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Performance Analysis of Natural-Refrigerants-Based Vortex Tube Expansion Refrigeration Cycles

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Abstract

The energetic analyses and comparison of three natural refrigerants, ammonia, propane and isobutane based vapour compression refrigeration cycles are presented in this article using a vortex tube as an expansion device. A simple thermodynamic model has been used for analyses of two vortex tube expansion refrigeration cycle layouts based on the Maurer model (1999) and the Keller model (1997). Effects of various operating and design parameters of the COP improvement using vortex tube instead of expansion valve are presented. Results show that the COP improvement over basic expansion cycle increases with increase in cycle temperature lift for both cycle layouts. The COP improvement of CYC1 can be realized for certain operating temperature combinations. Effects of design parameters on the performance improvement are negligible. Study shows that the COP improvement using vortex tube as an expansion device are dependent on the refrigerant varieties, operating conditions as well as cycle configurations. Using the vortex tube as an expansion device, isobutane yields a maximum COP improvement of 12.2% for CYC2 followed by propane (11.5% for CYC2), whereas ammonia yields negligible improvement for studies ranges.

Keywords: Vortex tube, expansion device, refrigeration cycle, natural refrigerant, COP improvement

1. Introduction

Performance improvement of refrigeration system by using suitable cycle modification became an important research area for energy conservation. Use of vortex tube as an expansion device in refrigeration system is one such modifications. Vortex tube, working on the Ranque–Hilsch effect, is simple, compact, light, quiet, has no moving parts and does not break or wear and therefore requires little maintenance [1]. Vortex tubes have been used for cooling of machine parts, firemen's suits, set solders, equipments in laboratories dealing with explosive chemicals, dehumidify gas samples, electric or electronic control cabinets, environmental chambers, food, and test temperature sensors, quick start-up of steam power generation, liquefaction of natural gas, temperature control of divers' air suppliers, manned underwater habitats, hyperbaric chambers, separating particles in the waste gas industry, cooling for low-temperature magic angle spinning nuclear magnetic resonance, nuclear reactors, etc [1]. The vortex tube is relatively inefficient as a stand-alone cooling device but it may become an important component of a refrigeration system when employed as an alternative to the conventional throttling valve [2]. Hopper and Ambrose [3] used vortex tube only for expansion of vapor after condenser. The authors tested the cycle with thirteen refrigerants and showed different improved performances. Collins and Lovelace [4] experimentally studied expansion of propane through vortex tube and showed that the temperature separation deteriorates, when the condition of the inlet fluid becomes a mixture of saturated liquid and vapor. Hence, the performance of a vapor compression refrigeration cycle cannot be augmented through the application of a conventional vortex tube. Both phase separation and temperature separation are required simultaneously in vortex tube used in vapor compression refrigeration cycle. This leads to development of new type of vortex tube with three outlets, in which, the liquid occurring during the expansion extracts from the flow and for that reason should not be able to influence the Ranque-Hilsch Effect [5].

According to initial research, a 5% improvement of the COP was calculated when using R134a, whereas improved COP values of 5.3% and 16.3% were calculated when using propane and carbon dioxide and no improvements were calculated for ammonia [5]. Li et al. [6] performed a thermodynamic analysis of vortex tube expansion transcritical CO_2 cycle showed the maximum increase in COP using a vortex tube, assuming ideal expansion process, was about 37% compared to the one using

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an isenthalpic expansion process. Christensen et al. [5] used CFD modelling for analysis of CO_2 expansion vortex tube. Sarkar [7] analyzed vortex tube expansion transcritical CO_2 refrigeration cycle with two cycle layouts based on simple thermodynamic model and showed moderate COP improvements. Xie et al. [8] and Liu an Jin [9] analysed CO_2 trans-critical two stage compression refrigeration cycle with vortex tube expansion and reported 2.4% to16.3% improvement over the cycle with expansion value. Liu et al. [10] theoretically showed that the performance of vortex tube expansion transcritical CO_2 heat pump cycle is better than that with throttle valve, and the COP improvement is 5.8%~13.9% at given conditions.

In the present investigation, detailed energetic analyses of three natural refrigerants ammonia, propane and isobutane (commonly used natural refrigerants for single stage compression refrigeration and heat pump applications [11]) based refrigeration cycles using vortex tube as an expansion device have been done. A simple thermodynamic model has been used for analyses with two different cycle layouts: one is based on the Maurer model [7] and another is based on the Keller model [7]. The comparisons based on cooling COPs and COP improvements over basic valve expansion cycle are presented as well.

2. Mathematical modeling and simulation

Two layouts have been used in the present study: CYC1 based on Maurer model (1999) and CYC2 based on Keller Model (1997). For CYC1, as shown in Fig. 1, in the vortex tube, the liquid is expanding from condenser pressure to evaporation pressure and divided into three fractions: saturated liquid (state 4), which is collected in a ring inside the vortex tube (100% separation efficiency), saturated vapor (state C) and superheated vapor (state H), which are created because of the Ranque-Hilsch effect. The saturated liquid and vapor are mixed again (state 6) and going through the evaporator to give useful cooling effect. The superheated vapor is cooled in the heat exchanger (desuperheater) to state 5 and mixed with the gas coming from the evaporator (state 7) before entering the compressor (state 1). It may be noted that sometimes vortex tube and heat exchanger can be combined as a cooled vortex tube.

For CYC2, as shown in Fig. 2, the liquid is evaporated in a two-stage expansion, since it is difficult to get liquid separation with vortex tube. The refrigerant is cooled in an intermediate cooler from state 3 to state 4. The refrigerant is then expanded to an intermediate pressure through a throttle valve and to a phase separator, where the vapor is separated to state 8 from the liquid at state 5. The vapor is then superheated to state 9 in the intermediate cooler and is expanded through the vortex tube, where it separates in a cold (state C) and warm (state H) fraction. The warm fraction must be warmer than the ambient to get advantage of the vortex tube. The warm fraction is then cooled to state 10 in a desuperheater and then mixed with the cold outlet of the vortex tube. The mixture is then mixed with the vapor from evaporator (state 7) before entering the compressor (state 1). It can be noted that the original Keller model (exit of desuperheater & cold stream is connected to evaporator inlet) is used only for ammonia.

The thermodynamic model proposed by Sarkar [7] has been used for the analyses of both cycle layouts. The following assumptions have been made for the analysis:

(i) Negligible pressure drop in all heat exchangers and the connection tubes.

(ii) Both mixing and separation processes are isobaric.

(iii) No heat loss/gain with the environment, except with fluids for cooling purpose.

(iv) The refrigerant condition at the evaporator outlet is dry saturated.

(v) The compression process is adiabatic but non-isentropic.

(vi) All the kinetic energies at the nozzle exit in the vortex tube are absorbed by the hot fluid only.

(vii) The flow inside the vortex tube is in steady state conditions.

(viii) No friction effect is considered in the vortex tube system.

(ix) Expansion through throttle valves is isenthalpic.

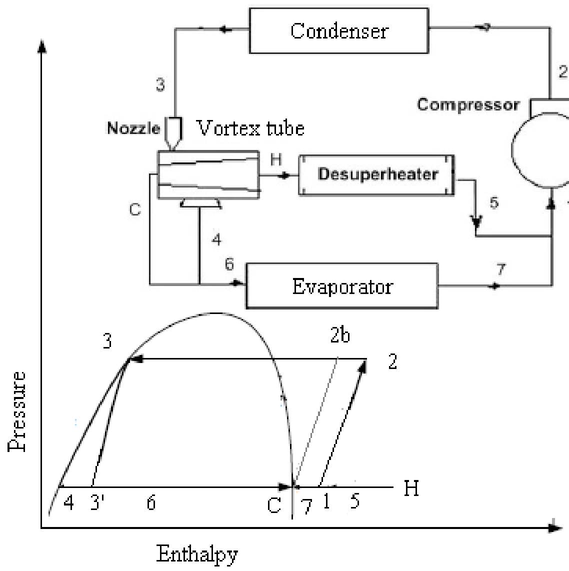

Fig. 1. Layout and p-h diagram of vortex tube expansion cycle (CYC1)

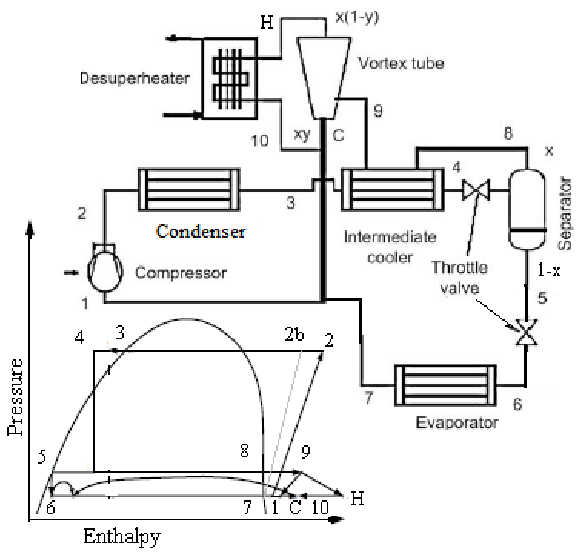

Fig. 2. Layout and p-h diagram of vortex tube expansion cycle (CYC2)

Using above assumptions, the equations for the vortex tube expansion refrigeration cycle were setup. Based on the theoretical model, the simulation code was developed to investigate the effect of different operating conditions for both cycles, which was integrated with the thermodynamic property subroutines developed earlier [12]. For given evaporator and condenser temperatures and component efficiencies, simulation procedures are described below:

For CYC1, properties at 3, 4 and C are calculated from property subroutine. The enthalpy at the vortex tube nozzle exit for given nozzle efficiency can be calculated by:

$$h_{3'} = h_3 - \eta_n \left[h_3 - h\left(p_{ev}, s_3 \right) \right] \tag{1}$$

Then the vapor quality is evaluated by $x = x(p_{ev}, h_{3'})$. For given cold mass fraction, amounts of separated liquid, cold fluid and hot fluid are (1-x), ($x*y$) and ($x*[1-y]$), respectively. Now, according to assumption, enthalpy at the hot end is given by,

$$h_H = \left(h_3 - (1-x)h_4 - xyh_C \right) / \left(x[1-y] \right) \tag{2}$$

State 5 can be found by using the effectiveness of desuperheater,

$$t_5 = t_H - \varepsilon(t_H - t_{wi}) \tag{3}$$

Inlet enthalpy of evaporator can be found by,

$$h_6 = \left([1-x]h_4 + xyh_C \right) / \left(1 - x + xy \right) \tag{4}$$

The inlet of the compressor is found by,

$$h_1 = (1 - x + xy)h_7 + x(1-y)h_5 \tag{5}$$

For the CYC2, properties at points 5 and 8 are evaluated using given intermediate pressure. Vapor quality and properties of points 4 and 9 are found by the iteration process using property code and following equations [7]:

$$\varepsilon = (t_9 - t_8)/(t_3 - t_8) \tag{6}$$

$$h_3 - h_4 = x(h_9 - h_8) \tag{7}$$

$$x = (h_4 - h_5)/(h_8 - h_5) \tag{8}$$

For given nozzle efficiency, similar to Eq. (1), enthalpy at state C can be calculated by:

$$h_C = h_9 - \eta_n \left(h_9 - h(p_{ev}, s_9) \right) \tag{9}$$

Properties at the hot end are given by,

$$h_H = \left(h_9 - yh_C \right) / \left(1 - y \right) \tag{10}$$

State 10 can be found by using the effectiveness of heat exchanger as similar to Eq. (3). Then the inlet of the compressor is found by,

$$h_1 = (1-x)h_7 + xyh_C + x(1-y)h_{10} \tag{11}$$

The compressor exit properties for both cycles are evaluated by using property subroutine and given compressor isentropic efficiency, which is given by [13],

$$\eta_{is,c} = 0.874 - 0.0134\left(p_{co} / p_{ev} \right) \tag{12}$$

The specific compressor work is given by,

$$w_c = h_2 - h_1 \tag{13}$$

The cooling output for the CYC1 and the CYC2, respectively, are given by,

$$q_{ev\_CYC1} = \left(1 - x + xy \right)\left(h_7 - h_6 \right) \tag{14}$$

$$q_{ev\_CYC2} = \left(1 - x \right)\left(h_7 - h_6 \right) \tag{15}$$

The COPs of vortex tube expansion refrigeration cycles and corresponding basic cycle have been evaluated by,

$$COP_{CYC1} = q_{ev\_CYC1} / w_c \tag{16}$$

$$COP_{CYC2} = q_{ev\_CYC2} / w_c \tag{17}$$

$$COP_b = \left(h_7 - h_3 \right) / \left(h_{2b} - h_7 \right) \tag{18}$$

The COP improvements using vortex tube have been evaluated by,

$$\Delta COP_{CYC1} = \left(COP_{CYC1} - COP_b \right) / COP_b \tag{19}$$

$$\Delta COP_{CYC2} = \left(COP_{CYC2} - COP_b \right) / COP_b \tag{20}$$

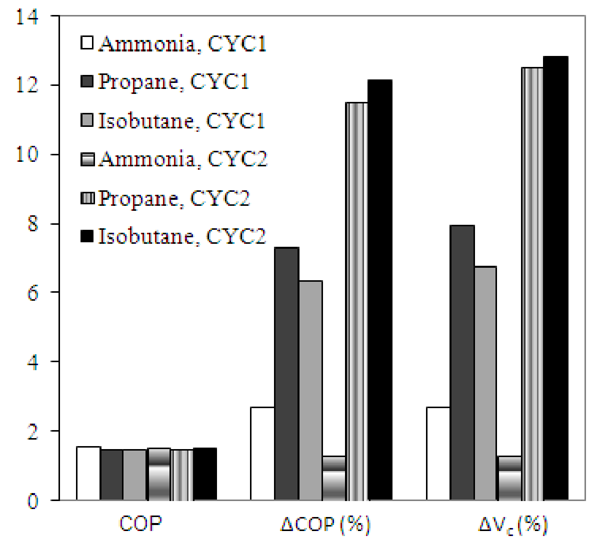

Fig. 3. Performance comparison at $t_{ev} = -20^oC$ and $t_{co} = 60^oC$

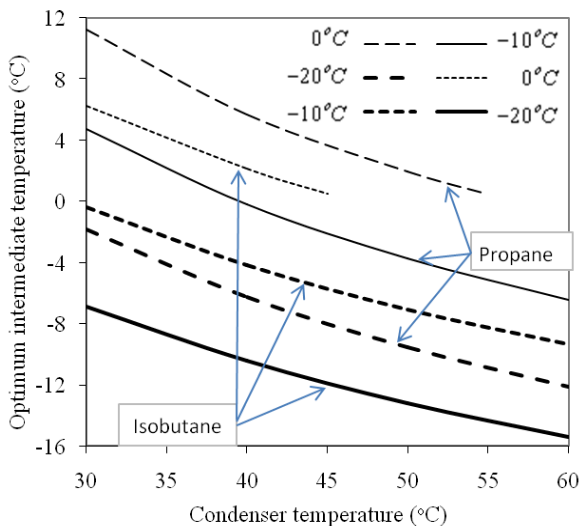

Fig. 4. Optimum intermediate temperature of CYC2

It may be noted that the present layouts (CYC1 and CYC2) are modified forms of Maurer and Keller models (5), respectively, proposed by Sarkar [7] to get higher performance for CO_2, which is also true for propane and isobutane. However, original models yield better performance for ammonia. Hence, CYC1 and CYC2 have been used for propane and isobutane, and original models have been used for ammonia in this study.

3. Results and discussion

To investigate the characteristics of the vortex tube expansion refrigeration cycles with ammonia, propane and isobutane as refrigerants, the vortex tube is assumed to have the isentropic efficiency of 0.8 for nozzle and cold mass fraction of 0.5. The heat exchanger effectiveness and water inlet temperature to the desuperheater have been taken as 0.85 and 27°C, respectively [7]. The performances (COP and volumetric capacity) of vortex tube-expansion refrigeration cycle are presented for various evaporator temperatures (–20°C to 10°C) and condenser temperatures (30°C to 60°C).

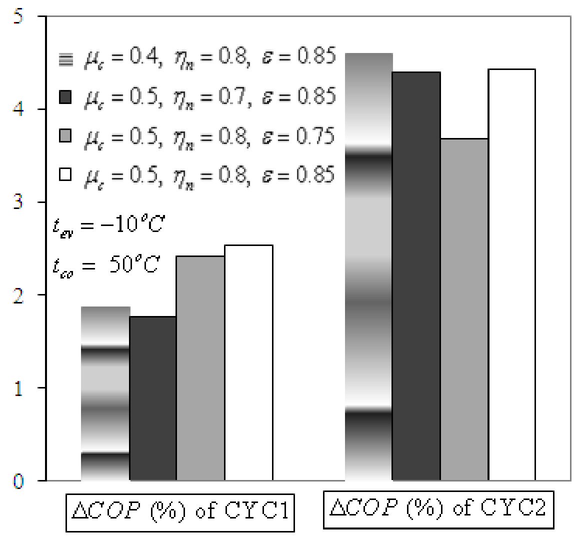

Fig. 5. Effects of various design parameters on performance for propane

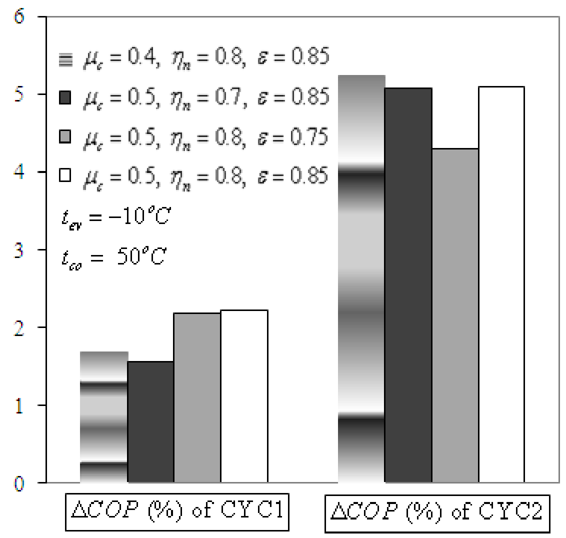

Fig. 6. Effects of various design parameters on performance for isobutane

Fig. 3 shows the performance comparison based on cooling COP, COP improvement and volumetric capacity improvement for ammonia, propane and isobutane with both cycle layouts at evaporator temperature of –20°C and condenser temperature of 60°C. It may be noted that for CYC2, the cooling COP as well as the COP improvement increase initially and then decrease with the increase in intermediate temperature and give some maximum value due to maximum heat rejection through desuperheater. This confirms the previous finding [5] that there exist an optimum intermediate temperature yielding maximum cooling COP and COP improvement and the present results are based on the optimum intermediate temperatures. Results (Fig. 3) show that the cooling COP of ammonia is still better than propane and isobutane. However, COP as well as volumetric cooling capacity improvements for ammonia are negligible compared to others and hence the effect of using vortex tube as expansion device in refrigeration cycle is negligible for ammonia. Hence, the foregoing results of vortex tube expansion refrigeration cycle are only for propane and isobutane. Comparison shows that both propane and isobutane yield very similar performances. Fig. 4 shows the optimum intermediate temperature variations for both propane and isobutane. As shown, at higher evaporator as well as higher condenser temperatures, optimum intermediate temperature reaches bellow the evaporator temperature, which is not feasible. Results show that the optimum intermediate temperature is dependent on both operating parameters and working fluid variety. The knowledge of optimum parameter will be useful for optimal cycle operation.

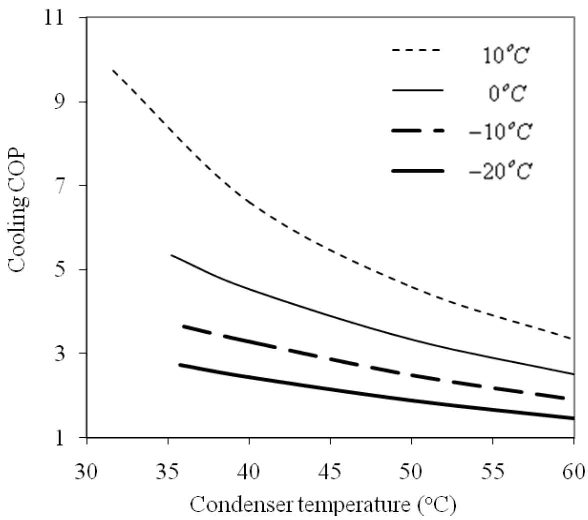

Fig. 7. Variation of COP of CYC1 at various evaporator temperatures for propane

Effects of the vortex tube cold mass fraction, nozzle isentropic efficiency and heat exchanger effectiveness on the performance improvements of both cycle layouts are shown in Fig. 5 and 6 for propane and isobutane, respectively. Results clearly show that the increase of all these parameters (individual component performances) will improve the cycle cooling COP for both layouts and also improve the COP improvement of CYC1; however, the effects on COP improvement are negligible for CYC2. Hence, the better individual component performance yields better system performance but not necessarily better improvement over the basic expansion refrigeration cycle.

Variations of cooling COP and COP improvement of CYC1 over basic isenthalpic expansion cycle with condenser temperature are predicted in Figs. 7 and 8, respectively, for propane at various evaporator temperatures. It is well known that the cooling COP increases with increase in evaporator temperature or decrease in condenser temperature for basic valve expansion cycle and similar trend is observed for vortex tube expansion cycle also. It may be noted that COP improvement varies from 0 to 7.4% for the given ranges. For the increase of condenser temperature or decrease of evaporation temperature, the vapor quality at nozzle exit increases and hence for the constant cold mass fraction, mass flow rate through the desuperheater increases and the heat loss though desuperheater increases for given water inlet temperature, which lead to more COP improvement. However the effect of condenser temperature is more predominant than that of evaporation temperature due more effect on vapor quality. For the high temperature lift, the use of vortex tube is more effective in term of higher COP improvement compared to basic cycle. It may be noted that the refrigerant inlet temperature to desuperheater is less than coolant inlet temperature for lower condenser temperature, which leads to no heat rejection and hence use of vortex tube become useless. Similar trends of cooling COP and COP improvement with evaporator and condenser temperatures can be found for CYC2 also as shown in Figs. 9 and 10. However, unlike to the transcritical system [7], the maximum cooling COP of CYC2 gives higher values (maximum of 4%) then CYC1. As shown, COP improvement varies from 0 to 11.5%. For the increase of condenser temperature or decrease of evaporator temperature as the optimum intermediate temperature decreases, the vapor quality increases and hence the mass flow rate through the heat exchanger increases, which lead to more COP improvement, as similar to CYC1.

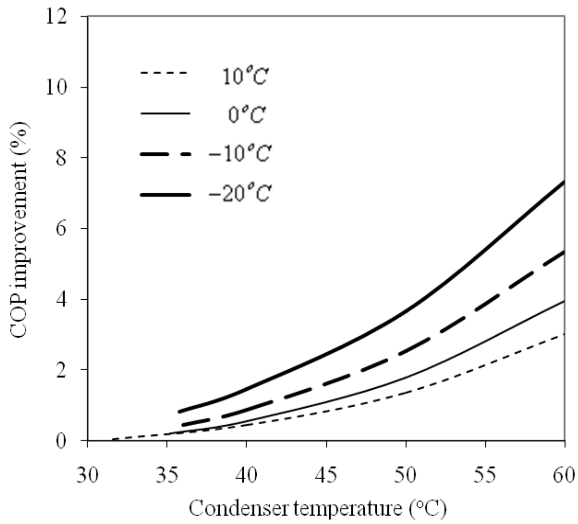

Fig. 8. COP improvement of CYC1 at various evaporator temperatures for propane

Variations of maximum cooling COP and COP improvement of CYC1 over basic isenthalpic expansion cycle with condenser temperature are shown in Figs. 11 and 12, respectively, for isobutane at various evaporator temperatures, and those for CYC2 are shown in Figs. 13 and 14, respectively. Variation trends of cooling COP for both cycles are similar with that for propane. However isobutane yields marginally higher values of COP (maximum deviation of 6%). Void portion in the Figs. 11 and 12 signifies that the vortex tube is not applicable at that lower condenser temperature for given evaporator temperatures and it is possible to get lower COP than basic expansion cycle. As discussed earlier, the optimization of intermediate temperature for CYC2, yields unfeasible values for higher evaporator as well as condenser temperatures and hence some fixed values have been chosen. The COP improvements also yield similar trends with evaporator and condenser temperatures for both CYC1 and CYC2. However, COP improvement values of isobutane are marginally more compared to propane.

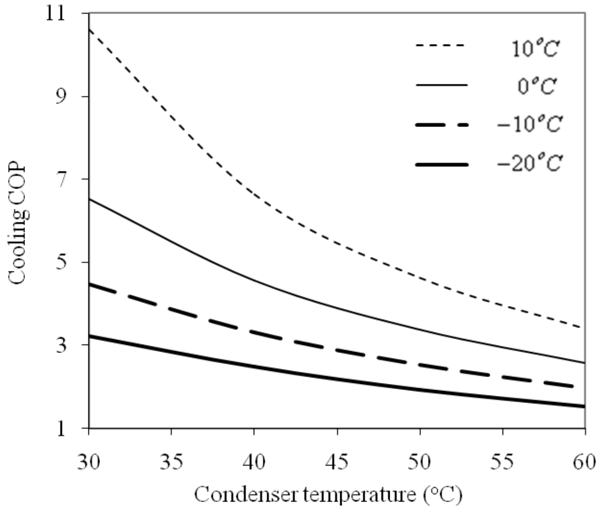

Fig. 9. Variation of COP of CYC2 at various evaporator temperatures for propane

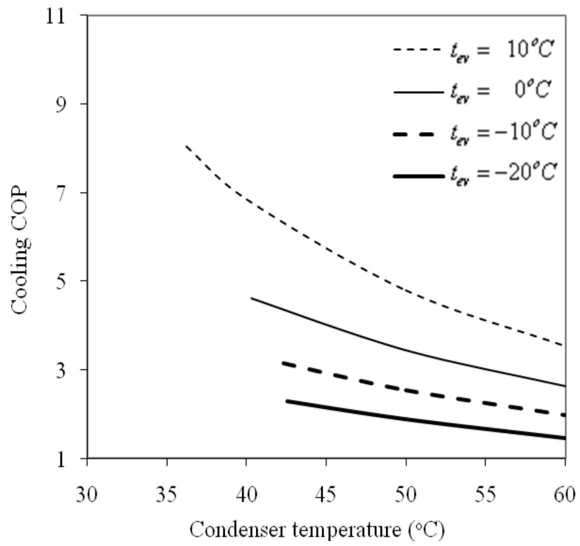

Fig. 11. Variation of cooling COP of CYC1 for isobutane

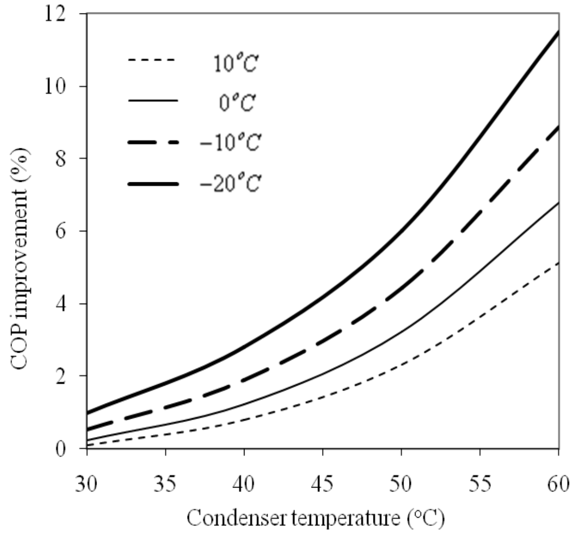

Fig. 10. COP improvement of CYC2 at various evaporator temperatures for propane

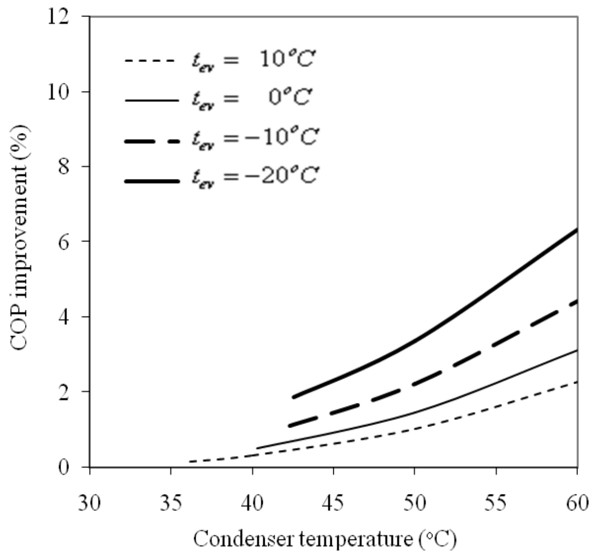

Fig. 12. Variation of COP improvement of CYC1 for isobutane

The energetic performance studies of the vortex tube expansion vapor compression refrigeration cycle with propane and isobutane as refrigerants show that the cycle performance are strongly dependent on the refrigerant varieties, cycle configurations as well as the operating conditions. The maximum performance improvement by using the vortex can be achieved in the case of CYC2 with isobutane, whereas minimum performance improvement can be achieved for CYC1 with propane. The optimum intermediate temperature of CYC2 for propane is more than that for isobutane. For the given ranges of study, the maximum COP improvement for CYC1 with isobutane is 6.4% and that for CYC1 with propane is 7.4%, for CYC2 with propane is 11.5% and for CYC2 with isobutane is 12.2% using the vortex as an expansion device in the vapour compression cycle. It may also noted that the CYC1 is less practically feasible compared to CYC1 due to phase separation difficulty in vortex tube [5].

3. Conclusion

Thermodynamic analyses and effects of various operating and design parameters on the COP improvement, followed by performance comparison of three natural refrigerants ammonia, propane and isobutane based vortex tube expansion refrigeration cycles with two different cycle layouts are presented in this study. Results show that the COP improvement over basic expansion cycle increases due to increase in heat rejection through desuperheater with the increase in condenser temperature and decrease in evaporator temperature for both cycle layouts. The optimum intermediate temperature of CYC2 increases with increase in both evaporator and condenser temperatures. Study shows that the COP improvement of CYC1 over basic cycle can be realized for certain combinations of evaporator and condenser temperatures. Effects of design parameters on system performance are significant, although negligible on performance improvement. Study shows that the COP improvement using vortex tube as expansion device are strongly dependent on the refrigerant varieties as well as cycle layouts. The maximum performance improvement by using vortex tube can be achieved in case of isobutane, whereas minimum performance improvement can be achieved for ammonia. Ammonia yields negligible COP improvement for both CYC1 and CYC2. For the studied operating ranges, CYC2 yields better performance improvement (maximum COP improvement of 11.5% for propane and 12.2% for isobutane) than CYC1 (maximum COP improvement of 7.4% for propane 6.4% for isobutane) using vortex tube as an expansion device in compression refrigeration system.

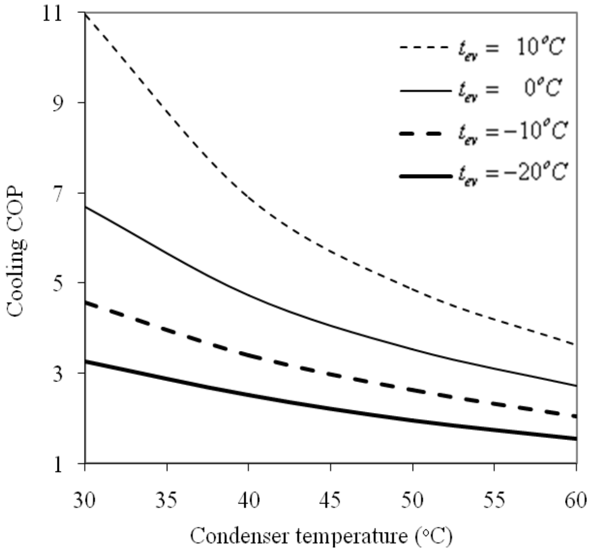

Fig. 13. Variation of cooling COP of CYC2 for isobutane

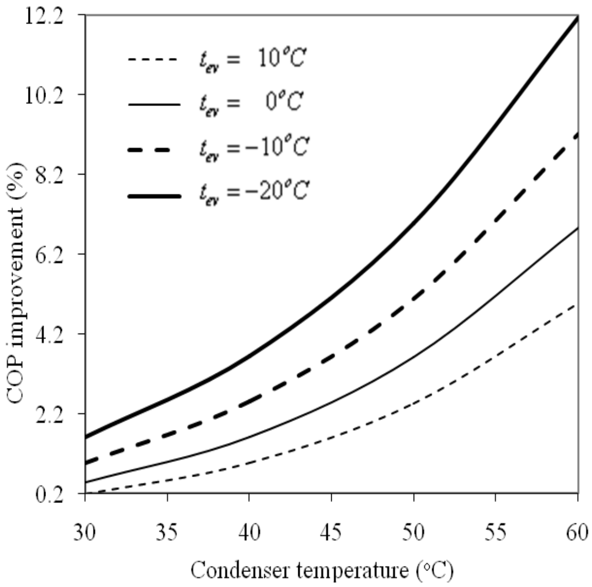

Fig. 14. Variation of COP improvement of CYC2 for isobutane

Nomenclature

COP	coefficient of performance
h	specific enthalpy, kJ/kg
p	pressure, kPa
q	cooling/heating effect, kJ/kg
t	temperature, °C
w	specific work, kJ/kg
x	vapor quality
y	cold mass fraction

Greek Symbols

ΔCOP	improvement of COP, %
ε	heat exchanger effectiveness
η	isentropic efficiency, %

Subscripts

b	basic cycle
c	compressor
C	cold outlet
co	condenser
ev	evaporator
H	hot outlet
n	vortex tube nozzle
wi	water inlet to desuperheater

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