

Study on the Nozzle Exit Position of an ejector in a Combined Cooling system

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ABSTRACT: In this study, 60 ejectors designed for CFD calculation. The purpose of this study is to investigate the use of CFD in prediction performance of an ejector used refrigeration applications. An ejector analyses often use coefficients derived from experimental data for a set operating conditions with limited functionality. But, this study is two dimensional calculation of ejector performance. The ejector was designed three kind of primary nozzle constructed with three different throat diameters, for the purpose of examining the influence of primary nozzle geometry of ejector performance. These nozzles had the same area ratio (throat to the exit plane) of 16 and included divergent angle of 10°.

Keywords: ejector, CFD, supersonic

NOMENCLATURE

A	area, [m^2]
d	diameter, [m]
h	enthalpy, [$KJ\ kg^{-1}$]
$\dot{m}$	mass flow rate, [$kg\ s^{-1}$]
M	Mach number
P_e	vapor pressure at the suction port of the ejector, [MPa]
P_g	vapor pressure at the nozzle inlet port of the ejector, [MPa]
T	temperature, [K]
T_e	vapour temperature at the suction port of the ejector, [K]
T_g	vapour temperature at the nozzle inlet of the ejector, [K]
T_{gs}	saturated vapour temperature corresponding to P_{gs} , [K]
V	gas velocity, [ms^{-1}]

x	nozzle position, [m]
y	position of the hypothetical throat

SUPSCRIPTS

c	exit of ejector, condenser
e	inlet port of the entrained flow
g	nozzle inlet
m	mixed flow
p	primary flow
pl	nozzle exit
py	primary flow at the location of choking for the entrained flow
s	suction or entrained flow
sy	entrained flow at the location of choking for the entrained flow
t	nozzle throat
y	location of choking for the entrained flow
1	nozzle exit
2	entrance of the constant-area section
3	exit of the constant-area section

1. INTRODUCTION

Ejector cooling system can be considered as one of the most suitable refrigeration systems for the present energy and environment situations. It utilizes a low temperature thermal energy from inexpensive or even free sources such as industrial wasted heat or a solar collector. The working fluid used is water or refrigerant, the most environmental friendly substance. The major disadvantage of ejector refrigeration is its relatively low coefficient of performance, compared to other types of refrigeration cycles.

Their typically low COP of around 0.2 can be improved substantially when combined with other systems such as absorption or vapour-compression [1].

The purpose of this study is to investigate the use of CFD in prediction performance of an ejector used refrigeration applications.

The Key problem is in the ejector design. The ejector design can be classified into two categories according to the position of the nozzle [1]. For the nozzle with its exit located within the constant-area section of an ejector, the mixing of the primary and the entrained flows occurs inside the constant-area section and the ejector is known as “constant-pressure mixing ejector”. For the nozzle with its exit located within the suction chamber which is in front of the constant-area section and the ejector is known as “constant-area mixing ejector”. For this kind of ejector, it was assumed that the mixing of the primary and the entrained streams occurs in the suction chamber with a uniform or constant pressure. It is known that the constant-area ejector and is thus widely used. Therefore, we focus on the design of a “constant-pressure ejector” in the present study, but with a new concept that the mixing occurs within the constant-area section.

The constant-area mixing theory of ejector developed by Keenan et al. [2] It was frequently used in the analysis of constant-pressure ejector. It is assumed that the pressure of the primary and the entrained

flows at the exit of the nozzle have an identical pressure. Mixing of the two streams begins there with a uniform pressure, constant pressure, until the inlet of the constant-area section.

In practice, two choking phenomena exist in the ejector performance. [3], [4] One in the primary flow through the nozzle and the other in the entrained flow. In addition to the choking in the nozzle, the second choking of an ejector results from the acceleration of the entrained flow from a stagnant state at the suction port to a supersonic flow in the constant-area section.

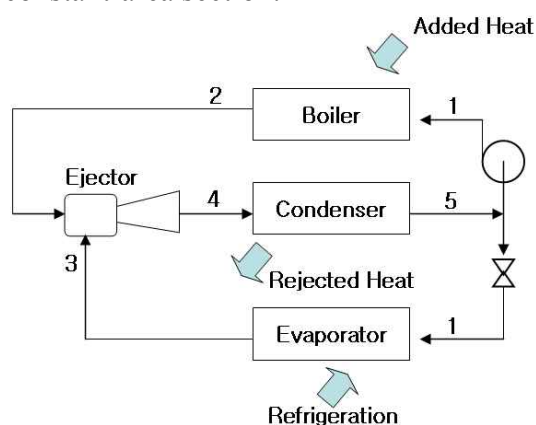


Figure 1: Ordinary ejector refrigeration system

Fig. 1 shows a schematic diagram of an ordinary ejector refrigerator unit. This cycle is similar to the conventional vapour compression system except that the compressor is replaced by a liquid circulation pump, boiler and ejector. Briefly, as heat is added to the boiler, the evaporated refrigerant is evolved at high temperature and pressure (2). This high pressure refrigerant, which can be called either “primary fluid” or “motive fluid”, expands through the primary nozzle in an ejector and produces a very low pressure region at the primary nozzle exit plane (3). This low pressure allows a liquid refrigerant in the evaporator to vaporize at low temperature to create the refrigeration effect. Heat used to vaporize this refrigerant is the cooling load of the system. The evaporated “secondary fluid” will be entrained from the evaporator and mixed with the primary fluid in a mixing

chamber of the ejector. The mixed stream is discharged via the diffuser to a condenser (4), where the vapour is condensed (5). The accumulated liquid refrigerant in the condenser is returned back to the boiler by the feed pump (1) whilst the remainder is expanded through the throttling valve to the evaporator (6), to complete the cycle.

2. MODEL DEVELOPEMT

2.1 Model Description

Geometries of the ejector are described in Fig. 2. The ejector was designed based on the methods provided in the literature [4, 5]. There were three primary nozzle constructed with three different throat diameters, for the purpose of examining the influence of primary nozzle geometry of ejector performance. These nozzles had the same area ratio (throat to the exit plane) of 16 and included divergent angle of 10° . The significant geometries of the nozzle were listed in Table 1. There were four nozzle exit position as $0.5, 1.0, 1.5$ and $2.0 d_3$. So, all of the designed ejectors are 60.

The primary nozzle was mounted with a threaded shaft which allowed for vertical movement of the nozzle. Thus the distance between the nozzle exit and the mixing chamber inlet can be adjusted in order to examine its influence on the performance of the ejector. Along the ejector axis, a static pressure at each operating conditions was tapped. This information can be used to

analyze and explain the flowing and mixing characteristics of two working through the ejector. [5].



Figure 3: Significant geometries of the primary nozzle

2.2 Ejector design

The purpose of the study is to investigate the use of CFD in prediction performance of an ejector used refrigeration applications. An ejector analyses often use coefficients derived from experimental data for a set operating conditions with limited functionality. So, the ejector design for the combined ejector cooling system was instigated from a two-dimensional analysis. The generator temperature at 100°C enabled heat utilisation from various cheap sources such as industrial waste heat, flue gas, automobile exhaust or solar energy, etc. It is well documented in a conventional ejector refrigerating or air conditioning diminishes with increasing evaporator temperature. The combined system permits a much higher evaporating temperature, sufficiently high for the ejector to achieve high efficiently but low enough to allow the compression cycle to reject heat. The condensing temperature of the ejector sub-cycle at 10°C was chosen to be as low as practical for best system performance yet high enough to allow heat rejection to the sink.

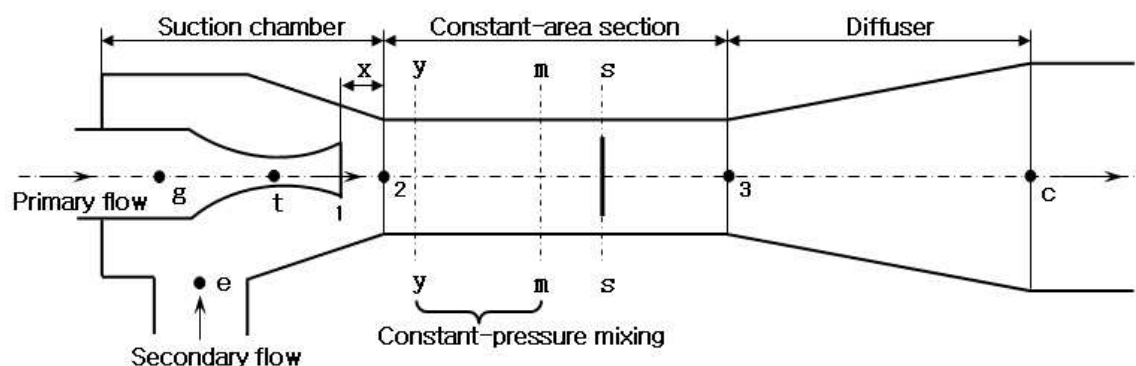


Figure 2: Ejector modelled in CFD and used as the base model.

3. CFD ANALYSIS OF AN EJECTOR PERFORMANCE

In order to simulation this particular situation. ICEM-CFD and Star-CD were used as the grid generator and the CFD solver, respectively. Fig. 4 shows grid system for 2-dimensional analysis. It generated by ICE-CFD.

3.2 Case Setup

As the super sonic flow field was expected, the flow was thought to be a turbulent compressible flow. The “Standard $\kappa-\varepsilon$ model” was selected to govern the turbulence characteristic. It was likely to provide a superior performance of the flows involving rotation, boundary layer under strong adverse pressure gradient and separation as presented

Table 1: Ejector geometry variation

Primary nozzles		Mixing chambers			Nozzle exit position
Nozzle	d_g mm	Mixing chamber	d_3 mm	A_3 / A_t	
A	2.64	A	6.70	6.441	$0.5 d_3, 1.0 d_3, 1.5 d_3, 2.0 d_3$
B	2.75	B	6.98	6.990	$0.5 d_3, 1.0 d_3, 1.5 d_3, 2.0 d_3$
C	2.93	G	7.34	7.730	$0.5 d_3, 1.0 d_3, 1.5 d_3, 2.0 d_3$
		C	7.60	8.287	$0.5 d_3, 1.0 d_3, 1.5 d_3, 2.0 d_3$
		D	8.10	9.414	$0.5 d_3, 1.0 d_3, 1.5 d_3, 2.0 d_3$

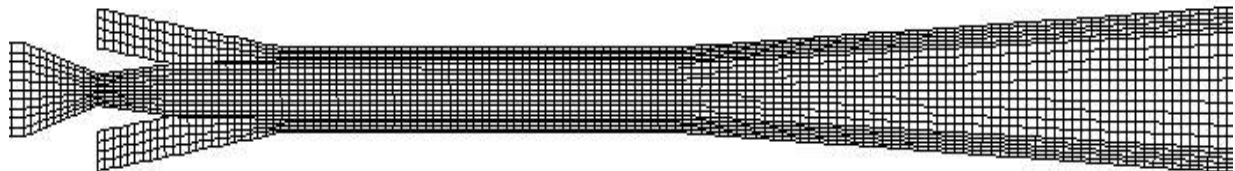


Figure 4: Grid system for 2-D calculation

3.1 Geometries and Grids

As proposed, ICEM-CFD was used to create the calculation domain and grid elements of the model. The mesh and model were created in a two dimension domain.

However, the axis metric solver was applied and therefore, the three dimension effect was taken into account in the simulation. The geometries of the calculation domain of the modelled steam ejectors were taken from those which were used outside data based on Huang et al. [5]. Their significant dimension has already been described in the previous section. The mesh was made 100,000 structured quadrilateral elements, as shown in Fig. 4.

in our ejector model.

3.3 Boundary Conditions

Boundary conditions of two faces entering a primary nozzle and ejector were set as pressure-inlet, whilst the on leaving ejector was set as pressure-outlet. These parameters were varied with the same operation condition as was conducted in the experiments. The values of each boundary were assigned as the saturation properties of each operating state. Since the velocity of the flow entering and leaving the domain was thought to be relatively small compared with the supersonic speed during the flow process of the ejector, there was no difference between an input of the stagnation pressure and static pressure.

4. RESULTS AND DISCUSSION

Aphornratana and Eames [6] reported that moving the nozzle into the mixing chamber reduced the system COP or cooling capacity with a resultant higher critical condenser pressure and vice versa. Varying the nozzle exit position by 20% with all other conditions unchanged did not affect the entrainment ratio significantly. However, the results seemed to support the characteristics above with lower entrainment ratio resulting when the nozzle exit was moved closer to the constant area duct entry. Keenan et al.' [2] reported as follows, at a nozzle exit distance $1.5 d_3$ the constant area duct diameter, the ejector produced better performance. [7].

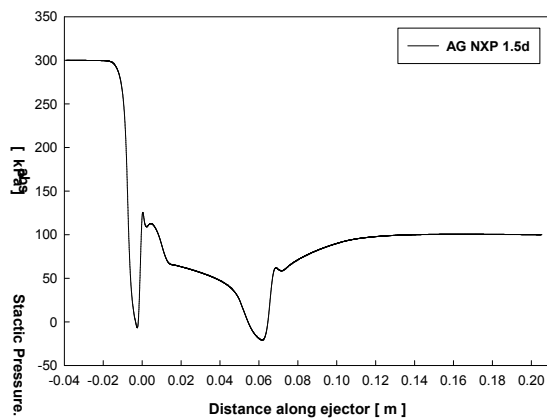


Figure 5: Static pressure distribution along the centreline of the ejector

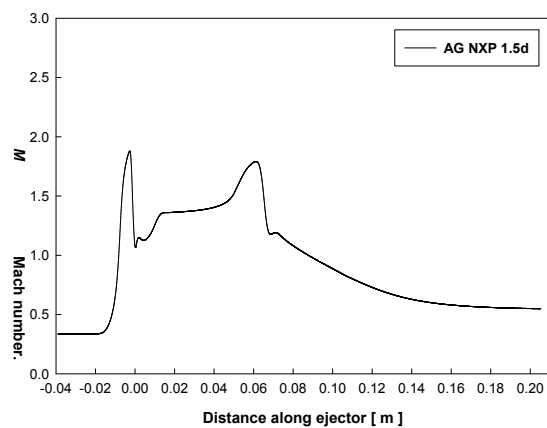


Figure 6: Mach number distribution along the centreline of the ejector

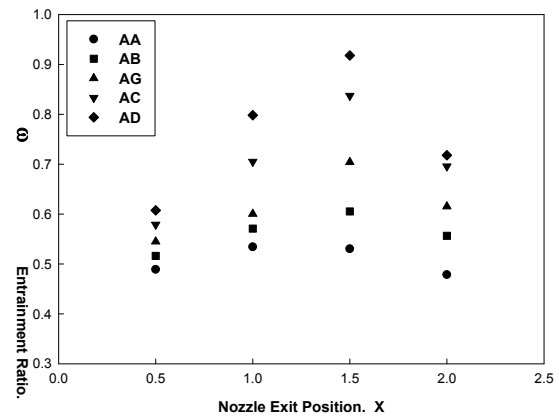


Figure 7: Variation of calculated entrainment ratio, effect of various NXP

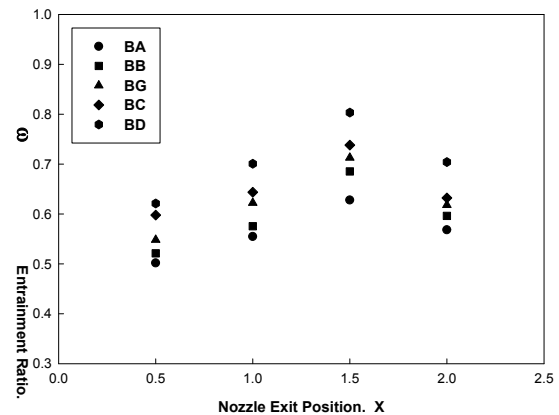


Figure 8: Variation of calculated entrainment ratio, effect of various NXP

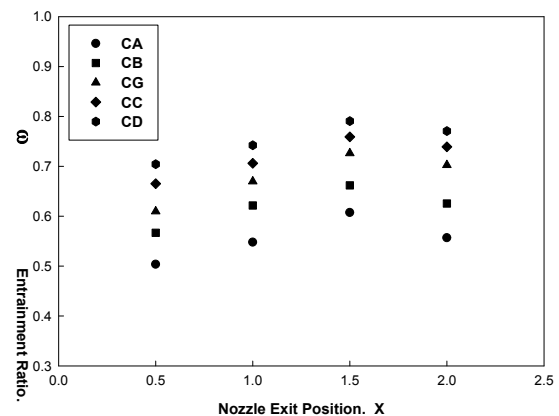


Figure 9: Variation of calculated entrainment ratio, effect of various NXP

In this study, we designed 60 ejectors. The ejectors simulated in CFD were A, B, C

series. Each code means nozzle and constant-area section identification codes used in the experiments of Huang et al. [1].

Fig. 5 and 6 represent Static pressure distribution and Mach number along the centreline of the ejector. Where 0.00 is primary nozzle exit position in the axis X. Fig. 5 and Fig. 6 shows typical supersonic ejector flow pattern.

Fig. 7 and 8, 9 represents the calculated entrainment ratio of the each series ejectors were varied. At each setting of the primary fluid and secondary fluid condition, the operation of an ejector is same. In the AA ejector, maximum entrainment ratio shows at $1.0 d_3$. In the other ejector, when a constant-area section area increased, the entrainment ratio slowly increased. At $1.5 d_3$ shows maximum entrainment ratio. In the Fig. 8 and 9, maximum entrainment ratio shows at $1.5 d_3$ on the all series. Totally, the ejectors has best performance around $1.5 d_3$, when increased primary nozzle area, entrainment ratio decreased, when increased constant-area section area, the entrainment ratio slowly increased. When the nozzle exit position increased 0.5 to $1.5 d_3$, the entrainment ratio slowly increased. In the $2.0 d_3$ shown the entrainment ratio decreased.

5. CONCLUSION

In the present study, we carried out a 2-D analysis for the prediction of the ejector nozzle exit position. So, we designed 60 ejectors, each ejector calculated with same operating condition.

The distance of nozzle exit calculated from the inlet of the constant-area section X is adjusted such that the ejector obtains the best performance at each operating condition. It is found that the entrainment ratio is around $1.50 d_3$ for the best ejector performance.

When constant-area section area increased, the entrainment ratio increased too. When

primary nozzle area increased, the entrainment ratio slowly decreased.

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