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Thermodynamic Analysis of a Heat Pipe-thermal Jet Refrigeration System

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ABSTRACT
Presently the emphasis is to utilize renewable energy sources such as solar energy, geothermal energy, ocean thermal energy, etc. and waste heat from industrial processes, to produce refrigeration. The aim of many researchers was to develop thermal refrigerator and heat pump systems using heat energy as energy input for achieving better performance, compactness and cost effectiveness. In the present study, a simple thermal jet refrigerator is constituted into a heat pipe system and a thermodynamic analysis is presented for the performance prediction and design of such system. The effects of generator, evaporator and condenser temperature on the Coefficient of Performance (COP) of the system and entertainment ratio of the ejector are studied. It is found that COP and ejector entrainment ratio are higher at high generator and evaporator temperatures whereas they are lower at high condenser temperatures.

Key words: Thermal jet refrigerator, heat pump, heat pipe, thermodynamic analysis, COP, ejector entrainment ratio

INTRODUCTION
Research achievements in the area of two phase thermosyphon technology have led to the development of thermal jet refrigeration systems. Thermal jet refrigerator operating with refrigerants as well as non-azeotropic mixtures has got impressive research improvements. Ling (2004) suggested sub-cooled hot water as the working fluid because of its better performance. Ling and Groll (1994) studied hot water thermal jet refrigeration systems and observed that sub-cooled hot water is the potential driving medium to improve COP and for direct utilization in the solar cooling, district cooling and building air-conditioning. The suggested system employs two-phase flashing nozzle to convert the sub-cooled hot water directly to compressible vapor-liquid dispersed jet flow in the two-phase ejector so as to avoid partially or wholly the input heat of vaporization as in the conventional vapor jet refrigeration system. Nguyen *et al.* (2001) developed a vapor jet refrigerating system using water as the working medium and solar energy was utilized as the heat input. Smirnov and Kosoy (2001) proposed thermodynamic classification of refrigerating heat pipe concept with vapor jet compression. Parand *et al.* (2009) carried out a theoretical investigation of thermosyphon heat pipe behavior in transient regime. They developed a simple computer simulation tool to estimate the temperature of heat pipe as well as the time needed to reach steady state. In this study, a concept of combining the heat pipe and two-phase

thermosyphon principle with the sub-cooled hot water thermal jet refrigeration system is suggested and thus a new separate heat pipe jet refrigerator or the heat pump configuration is performed and analyzed. A thermodynamic analysis is presented for the performance prediction, as an extension of work done by Ling (2004).

HEAT PIPE-THERMAL JET REFRIGERATOR

A simple jet refrigeration consists of a generator, an evaporator, a condenser, expansion device, ejector and circulating pump as shown in Fig. 1. In this system, water vapor (primary fluid) from the generator at high pressure flows through the nozzle of the ejector and entrains the vapor (secondary fluid) from the evaporator at low pressure. The primary fluid and secondary fluid then mix in the mixing section and recover a pressure in the diffuser. The combined fluid flows to condenser where it condenses. Then the condensate is divided into two parts: one is pumped back to the generator and the other flows through the expansion device and enters the evaporator, where it is evaporated to vapor. The vapor is finally entrained into the ejector again, thus finishing the ejector refrigeration cycle. Within one cycle, when the system obtains heat from a heat source in the generator and work input from the pump, it produces the cooling effect in the evaporator and dissipates the heat to the environment through the condenser.

This simple system is constituted into heat pipe system as shown in Fig. 2. The heat pipe has the evaporation section in the upper end and the condensation section in the lower end with the adiabatic section in the middle. The adiabatic section includes a two-phase ejector and thin throttle tube which connects the evaporation chamber with the condensation chamber and the difference in pressure or vacuum are balanced by the gravity of the water column and the resistance in the working. The hot sub-cooled water is connected to the two phase nozzle through an internal tube embedded in the evaporator end. The water in the evaporator chamber is evaporated at the suction pressure created by the high two-phase jet from the nozzle exit. By the evaporation, the water in the heat exchange tube loop is cooled for the refrigeration or cooling usage. The two streams in the ejector are mixed in the mixing chamber and diffused to the pressure in the diffuser and directly flow into the condensation chamber. The cooling tube loop cools the condenser and the two-phase wet steam is condensed in the condenser by heat transfer to the water in the cooling tube loop and the water temperature is raised for the required warm water usage. The condensed water flows out by two parts: one part through the connection throttle tube and the other part through the pump in the loop and returned to the hot end or heater.

Thermodynamic analysis of heat pipe-thermal jet refrigerator: Figure 3 shows the thermodynamic process of heat pipe-thermal jet refrigeration system. P_B, P_E, P_C are, respectively

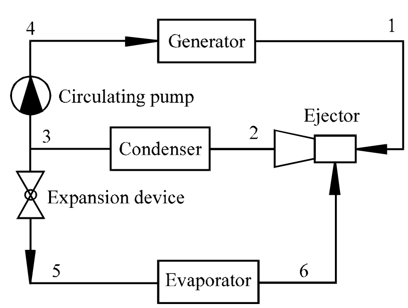

Fig. 1: Schematic diagram of a simple thermal jet refrigerator system

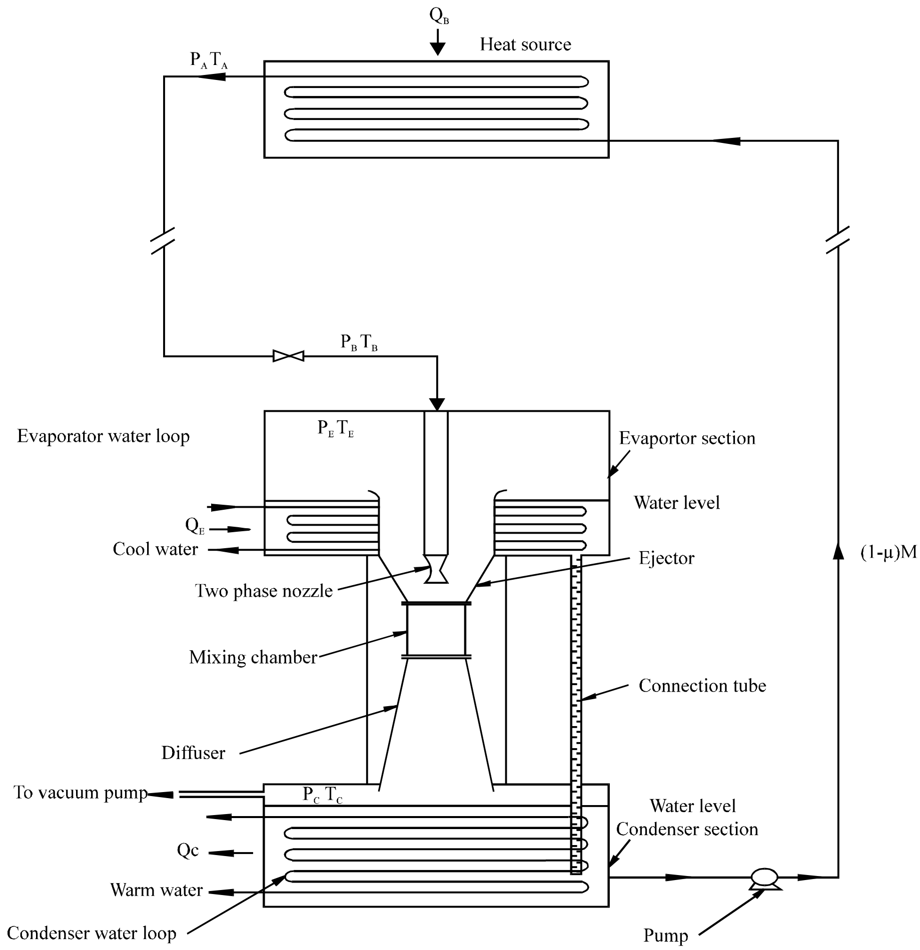

Fig. 2: Schematic diagram of heat pipe-thermal jet refrigerator system

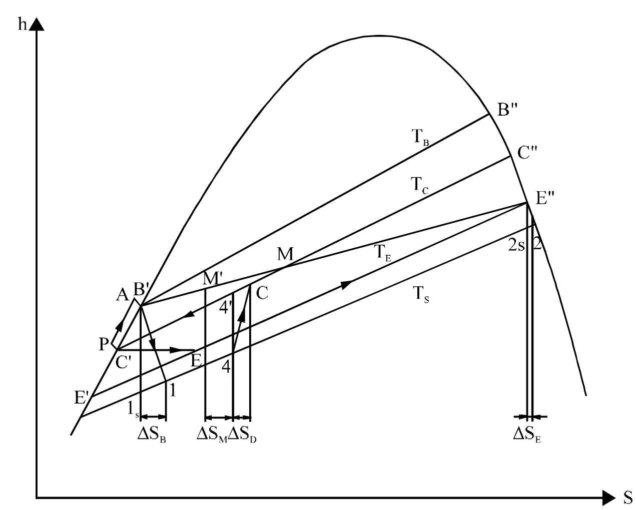

Fig. 3: H-S diagram of heat pipe thermal jet refrigeration system

the pressures of the driving sub-cooled hot fluid, the suction fluid from the evaporator section and at the exit of the diffuser. The fluid at A expands in the two-phase nozzle to P_S at point 1 through A→B'→1.

The evaporated steam at E″ is driven through the suction chamber, expands a little to P_S in the bell mouth section where it mixes with the high speed two-phase jet and experiences the momentum exchange and heat transfer in the mixing chamber. The real or irreversible final state at the exit is in point 4. Point C is the state at the exit of diffuser and the pressure is increased to P_C. C→C' is the process of condensation. One part of the condensate flows out of the heat pipe and pumped to point P and heated again by process P→B'. The other part under the pressure difference flows through the thin pipe (as throttled) to the evaporator section at point E where it is evaporated to E″ and to be sucked back again.

For an ideal and loss free mixing process the driving liquid at state B' mixed with the driven liquid at state E″ to the state M on the intersection of lines B'E″ and C'C″ with a flow rate ratio which is called the entrainment ratio of the ejector, expressed by:

$$\mu_{id} = M_E/M_B = (h'_B-h_M)/(h_M-h''_E) = (s'_B-s_M)/(s_M-s''_E) \tag{1}$$

From the thermodynamic relation:

$$h_M-h'_C = (s_M-s'_C)\ T_C \tag{2}$$

Combining Eq. 1 and 2, we can write:

$$h_M= [(h''_E-h'_B)\ (T_Cs'_C-h'_C)-T_C(s'_Bh''_E-s''_Eh'_B)]/[T_C(s''_E-s'_B)-(h''_E-h'_B)] \tag{3}$$

If the vapor content (or dryness) of two-phase flow mixture is defined as X for the ideal process of mixing:

$$X_M = (h_M-h'_C)/(h''_C-h'_C) = (s_M-s'_C)/(s''_C-s'_C) \tag{4}$$

The real turbulent mixing process is very complex. Due to the wall friction, momentum exchange and heat transfer etc., loss occurs and there is entropy increase. The final state at the exit of the diffuser is C. The vapor content at C is X_C. Thus we can define the dryness coefficient as:

$$\zeta_X= X_C/X_M \tag{5}$$

and

$$X_C = (h_C-h'_C)/(h''_C-h'_C) = (s_C-s'_C)/(s''_C-s'_C) \tag{6}$$

So:

$$h_C = h'_C+X_M\zeta_X(h''_C-h'_C) \tag{7}$$

From the energy balance relation of the ejector:

$$(M_B+M_E)h_C = (M_B h'_B)+(M_E h''_E) = (M_B+M_C)h'_C+X_C(M_B+M_E)r_C \quad (8)$$

Here, r_C denotes the heat of vaporization at T_C:

$$r_C = h''_C-h'_C \quad (9)$$

From the energy balance in the condenser section:

$$(M_B+M_E)\ (h_C-h'_C) = M_B(h'_B-h'_C)+M_E(h''_E-h'_C) \quad (10)$$

so:

$$\mu = [(2h'_B-h'_C)-(h_C+X_C r_C)]/[(h_C+X_C r_C)-(2h''_E-h'_C)] \quad (11)$$

PERFORMANCE ESTIMATION

From the three given temperatures, pressure P_A and assumed ζ_X, after getting the entrainment ratio or mass flow rate ratio μ, one can predict the performance of the thermal jet refrigeration system.

The mass flow rate of the suction flow or that circulating inside the secondary heat pipe is determined from the required cooling capacity Q_E:

$$M_E = Q_E/(h''_E-h_E) \quad (12)$$

And the mass flow rate of driving liquid:

$$M_B = M_E/\mu \quad (13)$$

The possible warm capacity is:

$$Q_C = (M_B+M_E)(h_C-h'_C) \quad (14)$$

The coefficient of performance of refrigerator is given by:

$$COP_{TRF} = Q_E/Q_B = \mu(h''_E-h'_E)/(h'_B-h'_C) \quad (15)$$

in which work of the pump is neglected.

The coefficient of performance of the heat pump is:

$$COP_{THP} = (1+\mu)(h_C-h'_C)/(h'_B-h'_C) \quad (16)$$

RESULTS AND DISCUSSION

The thermal jet refrigeration cycle is simulated by varying the input data viz., generator temperature from 80-90°C, condenser temperature from 30-40°C and evaporator temperature from 5-10°C. The capacity of the system is taken as 1 kW and dryness coefficient is assumed as 0.6.

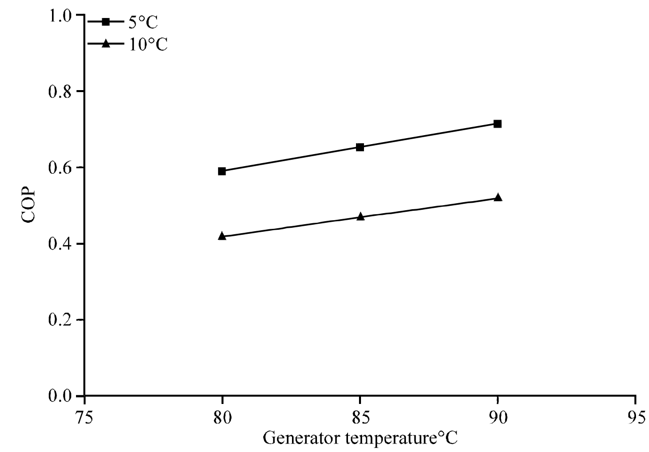

Fig. 4: Effect of generator temperature on coefficient of performance (COP) of thermal jet refrigerator at different evaporator temperatures, Refrigerator capacity: 1 kW, Condenser temperature: 30°C, Dryness coefficient: 0.6

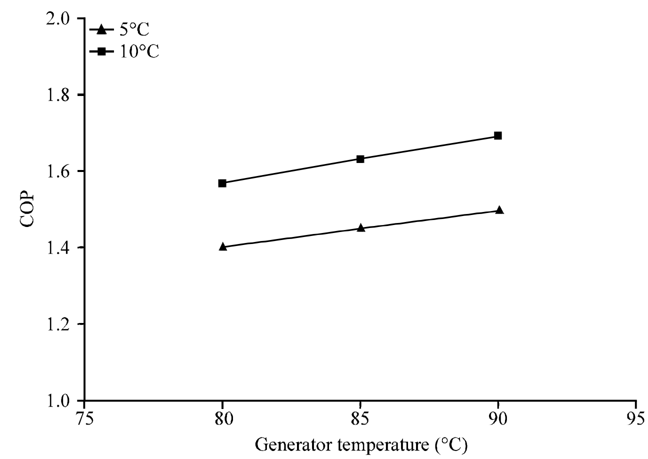

Fig. 5: Effect of generator temperature on coefficient of performance (COP) of thermal heat pump at different evaporator temperatures, Refrigerator capacity: 1 kW, Condenser temperature: 30°C, Dryness coefficient: 0.6

Thermodynamic properties of water viz., liquid and vapor enthalpy and entropy as functions of temperature have been calculated from the equations of Badr *et al.* (1990). Results have been obtained by means of a computer code written in the C^{++} programming language.

Figure 4 and 5 show the effect of generator temperature on COP of thermal jet refrigerator and heat pump, respectively for different evaporator temperatures. As the generator and evaporator temperature increase, COP also increases. At high generator temperatures, due to high liquid enthalpy, mass flow of hot water circulated (M_B) to produce constant refrigerating capacity is less which in turn decreases the amount of heat input required at the generator, resulting in higher COP. Similarly, at higher evaporator temperatures, due to high vapour enthalpy, mass flow rate of refrigerant water vapour (M_E) circulated is less for a constant refrigerating capacity which in turn decreases mass flow of hot water vapour in the ejector resulting in higher COP.

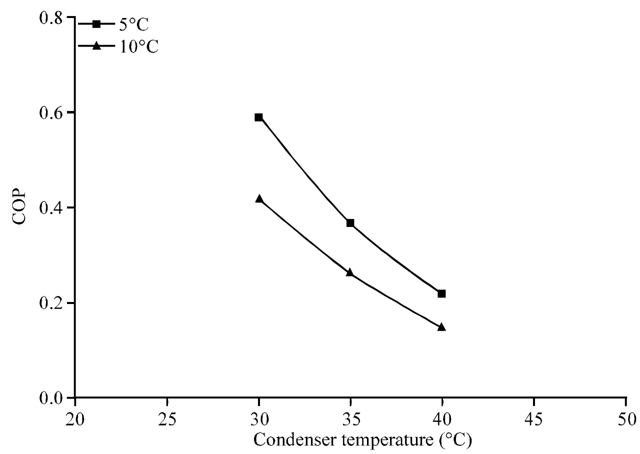

Fig. 6: Effect of condenser temperature on coefficient of performance (COP) of thermal jet refrigerator at different evaporator temperatures, Refrigerator capacity: 1 kW, Generator temperature: 80°C, Dryness coefficient: 0.6

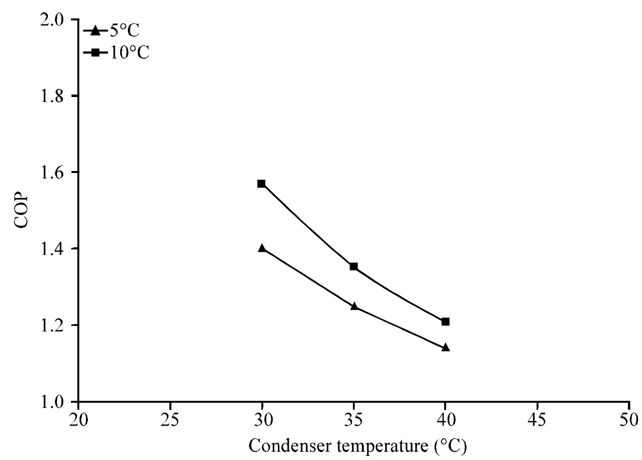

Fig. 7: Effect of condenser temperature on coefficient of performance (COP) of thermal heat pump at different evaporator temperatures, Refrigerator capacity: 1 kW, Generator temperature: 80°C, Dryness coefficient: 0.6

Figure 6 and 7 show the effect of condenser temperature on COP of thermal jet refrigerator and heat pump, respectively for different evaporator temperatures. As the condenser temperature increases, COP decreases. At higher condenser temperatures, due to high liquid enthalpy, mass flow of hot water circulated (M_B) to produce constant refrigerating capacity is more which in turn increases the amount of heat input required at the generator, resulting in lower COP.

Figure 8 depicts the effect of generator temperature on ejector entrainment ratio for different evaporator temperatures. As the generator and evaporator temperature increase, entrainment ratio also increases. As both temperatures increase, mass flow of hot water circulated (M_B) to produce constant refrigerating capacity decreases, resulting in higher entrainment ratio.

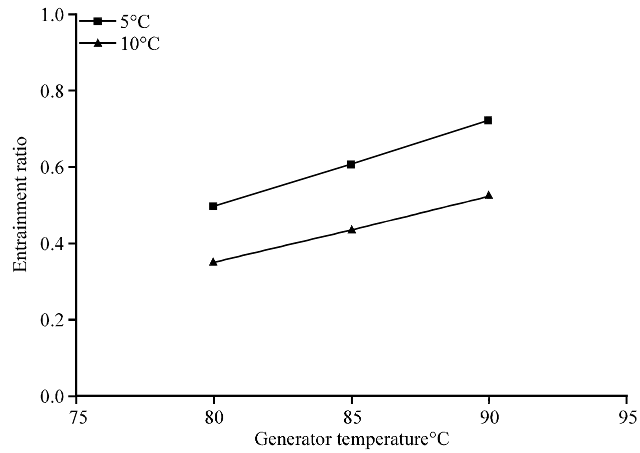

Fig. 8: Effect of generator temperature on entrainment ratio at different evaporator temperatures, Refrigerator capacity: 1 kW, Condenser temperature: 30°C, Dryness coefficient: 0.6

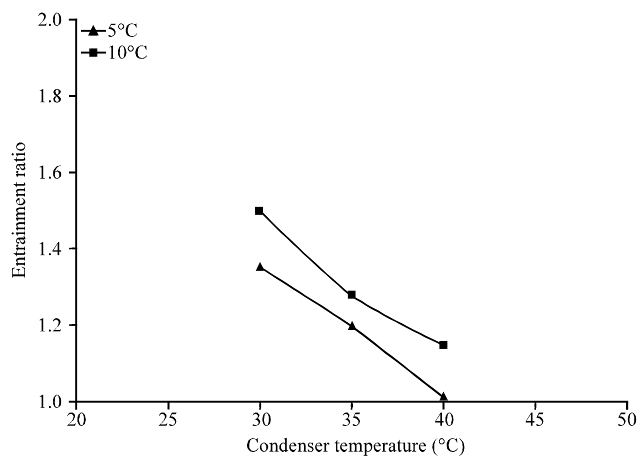

Fig. 9: Effect of condenser temperature on entrainment ratio at different evaporator temperatures, Refrigerator capacity: 1 kW, Generator temperature: 80°C, Dryness coefficient: 0.6

Figure 9 depicts the effect of condenser temperature on ejector entrainment ratio for different evaporator temperatures. As the condenser temperature increases, entrainment ratio decreases due to increase in mass flow of hot water circulated (M_B) to produce constant refrigerating capacity.

CONCLUSION

A simple thermal jet refrigerator is constituted into a heat pipe system and a thermodynamic analysis is carried out for the performance prediction and design of such system. The effects of generator, evaporator and condenser temperature on the Coefficient of Performance (COP) of the system and entrainment ratio of the ejector are studied. It is found that COP and ejector entrainment ratio are higher at high generator and evaporator temperatures whereas they are lower at high condenser temperatures. The predicted performance shows the feasibility of such system and further experimentation and development have to be carried out.

NOMENCLATURE

COP	=	Coefficient of performance
h	=	Specific enthalpy (kJ kg^{-1})
s	=	Specific entropy, (kJ kg^{-1} K)
T	=	Temperature (K)
M	=	Mass flow rate (kg sec^{-1})
X	=	Dryness fraction
ζ	=	Dryness coefficient
μ	=	Entrainment ratio
THP	=	Thermal heat pump
TRF	=	Thermal refrigerator
'	=	Means value of saturated liquid
μ''	=	Means value of saturated vapor

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