

Simulation on Phase Change in Supersonic Flow in Ejector

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Abstract: For considering the probably existed phase change phenomena in ejector, the real steam properties were adopted in the computation of two dimensional fluid flow and performance of an ejector. The results present that the condensation happens in converging-diverging nozzle and in mixing chamber. The water content could reach near 10% at the outlet of the primary nozzle. It makes a two-phase flow. Compared the computational results under different motive fluid superheating conditions, velocity distributions in the nozzle have a great change. The entrainment ratio increases about 3.7%.

Keywords: ejector, condensation, supersonic flow, nozzle

NOMENCLATURE

ρ	density, kg/m ³
u	velocity, m/s
P	absolute pressure, Pa
μ	kinetic viscosity, Pa s
h	enthalpy, kJ/kg
s	entropy, kJ/Kg K
x	condensation quality
T	temperature, K

Subscripts

g	gas phase(saturated or overheat)
l	saturated water
m	motive fluid
s	suction fluid
d	discharge fluid

1. INTRODUCTION

A steam ejector is a device in which the steam is used to transport vapour or liquid at low pressure to a higher pressure or to make vacuum. Its main characteristic is that no moving parts are needed, and no electric power or mechanical power is consumed for its functioning. The power for transportation of secondary fluid is from the direct thermodynamic conversion process of thermal energy or the kinetic energy of the motive or primary steam. As its simplicity and reliability, it is widely used in power systems, refrigeration system, heat pump, etc.

In an ejector, a converging-diverging nozzle is used to accelerate steam flow to supersonic velocity. Historically, the ejector is designed according to the calculation of perfect gas, one-dimensional isentropic flow process [1]. From 70's of last century, the

numerical method was used to compute the fluid flow, wave formation and inter-reaction and other phenomena in an ejector [2].

Ejector refrigeration systems have attracted many research activities in recent years. Because these systems usually operate with steam as refrigerant in many industries, these systems can curb the global warming and ozone depletion from CFC and HCFC fluids, which are widely used in vapour compression refrigerators. However, steam ejector systems have a very low COP values owing to low performance of steam ejector [3,4]. Over the years, many studies have been conducted on the performance of the ejector. A good understanding of the flow phenomena in the steam ejector is critical in its design and operation.

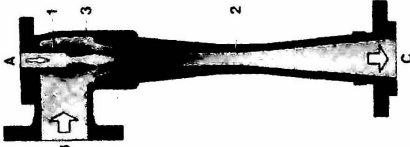
For the steam ejector with the motive fluid is saturated or near saturated steam, it is discovered by experiments that the condensation will happen in the convergence-divergence nozzle [5-7]. Since the properties of motive fluid have a great influence on the performance of the ejector, the condensation should be considered in the analysis of an ejector. The geometry of the nozzle should also fit with the change of the fluid properties. To analyze the phase change phenomena in the nozzle, a two dimensional numerical computation was performed, taking into account the change of steam properties since phase change. This paper will present the computational results on the condensation in the motive nozzle of an ejector.

2. DESCRIPTION AND MODELLING OF THE PROCESS IN EJECTOR

2.1 Description of Process in Ejector

In a steam ejector, the motive steam with relatively high pressure flows through a converging-diverging nozzle to be accelerated to supersonic. At

the outlet of the nozzle, the pressure is lower than that of the secondary fluid and it is drawn into the suction chamber of the ejector. Whereafter both the motive and suction fluids flow into the mixing chamber, in which the two streams get mixed and a subsonic, homogeneous flow velocity is formed at the end of the mixing chamber. And then the mixing stream flows into the diffuser. In diffuser the flow velocity is slow down and the pressure gets recovered to a certain level. The flow process and the general geometry of an ejector are shown in Fig. 1.



A – Motive fluid, B – Secondary fluid, C – Discharge
1 – Primary nozzle, 2 – Mixing chamber,
3 – Suction chamber

Figure 1: Geometry and flow process of ejector

In the case that the motive fluid is saturated steam, a part of the fluid will get condensed in the primary nozzle. It could be seen from the enthalpy-entropy chart of water and vapour, as shown in Fig. 2. Normally, as a reasonable simplification, the process in a nozzle could be taken as an isentropic process. In an isentropic expansion process of a nozzle, the process is from the point 1 at the cross of saturation line and inlet pressure line downwards to the point 2 on the outlet pressure line. The state of point 2 is in the region with both vapour and water, that is to say, the condensation happened in the process. For the real process, the process line should be inclined to the right with the increase of entropy, as the point 2' in Fig. 2. It is still in the vapour and water mixing region. Since the condensation, the properties of the motive fluid change sharply. That will cause the change of the performance of the ejector since it is related with the properties of the motive fluid closely. For precise prediction of the performance of ejectors, the condensation must be considered in the computation of ejector flow process.

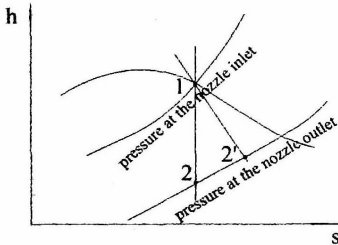


Figure 2: Steam isentropic expansion process in enthalpy-entropy chart

2.2 Modelling of the Process in Ejector

In modelling the fluid flow in converging-diverging nozzle, the two dimensional mass conservation, momentum conservation and energy conservation equations were used as the governing equations.

$$\frac{\partial \bar{p}}{\partial t} + \sum \frac{\partial (\bar{p} \bar{u}_i)}{\partial x_i} = 0 \quad (1)$$

$$\begin{aligned} \frac{\partial}{\partial t} (\bar{p} \bar{u}_i) + \sum \frac{\partial}{\partial x_j} (\bar{p} \bar{u}_i \bar{u}_j + \bar{p} \bar{u}_i' \bar{u}_j') = - \frac{\partial \bar{p}}{\partial x_i} \\ + \mu \left(\sum \frac{\partial^2 \bar{u}_i}{\partial x_j^2} + \sum \frac{\partial^2 \bar{u}_j}{\partial x_i \partial x_j} \right) + \bar{\lambda} \sum \frac{\partial^2 \bar{u}_j}{\partial x_i \partial x_j} + \bar{p} \bar{F}_i \end{aligned} \quad (2)$$

$$\begin{aligned} \frac{\partial}{\partial t} (\bar{p} \bar{h}) + \sum \frac{\partial}{\partial x_j} (\bar{p} \bar{u}_j \bar{h} + \bar{p} \bar{u}_j' \bar{h}') = - \bar{p} \sum \frac{\partial \bar{u}_j}{\partial x_j} \\ + \lambda \sum \frac{\partial^2 \bar{T}}{\partial x_j^2} + \mu \left\{ 2 \sum \left(\frac{\partial \bar{u}_j}{\partial x_j} \right)^2 + \sum_i \sum_j \left[\left(\frac{\partial \bar{u}_i}{\partial x_j} \right)^2 + \frac{\partial \bar{u}_j}{\partial x_i} \frac{\partial \bar{u}_i}{\partial x_j} \right] \right\} + \bar{\lambda} \left(\sum \frac{\partial \bar{u}_j}{\partial x_j} \right)^2 + \bar{S}_h \end{aligned} \quad (3)$$

For the computation of the phase change process, the steam properties (density, specific heat capacity, kinetic viscosity) were adopted in the model, replacing the normal assumption of idea gas for the calculation of the flow in a nozzle. While the condensation occurs, the properties of the vapour and droplet mixture are calculated with the following formula:

$$y = \frac{y_g * y_l}{y_l * x + (1 - x) y_g} \quad (4)$$

$$x = \frac{h - h_l}{h_g - h_l} \quad (5)$$

where, y stands for three property parameters, i.e. the density of the mixture; the specific heat capacity of the mixture at constant pressure, and the conduction coefficient of the mixture. The other property parameters, such as kinetic viscosity, were taken as that of the saturated steam.

The Chen-Kim $k-\epsilon$ model was taken to simulate the turbulent phenomenon. It seems that it is more suitable for the computation of flow in nozzle than the LVEL model, the standard $k-\epsilon$ model and the RNG $k-\epsilon$ model from our comparing computation, and it has a better convergence character [8].

The adiabatic, non-slip boundary conditions were applied to the nozzle walls and ejector walls. The upstream conditions were assumed to be isotonic, isothermal and uniform velocity at both motive and suction fluid inlets. The downstream condition was

given with a designated discharge pressures at the outlet of the ejector.

3. NUMERICAL METHODS

A commercial CFD code based on finite volume method was used to solve the governing equations. The body-fitted meshes with MBFGE(Multi-Blocking and Fine Grids Embedding) technique were adopted. The whole computation domain was divided into 11 blocks according to the character of geometries of the ejector and the need for local computation, as shown in Fig. 3(a). A part of the meshes are shown in Fig. 3(b). Block 3-6 and block 11 were refined to get clear view of the flow there, as shown in Fig. 3(b). Total grids are about 100,000 cells.

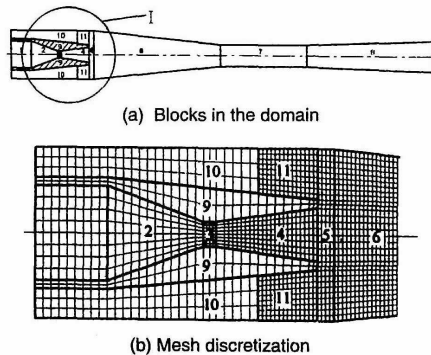


Figure 3: Discretization of the computation domain

The MBFGE grid for a problem is the grid system that is built from a set of separate grids and linked together through common boundaries using link-scheme developed for MBFGE method. Separate grids in a MBFGE grid can be whether the grids created for parts of complex computational domain, or fine grids embedded into the coarse ones, or any combination of these. Therefore computational grids for each of separate blocks are structured. This can take the advantage of structure grids, i.e. computation economics, and can increase the local resolution.

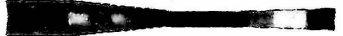
To get the solutions of nonlinear equations on the MBFGE grid, a specially developed unstructured version of 2-step Jacobi preconditioned conjugate residuals solver is adopted. As a result only the overall number of cells governs the rate of convergence and it is not affected by the number of linked domains or links set in the MBFGE grid.

4. RESULTS AND DISCUSSIONS

To show the differences among computation results between using steam real properties, i.e. considering phase change, and using ideal gas assumptions, the computational temperature distributions with different fluid properties are shown in Fig. 4.

The Fig. 4(a) is the results with real steam properties. The Fig. 4(b) shows the result with the ideal gas properties. In both computations the steam states are set the same: the motive steam is saturated with 0.037MPa, the suction steam is saturated with 0.0013MPa and the discharge pressure is 0.003MPa. From the figures it can be seen that with different conditions, the temperature distributions have a substantially difference. In Fig. 4(a), the temperature decreases gradually through the nozzle and equals to the saturation temperature corresponding to the local pressure. It is coincident with the process in ejector, as shown in Fig. 2. It is 56.1°C at the throat and 6.1°C at the outlet of the primary nozzle. In Fig. 4(b), the computed temperature in the nozzle is lower than that in Fig. 4(a). The temperature downwards to the throat of nozzle is below 0°C, which is not reasonable for steam. In addition, the computed temperature of the suction steam around the nozzle jet is higher than the suction fluid inlet temperature, which is not reasonable either. It is clear that for the computation of a steam ejector, it is unacceptable to simplify the steam properties as that of the ideal gas. The steam property code is emerged into the commercial CFD code for the computation of steam flow in ejector and the phase change phenomenon is revealed.

TEMP
7.427E+01
6.967E+01
6.513E+01
6.059E+01
5.605E+01
5.151E+01
4.697E+01
4.243E+01
3.789E+01
3.335E+01
2.881E+01
2.427E+01
1.973E+01
1.519E+01
1.065E+01



(a) With real steam properties

TEMP
1.62E+02
1.417E+02
1.212E+02
1.006E+02
8.002E+01
5.946E+01
3.889E+01
1.032E+01
-2.241E+00
-2.281E+01
-4.337E+01
-6.394E+01
-8.451E+01
-1.051E+02
-1.250E+02



(b) With ideal gas properties

Figure 4: Computational temperature distributions in ejector with different conditions

Fig.5 shows the computed water content variation along the axial distance under the different superheat

levels of the inlet motive steam, with the real steam properties. Three vertical lines in the figure show the position of the nozzle throat, the nozzle outlet, and the diffuser inlet. The given pressures are, $p_m = 0.037 \text{ MPa}$, $p_s = 0.0013 \text{ MPa}$ and $P_d = 0.003 \text{ MPa}$ respectively. It could be seen that the phase change occurred at the throat of the converging-diverging nozzle. Normally, at some position downwards the outlet of the nozzle, the water content reaches the maximum value. Then it changes in fluctuant and some water droplets get re-evaporated in the mixing chamber or in the diffuser. When the superheating level of the motive steam reaches 40°C at the given pressures, the water droplet will disappear in the mixing chamber and there will be no condensate in the diffuser. The maximum condensate content is about 9.5% for the motive steam being saturated.

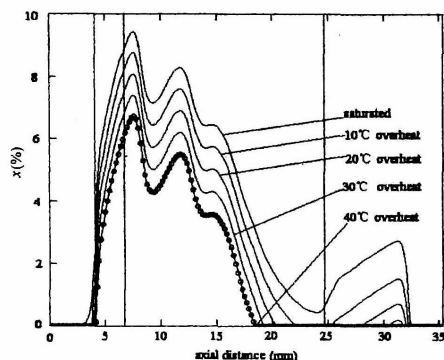


Figure 5: Water content variation with axial distance under different superheating of the motive steam

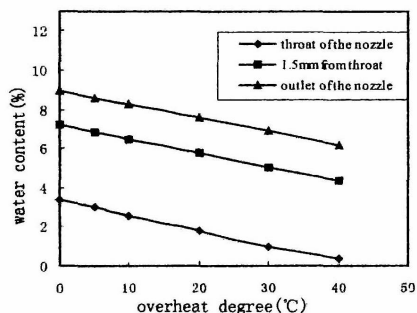


Figure 6: Water content variation with superheating of the motive steam

Fig.6 shows how the water content at three position in the nozzle varied with the superheat level of the motive steam. With the increase of the superheat level, the water content gets decreased.

There exits a linear relationship between the water content and superheat level.

Since the condensing in the nozzle, the density of the motive fluid changes in a certain degree. It causes the changes of other flow parameters. Fig.7 shows the speed differences between the flow at superheating condition and that at the saturated condition. Four lines represent four positions, i.e. the throat of the nozzle, 0.8mm after the throat, 1.6mm after the throat and the outlet of the nozzle, respectively. Although the pressures are the same, with the increasing of the superheat level, the flow speed will increase accordingly. It causes the entrainment ratio of the ejector changes from 0.396 to 0.411, as shown in Fig. 8, about 5% higher. This proves that the phase change in the ejector has a distinct influence on the flow process in the ejector and the entrainment ratio of the ejector.

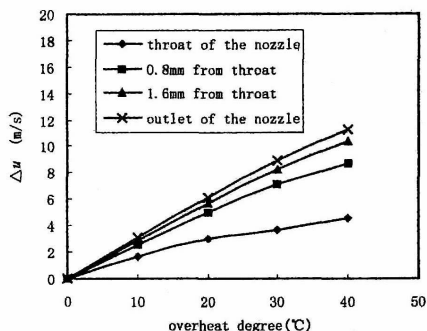


Figure 7: Flow speeds difference in the nozzle for different inlet superheating conditions

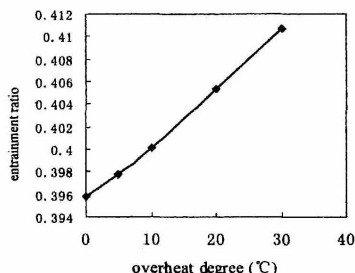


Figure 8: Entrainment ratios of the ejector for different inlet superheating conditions

5. CONCLUSIONS

1. The phase change in ejector is revealed by the computation of flow with the steam properties. It shows that to assume the steam as real gas in ejector computation is unacceptable.
2. Steam condensation occurs at the throat of the converging-diverging nozzle when the primary

steam superheating is not too high. The water content normally reaches the maximum value at some position downwards the outlet of the nozzle. Then it changes in fluctuant mode in the mixing chamber and decreases gradually. It may occur again in the diffuser.

3. Condensate content will decrease when the superheat of the motive fluid increase. When it is 40 °C or higher at the given pressures, the water content by condensation is low and will no longer exit in the diffuser.
4. Phase changing process affects the performance of the ejector. Through increasing superheating conditions of motive fluid (from 0°C to 40°C) can reduce condensation and increase entrainment ratio of the ejector (from 0.396 to 0.411) at the given pressure conditions here. The relative increase value of the entrainment ratio is 3.75%.

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