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COMPARATIVE ANALYSIS OF VAPOUR JET REFRIGERATION SYSTEM WITH WORKING FLUID R134A & R410A

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ABSTRACT

This paper discuss, the performance of vapor jet refrigeration system with working fluid R410a and R134a.The analysis based on one dimensional mathematical model. A performance comparison is made on various operating condition and ejector area ratio. These operating conditions are boiling temperature (333 to 353K), Condenser temperature (300 to 313K) & evaporator temperature (268 to 278K) and area ratio 4.0 & 5.67.The result shows that performance of jet refrigeration mainly depends on boiling temperature and ejector geometry, the performance R134a is better than R410a.

Keywords: Ejector, simulation program, ejector geometry, performance

1) INTRODUCTION

At present, most of conventional refrigeration system based on vapours compression cycle which are driven by the high grade electrical energy. The most attractive feature of Vapour jet refrigeration system over the conventional refrigeration system that it can driven by the low grade thermal energy such as solar energy, waste industrial heat and geothermal energy. Compared to other renewable energy operated refrigeration system, ejector refrigeration system has more simplicity, reliable, long life, low initial and running cost. The main disadvantage of Vapour jet refrigeration system has lower COP compare to other refrigeration cycle such as absorption refrigeration cycle (1). The performance of ejector refrigeration system mainly depends on the thermodynamic property of working fluid. At initial stage ejector refrigeration cycles utilized the water as working fluid but other refrigerant may also provided the better performance for Vapour jet refrigeration cycle. In this study eco friendly refrigeration's are taken for different area ratio.

2) SYSTEM DESCRIPPTION

The Vapour jet refrigeration cycle is shown fig (1) which consists of boiler, condenser, evaporator, ejector, pump, expansion valve. Low pressure primary fluid enters into boiler where it absorbs low grade heat and converts into high pressure vapour. This high pressure saturated vapour moved into ejector where it expanded in the convergent- divergent nozzle and convert into high velocity stream, the pressure difference produce by primary fluid cause the suction of secondary saturated vaporized fluid from evaporator. Primary and secondary fluid mixed into the mixing chambered section and pressure increased due to shock waves by flow through the diffuser. High pressure fluid condensed into condenser by rejecting heat. A part of fluid pumped into boiler by the pump, remaining fluid entered into evaporated while passed though the expansion valve and produced the refrigeration effect.

Nomenclature	
a	cross sectional area (m^2)
Cop	coefficient of performance
d	diameter (m)
f	friction factor
H	specific enthalpy (KJ/kg)
L	length (m)
m	mass flow rate (Kg/s)
P	pressure (bar)
Q	heat transfer (KJ/kg)
T	temperature (K)
V	velocity (m/s)
Ø	area ratio
ή	efficiency
µ	entrainment ratio
	density
υ	specific volume
1,2,3….	states in respected cross section

subscripts	
as	after shock
b	boiler
bs	before shock
c	condenser
d	diffuser
e	evaporator
is	isotropic
m	mixing chamber
p	primary nozzle
s	secondary nozzle

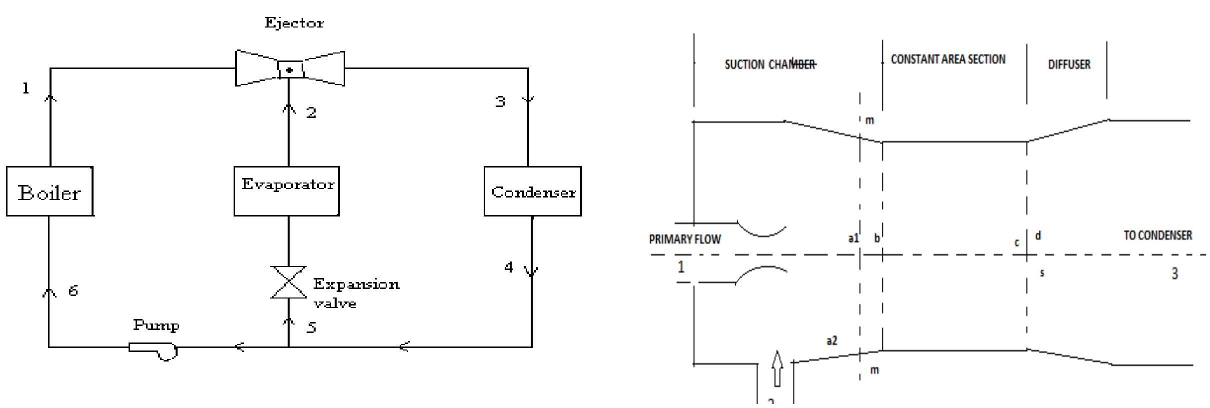

Fig 1 schematic diagram of Vapour jet refrigeration system

3) ANALYSIS OF EJECTOR REFRIGERATION SYSTEM

In present study a mathematical modal developed for single phase Vapour jet refrigeration. The model governing equation is obtained by applying mass momentum equation and energy balance in control volume (2,3,4,12).

1.1 Primary fluid flow

Mass flow rate of primary fluid

$$m_p = \frac{Ap * \sqrt{2 * \acute{\eta} p * (H1 - Hp)is}}{vp} \qquad (1)$$

Velocity primary fluid at exit of nozzle

$$V_p = \sqrt{2 * \acute{\eta} p * (H1 - Hp)is} \qquad (2)$$

3.2secondary fluid flow

Mass flow rate of secondary fluid

$$m_s = \frac{As * \sqrt{2 * \acute{\eta} s * (H2 * - Hs)}}{vs} \qquad (3)$$

Velocity secondary fluid flow at exit of muzzle

$$V_s = \sqrt{2 * \acute{\eta} s * (H2 - Hs)is} \qquad (4)$$

3.3 mixed fluid flow

Mixed mass flow rate

$$m_m = m_p + m_s \qquad (5)$$

Entrainment ratio

$$\mu = \frac{ms}{mp} \qquad (6)$$

Appling momentum balance on the fluid entering mixing chamber

$$V_{bs} = \frac{Vp + \mu * Vs}{(\mu + 1) * (1 + \frac{fm\, Lm}{2\ dm})} + \frac{(Ps - Pm) * Am}{\left(1 + \frac{fm\ Lm}{2\ dm}\right) * Mm} \qquad (7)$$

Appling energy balance on the fluid entering mixing chamber

$$H_{bs} = \frac{Hp + \frac{vp^2}{2} + \mu * (Hs + \frac{Vs^2}{2})}{1 + \mu} - \frac{Vm^2}{2} \qquad (8)$$

Mass flow rate after mixing

$$m_{mm} = \frac{Vm * Am}{vm} \qquad (9)$$

Applying mass momentum, energy and entropy balance across the normal shock

$$_{as} * V_{bs} = {}_{as} * V_{as} \qquad (10)$$

$$P_{as} - P_{bs} = {}_{bs} * V_{bs}^2 - as * V_{as}^2 \qquad (11)$$

$$H_{bs} + \frac{Vbs^2}{2} = H_{as} + \frac{Vas^2}{2} \qquad (12)$$

Governing equation at the exit of diffuser

$$d = \frac{ms + mp}{Ad * Vas} \quad (13)$$

$$Hd - Has = ή_d*(V_{as}^2 - V_d^2) \quad (14)$$

Neglecting pump work, the cop can be evaluated

$$COP = \frac{Qe}{Qb} \quad (15)$$

$$COP = \mu\left(\frac{H_2 - Hc}{H_1 - Hc}\right) \quad (16)$$

4) RESULT

Comparative calculation have been made on R134a and R410a in operating condition Boiler temperature T_b=333K and 353K, condenser temperature T_c= 300 to 320K and evaporator temperature 273 to 283K for area ratio $Ø = 4.0$ and 5.64.

In Fig 4.a & 4.b boiling temperature are varied from 335 to 360K at the constant evaporator temperature 273 K and condenser temperature 300K for area ratio 4.0 & 5.67 respectively. Fig 4 showed that COP is increased by increasing boiling temperature. By the Fig 4.a & 4.b, COP is higher for area ratio 5.64 compared to area ratio 4.0 for same operating condition. The performance of R134a are better compared R410a for both area ratio.

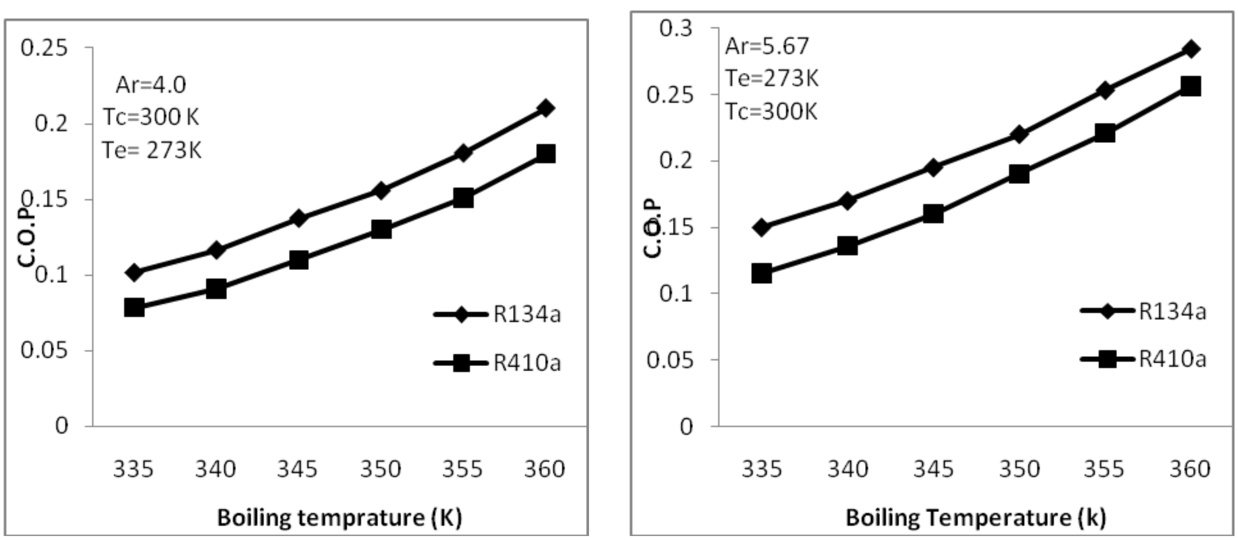

Fig 4.a & 4.b Effect Boiling temperature

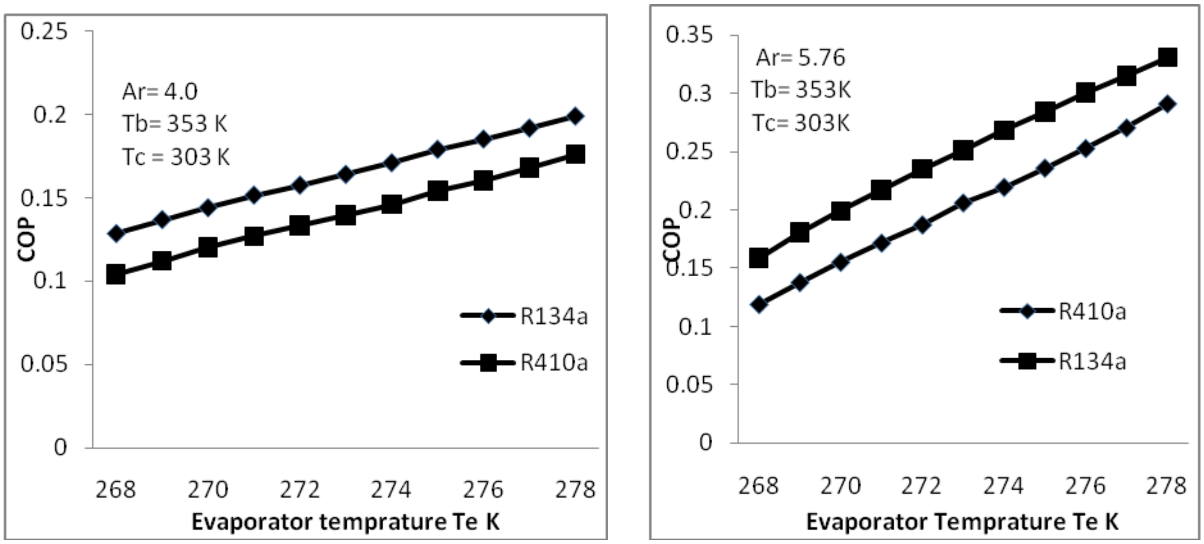

Fig 5 Effect of evaporator temperature

In fig Fig 5 (a-b) shows the effect of evaporator temperature in COP of system at the fixed boiler temperature T_b=353K and condenser temperature 303K for area ratio Ø=4.0 R 5.64. COP is increased while increased the evaporator temperature because increased evaporator temperature also increased mass flow rate and entrainment ratio. COP of system decreased by increasing the condenser temperature as shown fig 5.a and 5.b can be explained by that increased of condenser temperature decreased the driving ratio (ratio of boiler pressure and condenser pressure) hence cause low entrainment ratio and COP.

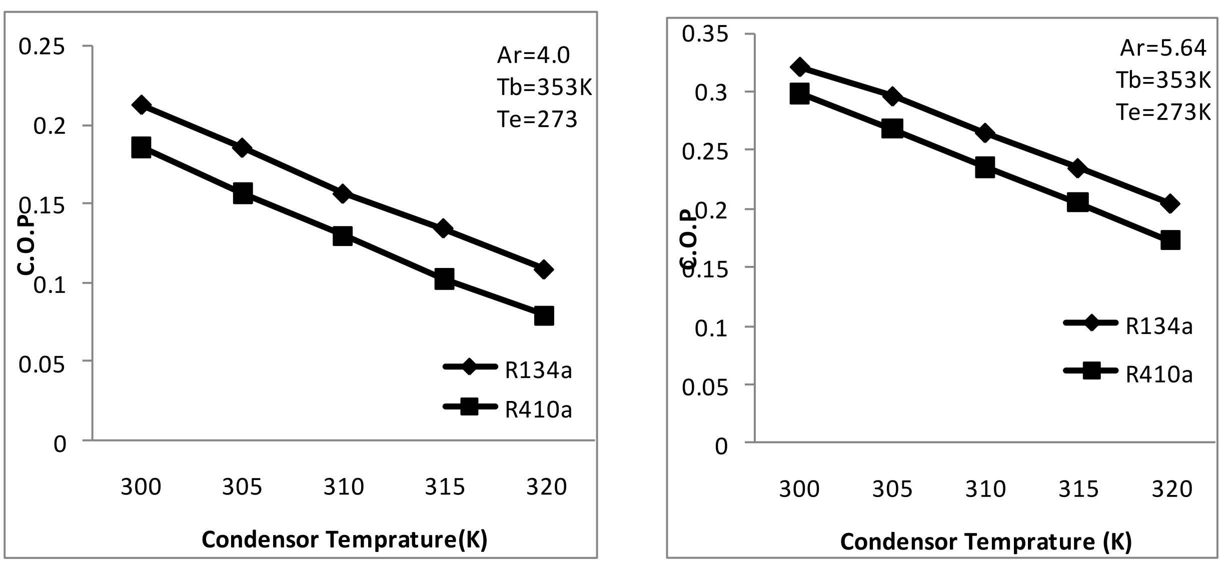

Fig 6 Effect of Condenser temperature

Fig 6 (a)-(b)shows the effect of Condenser temperature on the COP for refrigerants R410a and R134a at the boiling temperature 353K and evaporator temperature 273K for ejector ratio Ø=4.0 and 5.67. The COP decreases at increase the Condenser temperature.

5) CONCLUSION

Comparative analysis were made on Jet refrigeration cycles with working fluid R410a and R134a in same ejector geometry and same operating condition using one dimension modal. On based of study COP of ejector refrigeration cycle depends on ejector geometry, operation condition and property of working fluid. COP of system increased as boiler temperature increased while Cop is decreased when Compression ratio and condenser temperature increased. For different ejector ratio the performance of system are different but ejector ratio Ø= 564 at 353K have higher COP comparative other ejector ratio. For all area ratio and operating Condition the performance R134a is better than R410a.

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