Theoretical and experimental research on solar-driven single stage ejector refrigeration system

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Abstract

This paper presents effect of operating conditions and ejector geometry on the single stage ejector refrigeration using water as refrigerant, through the developed computer simulation program. In addition, utilization of the program is to design and fabricate solar driven 2 Tons single stage refrigeration system which is operated at generator and evaporator temperature of 110ºC and 10ºC, respectively. Parameters of operating conditions and ejector geometry, which are investigated, are generator and evaporator temperature and pressure and nozzle throat, inlet diameter of mixing chamber, nozzle exit plane and ejector throat. From testing results of solar driven single stage ejector refrigeration system, it is found that monthly average solar fraction, average COP and chilled water temperature is 0.60, 0.59 and 21ºC, respectively.

Keywords: *Refrigeration system, single stage ejector, refrigerant, system performance.*

1. Introduction

Steam ejector refrigeration system driven by solar thermal is a significant attempt to replace the vapor compression refrigeration system (VCC) which provides high electrical energy consumption and use of CFC refrigerant that has a negative impact on environment. In a steam ejector refrigeration system, water is used as motive fluid for an ejector and the system is easy to install and maintain. However, an important disadvantage of this system is low performance (COP~0.4) when compared to VCC (COP~2-4) and absorption cooling system (COP~0.7) [1]. Ejector uses a single pure substance as refrigerant while absorption system uses binary salt solution being able to have problem of crystallization. The structure of ejector system has more resemblance with VCC than absorption system and is less complex as well. Thus, an ejector refrigeration system can be attractive to improve system performance and can be developed to compete in cooling market.

Most previous researches related to ejector refrigeration system emphasized on the theoretical study on effect of operating conditions and ejector geometry on system performance. The theoretical simulation program using Computational Fluid Dynamic or CFD can be an effective tool to simulate behavior of fluid flowing through ejector and COP is simulated under different operating conditions and ejector dimensions [2]. Factors of operating conditions affecting COP of steam ejector refrigeration system are temperature & pressure of boiler, evaporator and condenser. Practical results about ejector refrigeration system have been provided insignificantly [3, 4].

Present study designs single stage ejector refrigeration system using water (R718b) as refrigerant through the developed computer simulation program. Designed parameters which are obtained from program for fabricating the system are such as condenser pressure and temperature, optimal mass flow rate of entrained fluid, load required at condenser, boiler and evaporator, ejector performance and COP. Experimental testing of system has been performed to investigate system operation and performance. This system is similar to the several previous research works [5-7] using solar thermal to be main source to drive ejector refrigeration system. This research aims to investigate effect of operating conditions and ejector geometry on system performance using water as working fluid.

Nomenclature.

A	area (m^2)
COP	coefficient of performance
D	diameter (m)
L_A	length of mixing chamber (m)
P	pressure (kPa)
T	temperature (°C)
X_A	length of Mixing (%)
M	mach number
Q	heat transfer rate (kW)
W	work rate (kJ/kg)
e	evaporator
h	enthalpy (kJ/kg)
m	mass flow rate (kg/s)
o	outlet
py	primary working fluid
sy	secondary working fluid
u	internal energy (kJ/kg)
ω	entrainment ratio
subscripts	
1A	throat of nozzle position
3, a, b,y1,y2	ejector location
d	diffuser

2. Performance of ejector refrigeration system

The performance of ejector refrigeration system can be expressed in terms of coefficient of performance COP_{ejc} which is defined as the heat flow rate of evaporator $\dot{Q}_{ev}$ and generator's $\dot{Q}_G$ following equation

$$COP_{ejc} = \frac{\dot{Q}_{ev}}{\dot{Q}_G}$$

Terms $\dot{Q}_{ev}$ and $\dot{Q}_G$ can be expressed in form of enthalpy difference and system performance as follows

$$COP_{ejc} = \omega \frac{h_e - h_c}{h_b - h_c}$$

Ejector performance is measured by the entrainment ratio, which is defined as mass flow ratio of secondary flow to primary flow

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

Overall system performance is expressed in term of solar fraction which is defined as fraction of thermal energy delivered to the chiller, provided by solar collector. Solar Fraction, SF can be written as

Solar Fraction = Solar Energy Used in the System/Thermal Energy delivered to chiller

3. Design of ejector refrigeration system by developing simulation program

Computer simulation program of single stage ejector refrigeration system has been developed to obtain parameters which are used to design and fabricate the refrigeration system. Two significances of ejector geometry and operating conditions are varied in the program to study their effect on ejector performance and COP. The flow chart of simulation has been shown in figure 1.

The computer simulation program for this study is written using visual basic program for system design. The varying operating conditions and ejector dimensions in the simulation program can provide the appropriate results to be used for fabricating the system having higher performance.

Assumptions applied for the program have been shown as follows.

1) Fluid flow is compressible.

2) Ejector is analyzed with 1-D model.

3) Density and specific heat capacity of fluid flow through all sections of ejector are constant.

4) Mixing process between motive and entrained fluid occurs at mixing chamber

5) Steady state is applied for the process.

6) Efficiency of nozzle (η_n) is 0.85

7) Efficiency of suction (η_s) is 0.85

8) Efficiency of diffuser (η_d) is 0.85

9) Temperature of condenser water T_{wC-I} and Evaporator (T_{wE-i}) are 30 °C

10) Specific heat of water at 20 °C (Cp_w) is 4.18 kJ/kg-°C

11) Mass flow rate of water ($\dot{m}_w$) is 1 kg/s

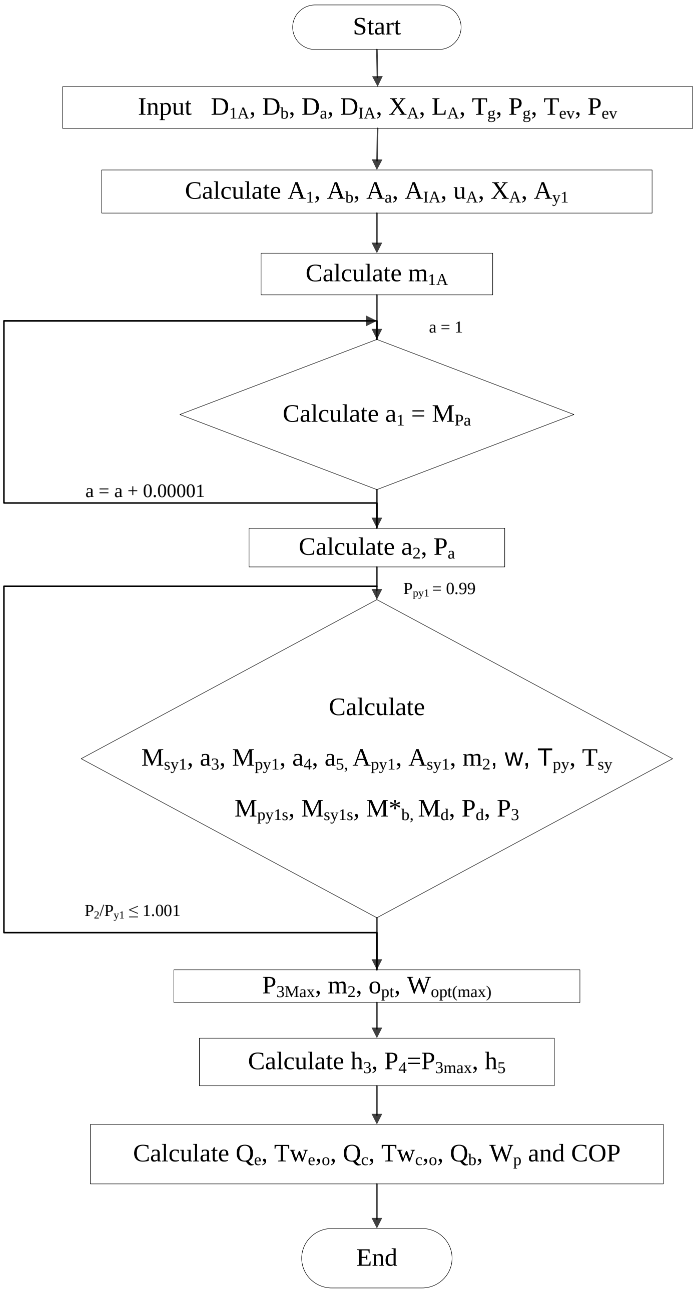


Figure 1 Flow chart of the simulation.

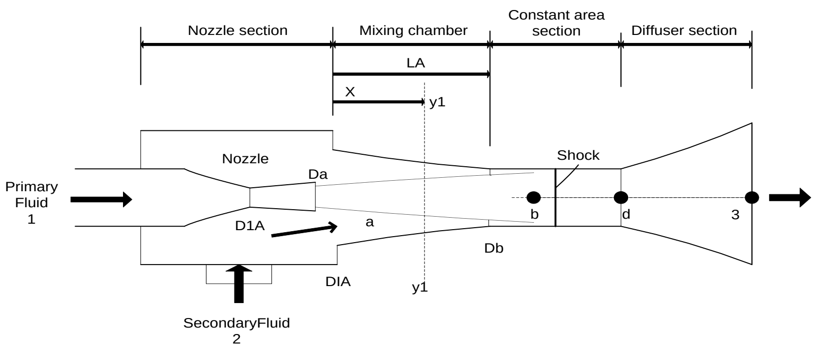


Figure 2 Flow of working fluid through various positions of ejector

Input parameters of simulation model are as follows;

1) Boiler (Generator) temperature (T_g)
2) Evaporator temperature (P_{ev})
3) Boiler (Generator) pressure (T_g)
4) Evaporator pressure (P_{ev})
5) Throat of nozzle (D_{1A})
6) Exit diameter of nozzle (D_a)
7) Inlet diameter of throat of ejector (D_b)
8) Diameter inlet of mixing chamber (D_{IA})
9) Length of mixing chamber (L_A)
10) Mixing fluid distance (X_A)

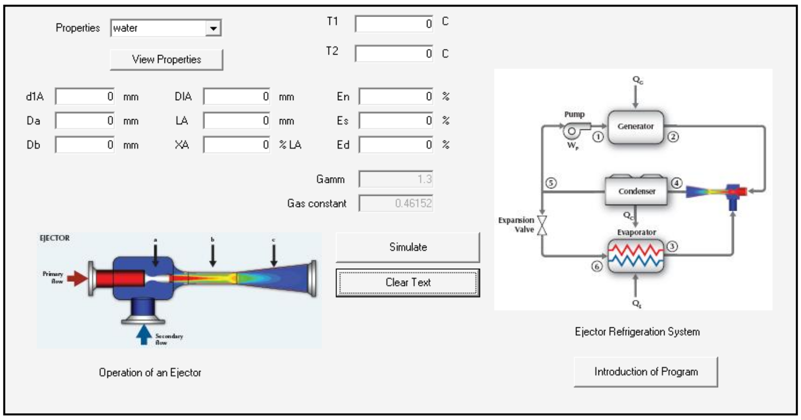


Figure 3 Input Form of simulation program.

Figure 3 shows the interface for entering input data of operating conditions and ejector geometry to the simulation program. The operating condition is temperature of generator and evaporator, the geometry is dimension of nozzle and ejector. The results from the simulation program are compared with another experiment for validation of simulation program.

Table 1 Deviation between simulation results with experiment data of Eames et al. [8]

Generator Temp (°C)	Evap Temp (°C)	Present work		Eames et al.		Deviation (%)	
		Condenser Pressure (kPa)	COP	Condenser Pressure (kPa)	COP	Condenser Pressure (kPa)	COP
120	5	3.212	0.3932	3.4	0.4044	-0.55	-2.76
125	5	3.617	0.3335	3.7	0.3442	-0.22	-3.08
130	5	4.069	0.2839	4.4	0.2756	-0.75	3.01
135	5	4.569	0.2423	5.1	0.2513	-1.04	-3.55
140	5	5.120	0.2074	5.4	0.1779	-0.52	16.59
120	7.5	3.375	0.4780	3.6	0.5004	-0.62	-4.47
125	7.5	3.786	0.4060	4.1	0.4189	-0.77	-3.06
130	7.5	4.246	0.3462	4.6	0.3553	-0.77	-2.56
135	7.5	4.758	0.2961	5.1	0.2965	-0.67	-0.12
140	7.5	5.326	0.2539	5.7	0.2334	-0.66	8.82
120	10	3.533	0.5626	3.8	0.5862	-0.70	-4.03
125	10	3.947	0.4784	4.2	0.5374	-0.60	-10.97
130	10	4.414	0.4084	4.7	0.4734	-0.61	-13.73
135	10	4.934	0.3498	5.3	0.3892	-0.69	-10.11
140	10	5.512	0.3005	6	0.3093	-0.81	-2.82

The results of simulation program at varying generator temperature ranging from 120 to 140 °C and an evaporator temperature ranging from 5 to 10 °C can be seen in Table1. The performance and condenser pressure values of single stage ejector refrigeration system are shown and compared with the results of Eames et.al experiment [8]. The deviation of condenser pressure and COP yields low average values of -0.67% and -2.19% respectively. Thus, the program can be used to design experimental single stage ejector refrigeration system for this research.

Experimental single stage ejector refrigeration system has been designed under conditions of generator temperature at 110 °C and evaporator temperature at 10 °C and ejector dimensions as presented in Table 2.

Table 2 Dimensions of an ejector to be used for system design

Throat of nozzle (D_{1A})	2	mm
Exit diameter of nozzle (D_a)	8	mm
Inlet diameter of throat of ejector (D_b)	18	mm
Diameter inlet of mixing chamber (D_{IA})	24	mm
Length of mixing chamber (L_A)	140	mm
Mixing fluid distance (X_A)	80	%

4. Effect of operating conditions on system performance, COP

4.1 Generator temperature

Effect of varying generator temperature on COP is shown in Figure 4. The result shows that the increase of generator temperature ranging from 90 to 140 ° C decreases the COP and the same result can be seen in figure 5 for entrainment ratio. Figure 6 shows that, when generator temperature increases, mass flow rate of primary fluid increases. This results in decreasing mass flow rate of secondary fluid which causes entrainment ratio to decrease as well. From varying generator temperature at fixed dimensions of an ejector as per Table 2, it is found that high COP can be obtained in generator temperature range of 105 -110 °C. These results are similar to research works of Sun et al. [9] and Ma et al.[10].

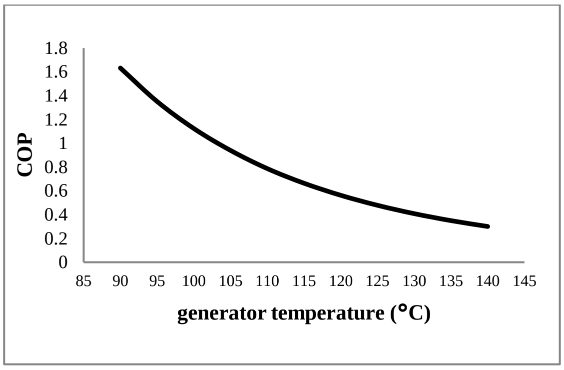


Figure 4 Variation of generator temperature on COP

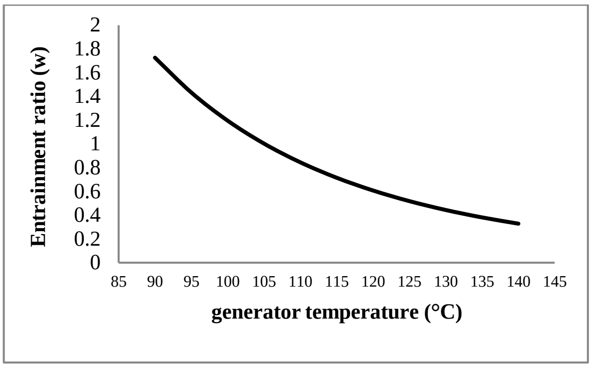


Figure 5 Variation of generator temperature on entrainment ratio

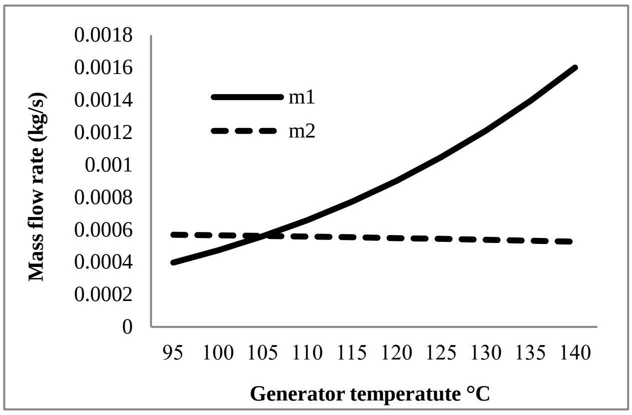


Figure 6 Variation of generator temperature on mass flow rate of primary and secondary fluid

4.2 Evaporator temperature

Figure 7 and 8 show the effect of variation of the evaporator temperature (5 to 20°C) on entrainment ratio and COP, respectively. The results show that entrainment ratio and COP increases with rise in evaporator temperature. Evaporator temperature at 5, 10, 15 and 20 °C, yields increasing COP of 0.55, 0.78, 1.10 and 1.52 respectively. These results are similar to research work of Pollerberg et al.[11] in which the increase of evaporator temperature ranging from 7 ºC to 17 ºC leads to increase COP value ranging from 53 to 132% respectively.

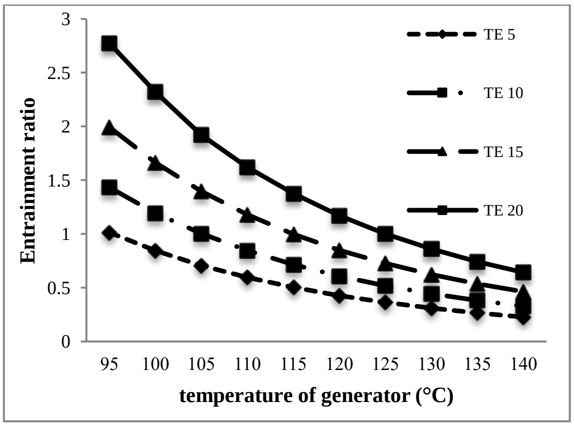


Figure 7 Variation of evaporator temperature on Entrainment ratio

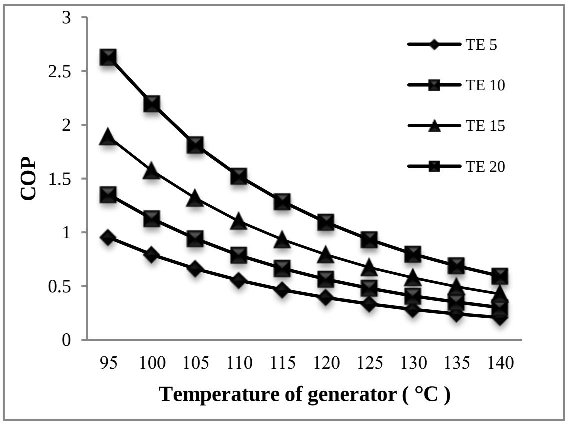


Figure 8 Variation of evaporator temperature on COP

5. Effect of ejector geometry on mass flow rate of primary and secondary fluid and COP

5.1 Nozzle throat

In this study, effect of varying nozzle dimension at constant generator temperature at 110 °C and evaporator temperature at 10°C on mass flow rate of primary and secondary fluid and COP is studied. As shown in Figure 9, as nozzle throat diameter increases, mass flow rate of primary fluid increases while mass flow rate of secondary fluid decreases slightly. This results in decrease in COP as shown in figure 10. The results show that nozzle throat diameter is increasingly varied ranging from 1.00 to 3.00 mm resulting in the decrease of COP ranging from 0.32 to 0.22. The results yield the same as Ruangtrakoon et al.[3] that the larger nozzle throat provides higher primary fluid mass flow rate than smaller one, therefore lower secondary fluid mass flow rate is entrained leading to a lower entrainment ratio.

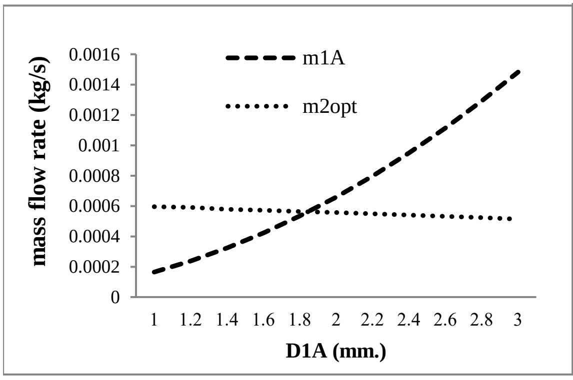


Figure 9 Variation of diameter of nozzle throat on mass flow rate of primary and secondary fluid

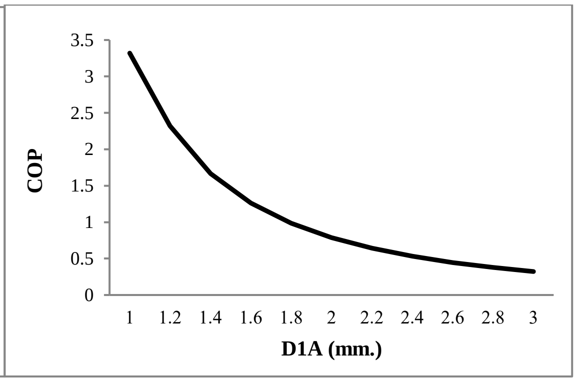


Figure 10 Variation of diameter of nozzle on system performance

5.2 Nozzle exit plane

Variation effects of diameter of nozzle exit plane ranging from 5 to 15 mm on mass flow rate of primary and secondary fluid and COP is investigated. From Figure 11 it can be seen that, variation in diameter of nozzle exit plane slightly affect mass flow rate of secondary flow and does not affect mass flow rate of primary fluid thus, resulting in slight decrease in COP as shown in figure 12. These results are similar with Ariafar and Toorani [12]. In addition, Ruangtrakoon et al. [3] explained that the increase of nozzle plane will decrease of mass flow rate of secondary fluid.

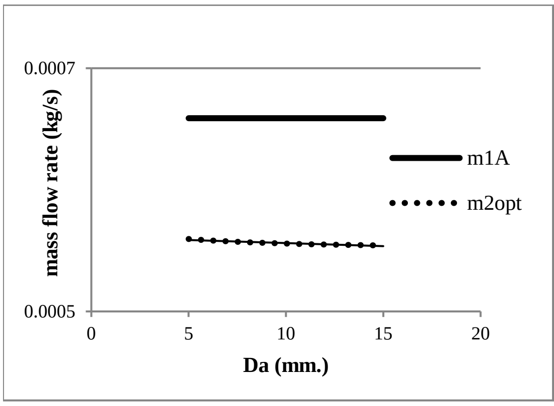


Figure 11 Variation of diameter of nozzle exit plane on mass flow rate of primary and Secondary fluid

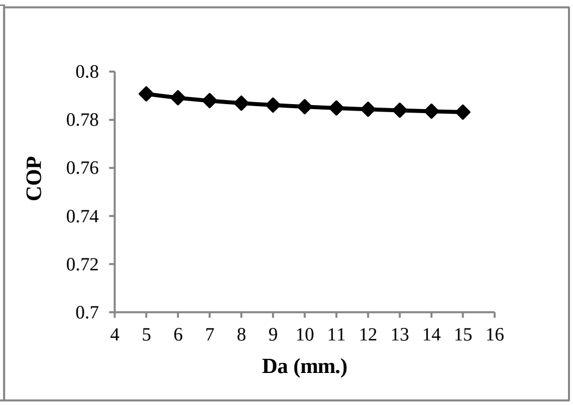
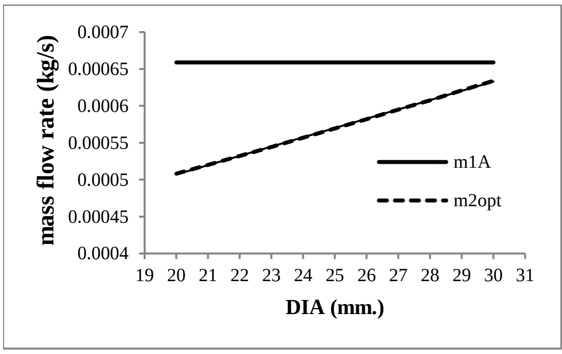


Figure 12 Variation of diameter of nozzle exit plane on system performance

5.3 Inlet diameter of mixing chamber

Varying inlet diameter of mixing chamber (D_{IA}) ranging from 20 mm. to 30 mm increases the mass flow rate of secondary fluid ranging from 0.00050 to 0.00063 kg/s. This results in the increased system performance, COP, ranging from 0.71 to 0.89 because of the increase in mass flow rate of entrained fluid, as shown in Figure 13 and 14, respectively.



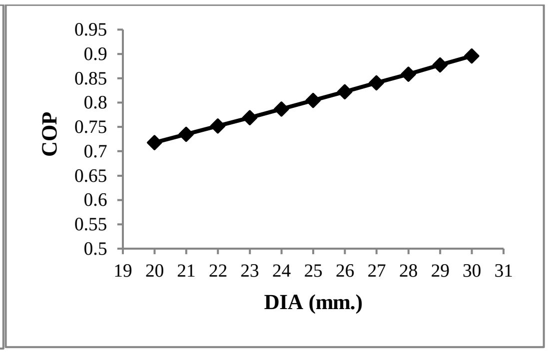
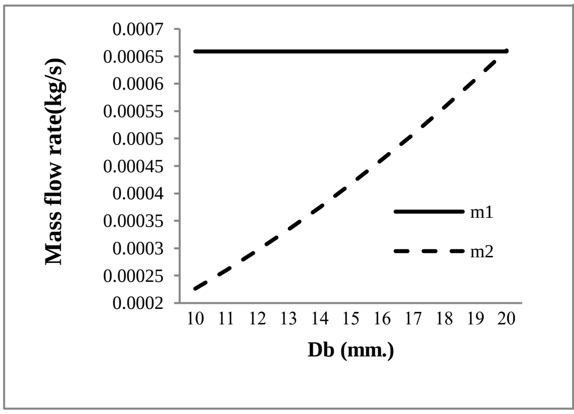


Figure 13 Variation inlet diameter of mixing Chamber dimension on mass flow rate

Figure14 Variation in Diameter inlet of mixing Chamber dimension on performance

5.4 Ejector throat

Effect of diameter of ejector throat (D_b), which is varied ranging from 10 mm. to 20 mm, on mass flow rate of primary and secondary fluid and COP is investigated. The increase in diameter of ejector throat as referred above has influence on significantly increasing mass flow rate of entrained fluid, ranging from 0.00022 to 0.00066 kg/s, which lead to increase system performance, COP, ranging from 0.36 to 0.93 as shown in figure 15 and 16, respectively. The obtained results are similar to Yen et al. [13] in which the increase of ejector throat affects the increase of entrainment ratio and COP.



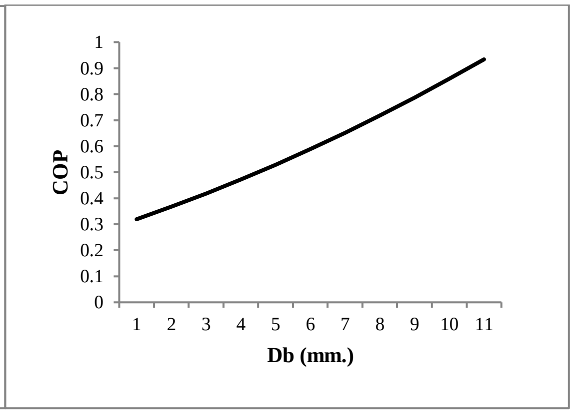


Figure 15 Variation of Diameter of throat ejector on mass flow rate

Figure 16 Variation of Diameter of throat ejector on performance

6. Experimental investigation of solar driven single stage ejector refrigeration system

A full scale solar-driven 2 tons single stage ejector refrigeration experimental system, as shown in Figure 17, has been fabricated as per design data obtained from simulation program. Ejector geometry is shown in table 1 and the system is operated at 110 ºC of boiler temperature and 10 °C of evaporator temperature. The fabricated system is located at Energy Park Area, School of Renewable Energy Technology (SERT), Naresuan University in Thailand and is tested from 09.00 am to 16.00 pm. A schematic diagram of overall system can be shown in Figure 18.

Figure 17 A full scale solar driven 2 tons single stage ejector refrigeration experimental system

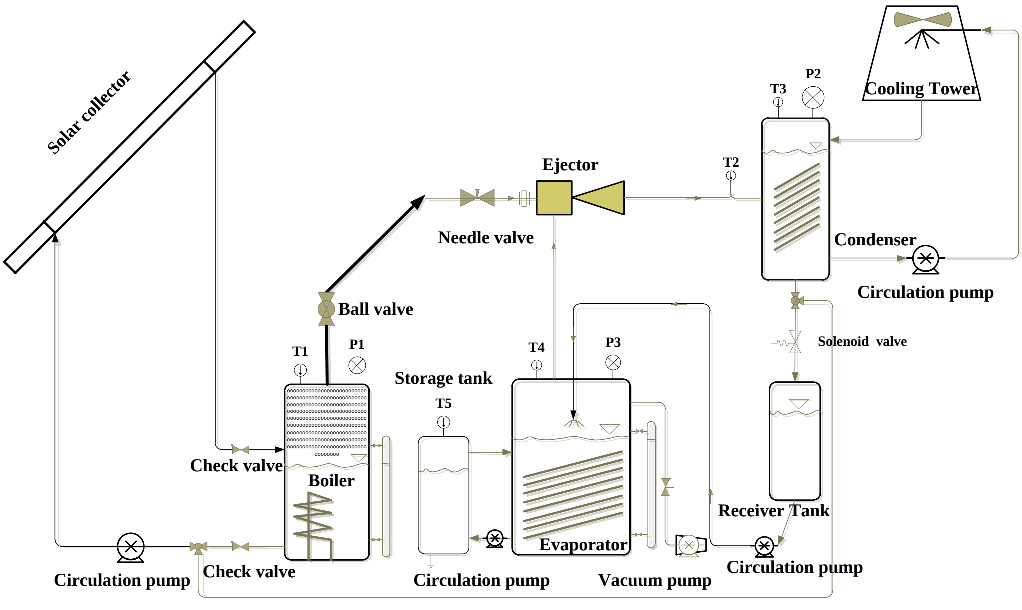


Figure 18 Schematic diagram of solar driven 2 Tons single stage ejector refrigeration system

Thermal energy delivered to heat generator is mainly obtained from 3.5 m^2 heat-pipe evacuated tube collector and 3 kW of auxiliary heater to produce saturated steam (at110°C) which is the primary fluid in this case. Primary fluid enters into the nozzle of ejector when needle valve opens. At nozzle's exit, primary fluid has low pressure resulting in boiling of secondary or entrained fluid at low evaporator pressure where this fluid is sucked to mix with primary steam in mixing chamber.

Mixed fluid flows through throat of ejector, diffuser and vacuum condenser, respectively. Heat at condenser is released using cooling tower and saturated water leaves from condenser while flow is separated into 2 parts. In the first part, liquid water flows to solar collector to receive heat and heat generator, respectively and in the 2nd part, liquid flows through expansion valve reducing the pressure to evaporator pressure level. Water from chilled water tank is circulated by pump to evaporator tank and heat from this water is applied to secondary fluid for vaporization. Chilled water is produced and flows out from evaporator tank to chilled water tank. Mass flow rate of primary and secondary fluid is measured by the change of water level attached with heat generator and evaporator. Additionally, temperature and pressure at various positions are measured by thermocouples and pressure gauges, respectively. All testing data are collected by data logger.

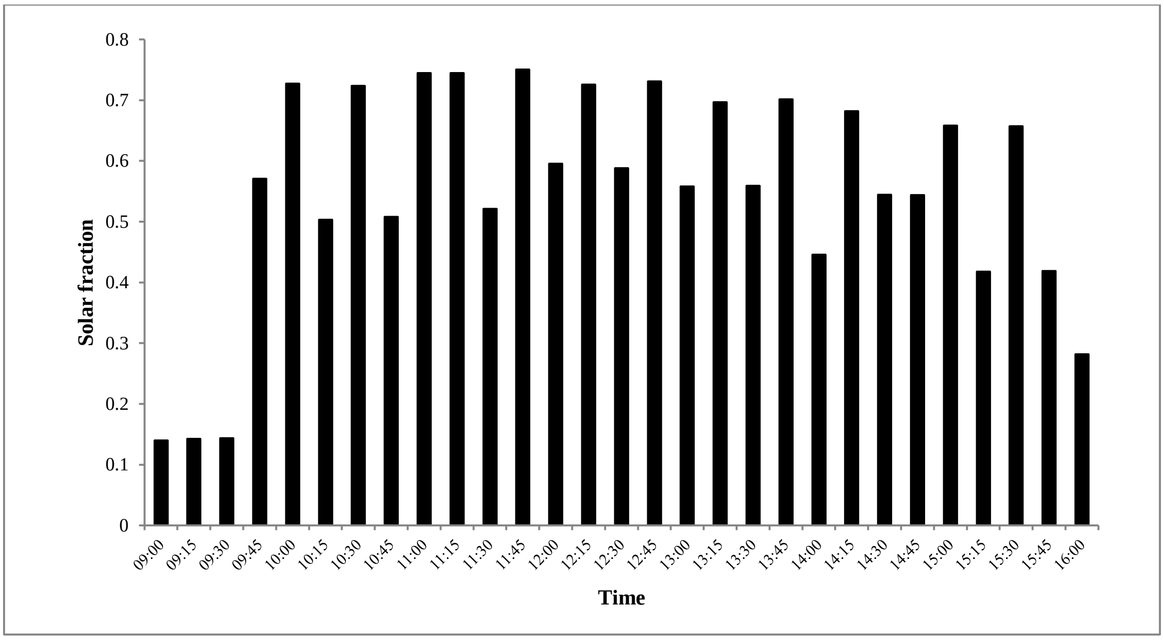


Figure 19 Relationship between solar fraction and time of solar steam ejector refrigeration system

Figure 19 shows overall system efficiency of solar steam ejector refrigeration system defined as solar fraction (SF) which is plotted with time on October 17th, 2014. Data is collected for 1 month of October and monthly average SF is 0.60.

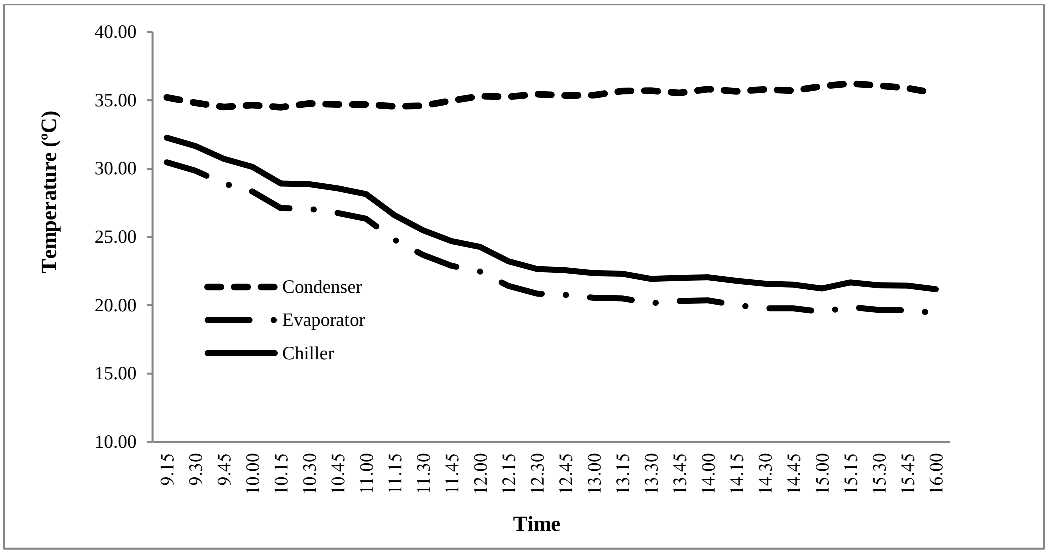


Figure 20 Temperature of condenser, evaporator and chilled water with time on October 17th, 2014

Figure 20 shows the relationship among temperature of condenser evaporator and chilled water with time on October 17th, 2014. From Figure 20, it can be seen that evaporator and chilled water temperature curve give the same trend. Evaporator and chilled water temperatures are lowest at 18 °C and 21°C, respectively. Allouche et al [14] studied the change of evaporator temperature by predicting from CFD program and the results are similar to this research.

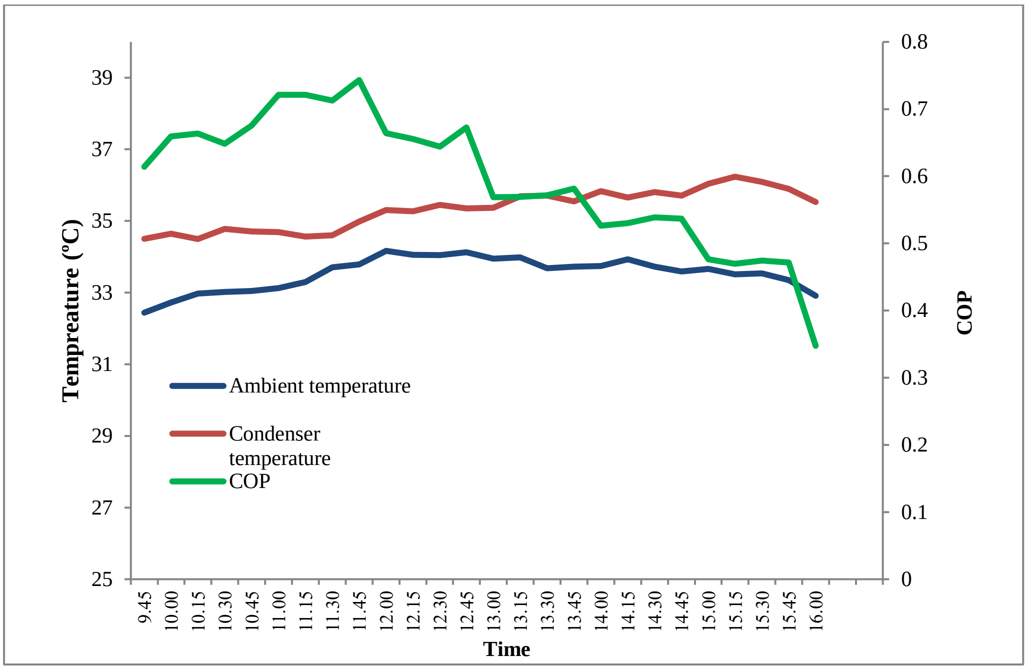


Figure 21 Ambient and condenser temperature and COP at any time

Figure 21 shows relationship of the COP, condenser and ambient temperature plotted with time. The average COP obtained is 0.59 and the refrigeration system performance yields the highest COP of 0.74 at 11.45 am and COP reduces to be 0.34 at 16.00 pm. From Figure 20, the change of COP can be explained as the change in evaporator temperature. COP is higher in the first period at higher evaporator temperature and COP is decreased atlower evaporator temperature at longer time. This results is same as Pollerberg C et al. and Yen et al. [5,13]. Furthermore, the results showed that condenser temperature significantly affects COP as well. That is, as condenser temperature decreases, COP increases which is similar to Pollerberg et al. [15] and Petrenko et al.[16] results.

7. Conclusion

Solar driven 2 Tons single stage ejector refrigeration system has been designed, fabricated and experimentally tested. Computer simulation program of steam ejector refrigeration system has been developed to study effect of operating conditions and ejector geometry on ejector performance and COP. Parameters of operating conditions and ejector geometry which are studied are generator and evaporator temperature and pressure and nozzle throat, inlet diameter of mixing chamber, nozzle exit plane and ejector throat. From testing results of solar driven single stage ejector refrigeration system, it is found that monthly average solar fraction, average COP and chilled water temperature is 0.60, 0.59 and 21°C, respectively.

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