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ABSTRACT

Steam ejectors are capable of drawing large volumes of vapor within a relatively small space and at a low cost. In this study, the compressor is replaced by a constant-area mixing ejector to reduce the energy consumption in refrigeration cycle. The influence of various parameters on the performance of the system is obtained by an iterative program and reasons are analyzed in this paper. The effect of pressure difference, the difference of evaporation pressure and primary nozzle outlet pressure, on the COP and the exergy loss of every component in system is considered. Finally the key points to optimize the ejector cycle and the minimum exergy loss location to optimize the ejector design are obtained by theoretical research. A better understanding for the real industrial application is provided by this theoretical analysis on the steam ejector refrigeration system and a foundation for the simulation and experimental reach is laid.

1. INTRODUCTION

Steam jet refrigeration system can operate with low-grade thermal energy such as the industrial waste heat, solar energy, steam exhaust or other low-grade energy heat, which makes it environment-friendly. Compared with other type of systems, Steam jet refrigeration system has many advantages such as structural simplicity, high reliability, easy to maintain, low cost and can be used with water which is the most harmless refrigerant. The development of the first steam ejector refrigeration cycle proposed by Maurice Leblanc (Chunnanond and Aphornratana, 2004a) in 1910s, due to the poor design and processing level at that time, is limited by its low coefficient performance (COP). However, the investigation of ejector refrigeration system has drawn the researcher's attention again in 1970s as it's environment-friendly and economically feasible.

Ma *et al.* (2010) carried out an experimental investigation of a novel 5kW steam jet refrigerator suitable for solar energy applications. The results showed that with the increase in boiler temperature, the coefficient of performance (COP) did not always increased. The maximum coefficient of performance (COP) and cooling capacity was found at a boiler temperature of about 90℃. Therefore, for the given operating parameters, every ejector refrigeration system has an optimum boiler temperature, at which the maximum coefficient of performance (COP) could be obtained. During the experiment, the primary flow was controlled by a spindle. The similar conclusion was stated by Chunnanond and Aphornratana (2004b). The results showed that, the cooling capacity and coefficient of performance (COP) can be raised by the decrease in the boiler pressure or the increase in evaporation pressure. The author also stated that the amount of secondary fluid sucked into by the primary fluid and the momentum of mixed steam dominated the system performance. In recent years, Butterworth and Sheer (2007) used the high-pressure water, which is available from vertical pipelines in deep mine shafts, to drive an ejector refrigeration system and the system performance has been improved. Effect of the area ratio γ_A between primary nozzle and constant area section on the system performance was studied by Oliveira *et al.* (2009) using CFD. The result indicated that an increase in γ_A caused an increase in entrainment ratio and a decrease in the critical back (condenser) pressure, so an optimal value should exist in theory for the given operating conditions. In order to consider the significance of the primary nozzle geometries, Aphornratana *et al.* (2013) carried out CFD analysis of eight different primary nozzles. The results demonstrated that the expansion angle in the primary nozzle outlet of the primary fluid and the position in the mixing chamber of the mixed fluid played an important role in the ejector performance.

At present, experiment and simulation on the steam jet refrigeration system usually be used to improve the system's performance. Many parameters related to the system's performance have been analyzed and improved for the refrigeration cycle with particular configuration. In this paper, an iterative program on a constant-area ejector refrigeration system, in which water was used as the refrigerant, was employed to optimize the design of the ejector.

2. THE EJECTOR REFRIGERATION CYCLE

As the critical component in ejector refrigeration cycle, an ejector is consisted of the primary nozzle, the suction chamber, the mixing chamber and the subsonic diffuser. The high pressure saturated steam produced in the boiler expands and accelerates through the primary nozzle; it draws the secondary fluid from the evaporator into the mixing chamber. The combined fluid assumed to be completely mixed further compressed in subsonic diffuser and then discharged to the condenser.

μ is defined as the entrainment ratio of the ejector:

$$\mu = m_7 / m_1 \tag{1}$$

A schematic view of the steam ejector refrigeration system and a P-h diagram are shown in Fig.1and 2. Normally, the steam ejector refrigeration system includes a boiler, ejector, condenser, expansion valve, evaporator and a fluid pump.

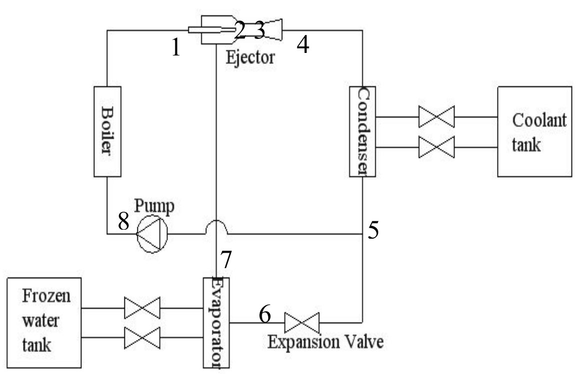

Figure 1: A schematic view of the steam ejector refrigeration system

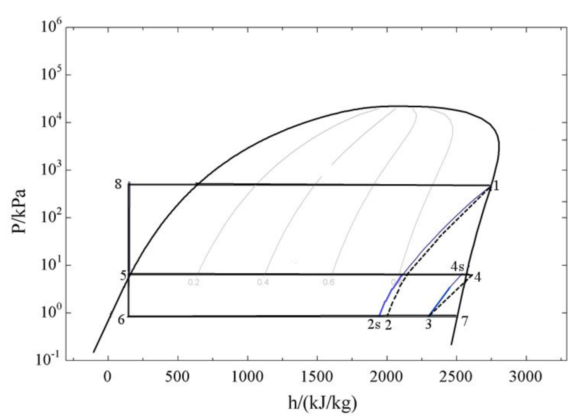

Figure 2: The P-h diagram of the steam ejector refrigeration system

The primary fluid at pressure P_s at state (1) enters the primary nozzle, expands and accelerates isentropically to the evaporation pressure P_0 at state (2s). The real steam expanded process to the evaporation pressure P_0 with a nozzle efficiency η_n =0.7, ends at state (2). The accelerated primary fluid sucks the secondary fluid from evaporator at pressure P_0 corresponding to state (7) into the suction chamber. Combined fluid assumed to be completely mixed in constant-area section at state (3) further compresses to state (4) in subsonic diffuser. The subsonic diffuser is considered to have a diffuser efficiency of η_d =0.8 with the isentropic outlet at state (4s). Then mixed fluid is discharged to the condenser and cooled at pressure P_k to state (5).

The stream leaving the condenser is divided into two flows; one of two flows enters the expansion valve and expands to pressure P_0 at state (6). Another flow is pumped to pressure P_s at state (8), and then enters the boiler.

3. ANALYSIS OF THEORETICAL MODEL

To simplify the steam jet refrigeration cycle model, assumptions are also made as follows:

(1) The pressure losses of condenser, evaporator and the connection pipeline of system components are neglected;
(2) In addition to the condenser and evaporator, there is no heat exchange between other parts of the system and the environment;
(3) The nozzle efficiency η_n and diffuser efficiency η_d of the ejector are given values (Alexis and Rogdalis, 2003);
(4) The throttling process is seen as isenthalpic process;
(5) The subcooling degree and evaporation and condensation temperature are known;
(6) The pressure of two fluids into the suction chamber is the same and the given value, and the fluid in the ejector is one-dimensional homogeneous flow.

3.1 Energy Analysis

Based on the above assumptions, the expanding and accelerating process of the primary fluid in the primary nozzle meets energy conservation,

$$h_1 + u_1^2/2 = h_{2s} + u_{2s}^2/2 \tag{2}$$

$$m_1 = u_1 A_1 / v_1 \tag{3}$$

$$\eta_n = \frac{h_1 - h_2}{h_1 - h_{2s}} \tag{4}$$

Where A_1 refers to cross-section area of primary nozzle's inlet.

The velocity of the secondary fluid from the evaporator can be calculated by the formula:

$$m_7 = u_7 A_7 / v_7 \tag{5}$$

Where A_7 is the cross-section area of the inlet pipe in suction chamber which is connected with the evaporator.

The mixing process of two fluids in the mixing chamber satisfies momentum conservation and energy conservation,

$$(m_1 + m_7)u_3 = P_0(A_2 + A_7) + m_1 u_2 + m_7 u_7 - P_3 A_3 \tag{6}$$

$$(m_1 + m_7)(h_3 + u_3^2/2) = m_1(h_2 + u_2^2/2) + m_7(h_7 + u_7^2/2) \tag{7}$$

Where A_2 is the primary nozzle outlet cross-section area and A_3 is cross-section area of uniform mixing chamber.

The iterative program takes mass conservation of mixing process as the iterative criterion,

$$m_3 = u_3 A_3 / v_3 \tag{8}$$

The fluid state in ejector outlet is obtained through the overall energy balance:

$$(m_1 + m_7) \cdot (h_4 + u_4^2 / 2) = m_1 \cdot (h_1 + u_1^2 / 2) + m_7 \cdot (h_7 + u_7^2 / 2) \tag{9}$$

Where u_4 is fluid's velocity in the ejector outlet, by default initial value, and is obtained by iterative calculation.

The diffuser efficiency in diffusion process is:

$$\eta_d = \frac{h_{4s} - h_3}{h_4 - h_3} \tag{10}$$

Cooling capacity of steam jet refrigeration cycle is given as:

$$Q_0 = m_7 (h_7 - h_6) \tag{11}$$

Energy consumption of steam jet refrigeration cycle is presented by:

$$W = m_1 (h_1 - h_5) \tag{12}$$

The coefficient of performance for system is:

$$COP = Q_0 / W \tag{13}$$

3.2 Exergy Loss Analysis of Individual Components

The exergy loss of the condenser (Jiang *et al.*, 2007):

$$I_k = (m_1 + m_7) \cdot [h_4 + u_4^2 / 2 - h_5 - T_m (s_4 - s_5)] - (m_1 + m_7) \cdot (1 - \frac{T_m}{T_k}) \cdot (h_4 - h_5) \tag{14}$$

The exergy loss of throttle valve:

$$I_f = m_7 T_m (s_6 - s_5) \tag{15}$$

The exergy loss in the evaporator:

$$I_0 = m_7 \cdot [h_6 - h_7 - u_7^2 / 2 - T_m (s_6 - s_7)] - m_7 \cdot (1 - \frac{T_0}{T_m}) \cdot (h_7 - h_6) \tag{16}$$

The exergy loss in the boiler:

$$I_e = m_1 \cdot [h_8 - h_1 - u_1^2 / 2 - T_m (s_8 - s_1)] + m_1 \cdot (1 - \frac{T_m}{T_s}) \cdot (h_1 - h_8) \tag{17}$$

The exergy loss of primary fluid passing through the primary nozzle:

$$\pi_1 = m_1 [h_1 + u_1^2 / 2 - h_2 - u_2^2 / 2 - T_m (s_1 - s_2)] \tag{18}$$

The exergy loss of mixing process in mixing chamber:

$$\pi_2 = m_1[h_2 + u_2^2/2 - h_3 - u_3^2/2 - T_m(s_2 - s_3)] + m_7[h_7 + u_7^2/2 - h_3 - u_3^2/2 - T_m(s_7 - s_3)] \tag{19}$$

The exergy loss in pressure expanding process in the subsonic diffuser:

$$\pi_3 = (m_1 + m_7)[h_3 + u_3^2/2 - h_4 - u_4^2/2 - T_m(s_3 - s_4)] \tag{20}$$

Where T_m is the ambient temperature and given as 300K; T_k is the Condensing temperature, K; T_0 is the Evaporating temperature, K; T_s is the Boiler temperature, K.

3.3 The Iterative Calculation Program

According to the above thermodynamic analysis, the designed iterative program is shown in Fig.3.

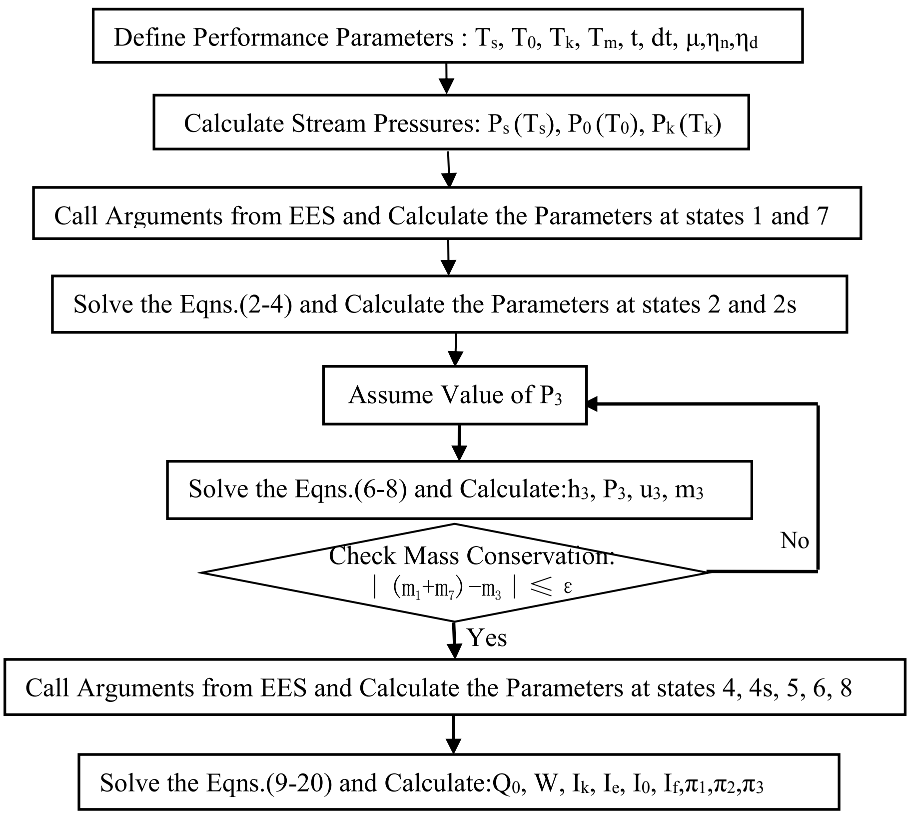

Figure 3: Solution algorithms of the iterative process

4. RESULTS AND DISCUSSION

According to the system's design condition, in which the boiler temperature is 423K, the condensing temperature is 310K and the evaporating temperature is 277K, the ejector was designed. The EES (Klein S *et al.*2011) was employed to solve the iterative program and some results were obtained.

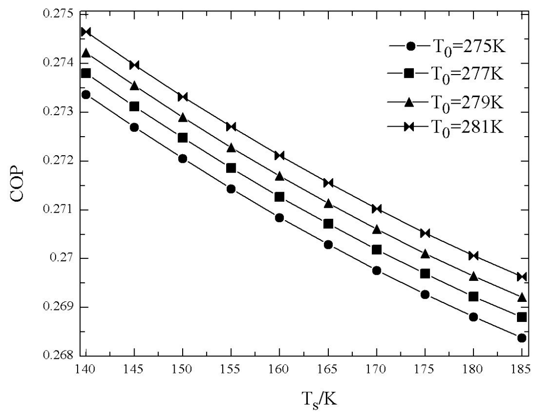

Figure 4: COP of ejector refrigeration cycle versus boiler temperatures and evaporating temperatures

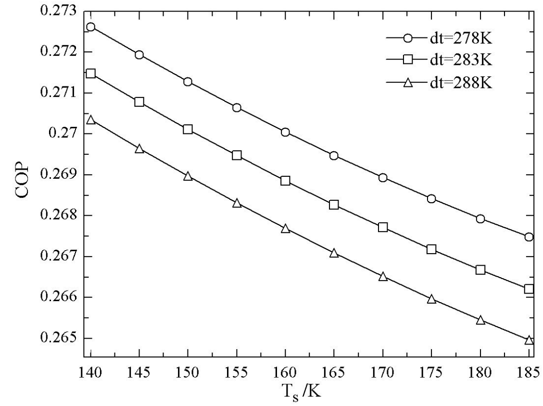

Figure 5: COP of ejector refrigeration cycle versus boiler temperatures and superheat degree

Fig.4 shows that increasing the boiler temperature and decreasing the evaporating temperature decreases the coefficient of performance (COP) and it has been proved by Ahmed and Chandra (2014). When the boiler temperature is constant, increasing the evaporating temperature increases the pressure difference between the evaporator and the primary nozzle outlet and therefore the secondary fluid passing through the mixing chamber is increased and caused the entrainment ratio and COP increase. When the evaporating temperature is constant, the increase of boiler temperature increases the expansion angle in the primary nozzle outlet of the primary fluid and the mixing chamber is choked, thus the secondary fluid and the COP is decreased.

As shown in Fig.5, COP is decreased with the superheat degree increase of the primary fluid. The superheat degree increases the steam's quality in the primary nozzle outlet and decreases the entrainment ratio and COP.

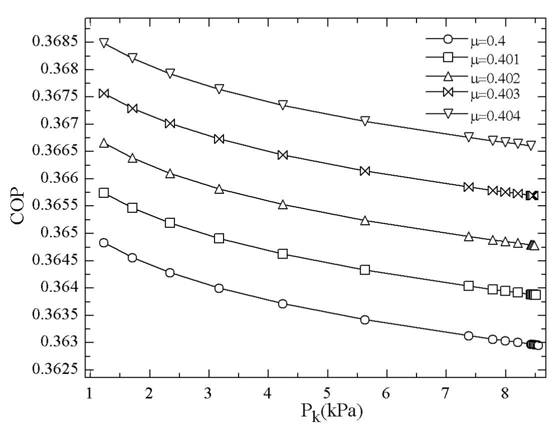

Figure 6: COP of ejector refrigeration cycle versus boiler temperatures and entrainment ratios

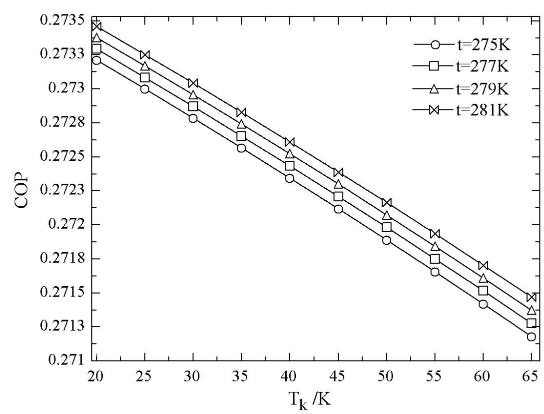

Figure 7: COP of ejector refrigeration cycle versus condensing temperatures and supercooling degrees

Effects of entrainment ratio and condensing pressure on the system's COP are shown in Fig.6. With the same entrainment ratio, the increase of condensing pressure causes a choke in ejector (Allouche *et al.*, 2013) and therefore decreases its performance. The choke in the ejector will be more serious with the growth of entrainment ratio. The entrainment ratio affects the COP very seriously; the relationship between them is $\Delta COP \propto \Delta\mu$ mostly.

Fig.7 shows the influence of the supercooling degree on system's COP for different condensing temperatures. It can be seen from the figure that the lower the condensing temperature, the higher the COP, and the rise of the supercooling degree increases the COP (Yu *et al.*, 2012).

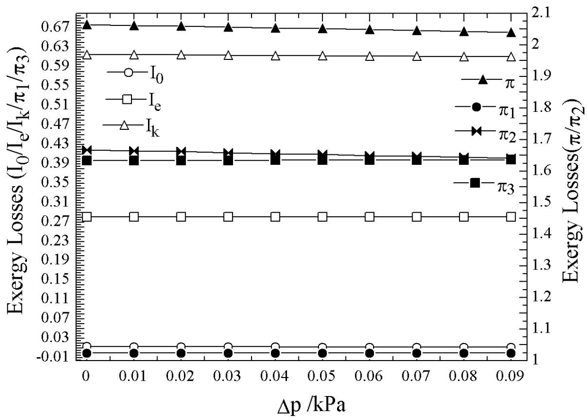

Figure 8: The exergy losses of different components versus pressure difference Δp

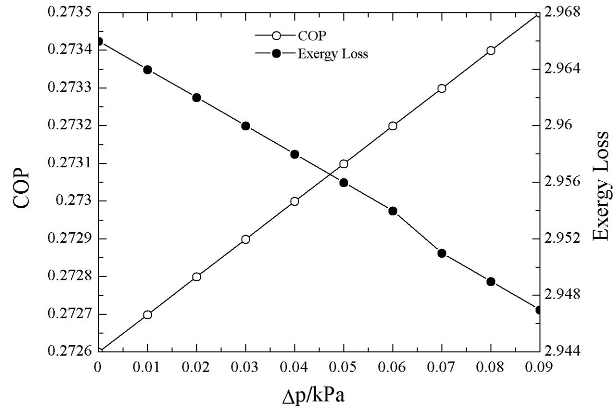

Figure 9: COP of ejector refrigeration cycle and exergy loss versus pressure difference Δp

Fig.8 and 9 provide a visual interpretation of the influence of pressure difference between the evaporator and the primary nozzle outlet on the COP and the exergy loss. It is clear from the figures that the pressure difference between the evaporator and the primary nozzle outlet has a great impact on the COP and the exergy loss, as for the higher pressure difference, the higher COP and the lower exergy loss for the ejector refrigeration system. It can be seen from Fig.8 that the maximum exergy loss occurs in the mixing process of the primary fluid and the secondary fluid. The entrainment ratio and the secondary fluid's velocity are increased with the increase of the pressure difference and the relative velocity between the primary fluid and the secondary fluid decreased; therefore the exergy loss in the mixing process is reduced. On the other hand, the increase of the pressure difference causes the ejector outlet pressure close to the condensing pressure and reduces the heat transfer loss and exergy loss in condenser.

5. CONCLUSIONS

This paper presents the theoretical calculation and analysis on steam jet refrigeration cycle and draws the following conclusions:

- Both increasing the temperature of boiler and the superheat of the primary fluid can cause choking within the ejector, entrainment ratio decreasing, the system's COP decreasing;
- The increase of the condensing pressure will reduce the system's COP while the increase of subcooling will increase the COP;
- The maximum exergy loss of system mainly exists in the mixing chamber. The pressure difference between evaporator and the primary nozzle outlet has a great effect on exergy loss in the mixing process. The key point of optimizing the system is optimizing ejector, improving the secondary fluid's velocity into the suction chamber, which can reduce the relative velocity of two fluids and the exergy loss of ejector.
- Most of the energy consumption of this experiment cycle is the bolier added heat. When the system is employed in the thermal power plant or other fields in which the low-grade water steam is generated, the system' COP will be improved rapidly.

NOMENCLATURE

A	Cross-section area	m^2
COP	Coefficient of performance	
h	The fluid's enthalpy	kJ/kg

I	The exergy loss	kJ
m	Mass flow rate	kg/s
P	Pressure	kPa
Δp	Pressure difference	kPa
Q_0	Cooling capacity	kW
s_*	The entropy at state *	kJ/(kg·k)
T	Temperature	K
u_*	The fluid's velocity at state *	m/s
W	Energy consumption	kW
γ_A	Area ratio	
μ	Entrainment ratio	
η_n	Nozzle efficiency	
η_d	Diffuser efficiency	
π_1	The exergy loss in the primary nozzle	kJ
π_2	The exergy loss of mixing process	kJ
π_3	The exergy loss in the subsonic diffuser	kJ

Subscript

*	state of 1, 2, 2s, 3, 4, 4s, 5, 6, 7, 8

REFERENCES

Chunnanond K., Aphornratana S., 2004a, Ejectors: applications in refrigeration technology, *Renewable and Sustainable Energy Reviews*, no. 8: p. 129-155.

Ma X.L., Zhang W., Omer S.A., Riffat S.B., 2010, Experimental investigation of a novel steam ejector refrigerator suitable for solar energy applications, *Applied Thermal Engineering*, no. 30: p. 1320-1325.

Chunnanond K., Aphornratana S., 2004b, An experimental investigation of a steam ejector refrigerator: the analysis of the pressure profile along the ejector, *Applied Thermal Engineering*, no. 24: p. 311-322.

Butterworth M.D., Sheer T.J., 2007, High-pressure water as the driving fluid in an ejector refrigeration system, *Applied Thermal Engineering*, no. 27:p. 2145-2152.

Varga S., Oliveira A.C., Diaconu B., 2009, Numerical assessment of steam ejector efficiencies using CFD, *Int. J. Refrig*., no. 32: p. 1203-1211.

Ruangtrakoon N, Thongtip T. et al., 2013, CFD simulation on the effect of primary nozzle geometries for a steam ejector in refrigeration cycle, *International Journal of Thermal Sciences, no.* 63: p. 133-145.

Alexis G.K., Rogdalis E.D., 2003, A verification study of steam-ejector refrigeration model, *Applied Thermal Engineering*, no. 23: p. 29-36.

Deng J.Q., Jiang P.X., Lu T., Lu W., 2007, Particular characteristics of transcritical CO_2 refrigeration cycle with an ejector, *Applied Thermal Engineering*, no. 27: p. 381-388.

Klein S, Alvarado F. Engineering equation solver. Middleton, WI: F-Chart Software; 2011.

Chandra V.V., Ahmed M.R., 2014, Experimental and computational studies on a steam jet refrigeration system with constant area and variable area ejectors, *Energy Conversion and Management*, no. 79: p. 377-386.

Allouche Y., Bouden C., Varga S.,2013, A CFD analysis of the flow structure inside a steam ejector to identify the suitable experimental operating conditions for a solar-driven refrigeration system, *Int. J. Refrig*, http://dx.doi.org/10.1016/j.ijrefrig.2013.07.027.

Yu W.Y., Tian R., Yan S.Y., 2012, Numerical simulation on the flow structure and performance of the ejector, *Journal of Engineering Thermophysics*, Vol. 33, no. 11: p.1881-1883.

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