles. This is due to the fact that as the ejector compressor ratio (τ_{ej}) increases, mass flow rate and capacity through the evaporator decrease. Consequently, the cascade system I absorbed less heat (1.4 kW) from cold source at low temperature (air) to provide sufficient heating.

For the cascade system II and under the same operating conditions mentioned above, the compressor uses less flow rate and consumes 54% work less than the conventional heat pump, resulting in 113% enhancement electrical energy efficient. Moreover, this cycle can extract more heat from the cold source at low temperature (air) and consequently can deliver more heat to the condenser. On the other hand, the driving thermal power (Q_g) in this case provides only 50% of the total heating capacity (10.2 kW).

Table 3 : System performances at on-design conditions

	CONVENTIONAL (COMPRESSOR I)	HYBRID (COMPRESSOR II)	CASCADE SYSTEM I (COMPRESSOR II)	CASCADE SYSTEM II (COMPRESSOR II)
ω	-	0.72	0.271	1.022
τ_{ej}	-	1.24	2.69	1.89
τ_{comp}	9.93	8	4.35	9.5
Q_e(kW)	5.9	2.88	1.4	3.02
Q_{e_comp}(kW)	5.9	3.23	7.69	3.02
Q_c(kW)	10.15	10.22	10.27	10.18
Q_{c_comp}(kW)	10.15	5.08	10.27	4.97
Q_g(kW)	-	5.54	6.4	5.16
W_{comp}(kW)	4.5	1.85	2.59	2.06
W_{pump}(kW)	-	0.06	0.06	0.05
COP_{el}	2.26	5.35	3.88	4.82
ΔCOP_{el} (%)	-	+136%	+71%	+113%
COP_{comp}	2.26	2.74	3.96	2.41
ΔCOP_{comp} (%)	-	+21%	+75%	+6%

Regarding the mechanical performance, cascade system I presents better mechanical performance potential (COP_{comp}) and has the highest mechanical heating capacity (Q_{c_comp}). On the other hand, the extraction of heat from the interfacial heat exchange (Q_{e_comp}) is the most important which is nearby 75% of the heating capacity.

The hybrid system and the cascade systems show very interesting performance features. However the cascade cycle may turn out to be easier to control since the ejector and compressor loops are physically separate. Another advantage is the possibility to use two different refrigerants which provide more flexibility for performance and operational improvements. Moreover and among the cascade cycles studied, cascade system II presents better performance and can extract more heat from the cold source (air). For these raisons, cascade system II is assessed here in more detail.

4.2 Effect of interface temperature on performance cascade system II

For the conditions already set (T_c = 50 °C, T_e = -20 °C and T_g = 90 °C and ΔT_{int} =5 °C), all operation and performance parameters can be obtained such as the ejector optimal geometry, the heating capacity, the compressor performance, primary and secondary mass flow rates and the COPs. Key parameters are discussed here since they represent the overall behaviour of the cycle and can be used as a common evaluation measure for differently configured systems. Figures 8a. and b. represent the effect of the intermediate temperature (T_{int}) on the COP and ejector efficiency for compressor II. Fig. 8a. shows that for fixed conditions, COPs and Q_c decrease with increasing intermediate temperature (T_{int}). This can be expected because an increase in this value means a higher lift for the compressor i.e. increasing the compressor discharge pressure, and a smaller one for the ejector (see Fig 8b.). This also increases the entrainment ratio (ω) and decreases the heating capacity (Q_c) by decreasing the refrigerant mass flow rate through the condenser. On the other hand Fig. 8b. displays ejector efficiency increases with intermediate temperature (T_{int}). It is known that ejectors are more efficient with low compression while the opposite is true for the compressors

With regard to the contribution of the driving thermal power (Q_g) on the total heating capacity (Q_c), it can be clearly observed in Fig 8a, that this contribution is more important for small values of intermediate temperature (T_{int}) and decreases with (T_{int}).Thus, for T_{int} = 27 °C, Q_g provides 92% of the heating capacity (Q_c) while for T_{int} = 48 °C, Q_g provides only 50%.

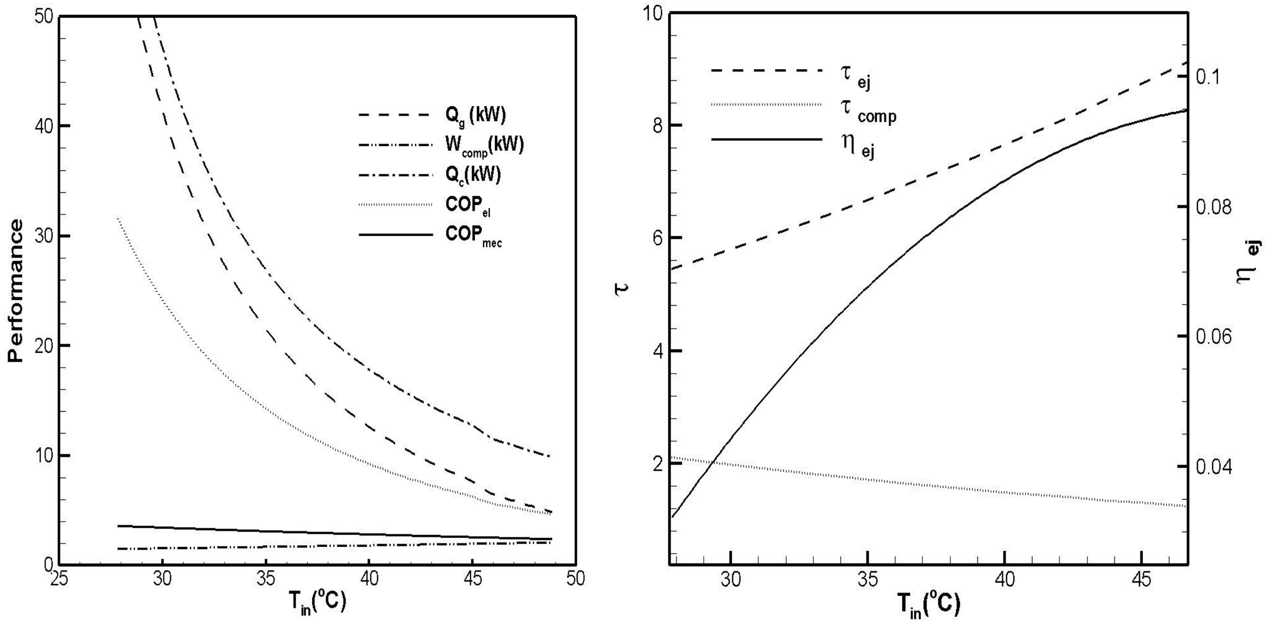

Figure 8: Effect of intermediate temperature (T_{in}) on performance of :

a) the cascade system II and b) the ejector

5 CONCLUSION

In this paper, different configurations of an ejector system assisted by solar energy or waste heat are proposed and investigated, in order to increase the performance of residential air source heat pumps in cold climate conditions. The model employed combines the use of thermodynamics and energy analysis of the loops making the overall cycle as well as the modeling of the ejector which is based on fluid flow analysis. Its application has allowed us to analyse three thermodynamic cycles and compare them to a conventional compression cycle: a hybrid ejector compression system and two cascade systems. The numerical results have provided useful information on cycles operation and conjugate effects of the thermal and mechanical compression components interactions in heat pumps integrating ejectors. For a representative performance comparison of these cycles a conventional heat pump cycle for the same conditions and the same refrigerant was used as a reference (R134A for ejector and mechanical units). Hybrid ejector-compressor and cascade cycles are promising in terms of performance. The cascade cycle has an advantage on the hybrid cycle in terms of flexibility and control. Furthermore, cascade system II presents better performance than cascade system I and can extract more heat from the cold source (air). This paper demonstrates the potential of integrating ejectors in conventional heat pumps that will allow for substantial performance improvements in cold climate conditions.

6 REFERENCES

Ablwaifa A. 2006. A theoretical and experimental investigation of jet-pump refrigeration. Ph.D. thesis, Nottingham University, England, UK.

Aidoun Z., Giguère D., Scott D., Hosatte S., 2011. Ejector applications in refrigeration and heating: an overview of modeling, operation and recent developments. Paper No. 294, 10th IEA Heat Pump Conf., Tokyo, Japan.

Bartosiewicz Y, Aidoun Z, Mercadier Y., 2006. Numerical assessment of ejector operation for refrigeration applications based on CFD. Appl. Th. Eng.; 26: 604–12.

Ben Mansour R.B., Ouzzane M., Aidoun Z.,2014. “Numerical Evaluation of Ejector-Assisted Mechanical Compression Systems for Refrigeration Applications”. Int. journal of Refrigeration (article in press)

Cardemil JM, Colle S. A., 2012. general model for evaluation of vapor ejectors performance for application in refrigeration. Energy Convers. Manage ;64:79–86.

Chen F., Liu C., Yang J. ,1994. Supersonic flow in the second-throat ejector-diffuser system. Journal of Spacecraft and Rockets31: 123–9.

Chen,X. Omer, S. Worall, M. Riffat, S..2013. Recent development in ejector refrigeration technologies - a Review. Renewable and Sustainable Energy Reviews . 19: 629-651.

Dvorak V. and Vit , T. , 2005. Experimental and numerical study of constant area mixing”. Int : 16^{th}. Symp. On transport phenomena, Prague.

Guo J, Shen HG. 2009. Modeling solar-driven ejector refrigeration system offering air conditioning for office buildings. Energy Build.;41:175–81.

Huang B., Chang J M, Petrenko V A, Zhuk K B., 1998. A solar ejector cooling system using refrigerant R141b. Solar Energy; 64(4-6): 223-6.

Huang B.J., Chen S.L., Petrenko V.O., Shestopalov, K.O., 2011.Theoretical and experimental investigation of the performance characteristics of an ejector cooling machine operating with refrigerant R245fa. In: The 23rd IIR International Congress of Refrigeration, August 21e26, 2011, Prague, Czech Republic.

Keenan J H, Newman E P., 1946. A simple air ejector. J. Appl. Mech. Trans. ASME; 64:, A75-A81.

Keenan J. H., Neumann E. P., Lustwerk F., 1950. An investigation of ejector design by analysis and experiment. J. App. Mechanics.17: 299-341

Lemmon E.W., Huber M.L., McLinden M.O., 2013. NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties-REFPROP, Version 9.1, National Institute of Standards and Technology, Standard Reference Data Program, Gaithersburg,

Munday J. T. and Bagster D. F., 1977. A New Ejector Theory Applied to Steam Jet Refrigeration. Ind. Eng. Chem. Process Res. Dev.; 16(4): 442-9.

Ouzzane M. and Aidoun Z., 2003. Model development and numerical procedure for detailed ejector analysis and design, Applied Thermal Engineering. 23: 2337-51.

Sun D. and Eames I. W., 1995. Recent Development in the Design Theories and Applications of Ejector - a Review, Journal of the Institute of Energy. 68: 65-79.

Sun LM, Feng Y, Pons M., in Abdulateef J M, Sopian K, Alghoul M A, Sulaiman M Y., 2009. Review of solar-driven ejector refrigeration technologies. Renewable and sustainable Energy Reviews 13:1338-1349.

Sokolov M. and Hershgal D.,1990. Enhanced ejector refrigeration cycles powered by low grade heat. Part 1: Systems characterization. Int. J. of Refrigeration. 13: 351-6.

Sokolov M. and Hershgal D. 1991. Enhanced ejector refrigeration cycles powered by low grade heat. Part 3: Experimental results. Int. J. of Refrigeration. 14: 24-31.

Sokolov M. and Hershgal D. 1993. Solar powered compression enhanced ejector air conditioner. Solar Energy. 51: 183-94.

Yan J. Chai W. Zhao L. Li Y. and Lin Ch.. 2013. Performance evaluation of a combined ejector-vapor compression cycle. Renewable energy.Pp. 331-337.