

CHAPTER 2

THEORIES AND LITERATURE REVIEW

2.1 Background of ejector

This chapter provides the ejector's background, theory, and its application in refrigeration technology. The literature reviews on the ejector and their application in refrigeration over the past decades are given.

An ejector was first invented in the late 1800s by Sir Charles Parsons [1]. It was used for removing air from a steam engine's condenser. Since then, an ejector has been applied and utilized in many chemical and industrial processes. The notable example is to produce a vacuum environment or to be used as a jet-conveyer in particulate solid transportation.

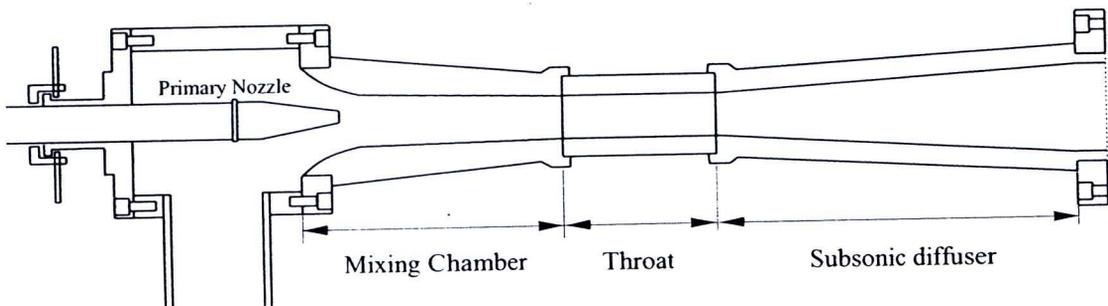


Figure 2.1: The component of ejector.

Basically, an ejector composes of four principle components: the primary nozzle, the mixing chamber, the throat, and the subsonic diffuser as shown in figure 2.1. A schematic view of the variation in static pressure and velocity as a function of a location along the ejector is shown in figure 2.2. The flow process is also presented in the Mollier's chart as shown in figure 2.3. The flow processes provided in these figures are based on the classical 1-D theory which was first proposed by Keenan et al, in 1940s [12]. This theory was based on an assumption of ideal gas combined with the conservative equation of mass,

momentum, and energy. Later, Munday and Bagster [9] explained the mixing process inside the ejector by introducing a concept of an *effective area*.

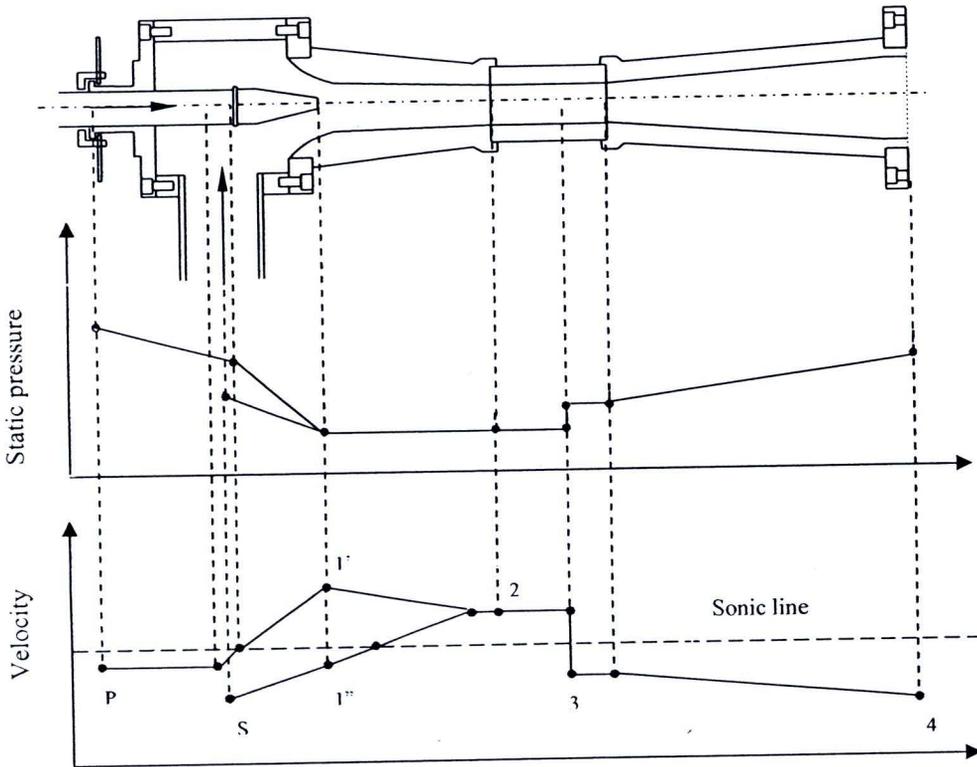


Figure 2.2: Variation of static pressure and velocity as a function of locations along an ejector [1].

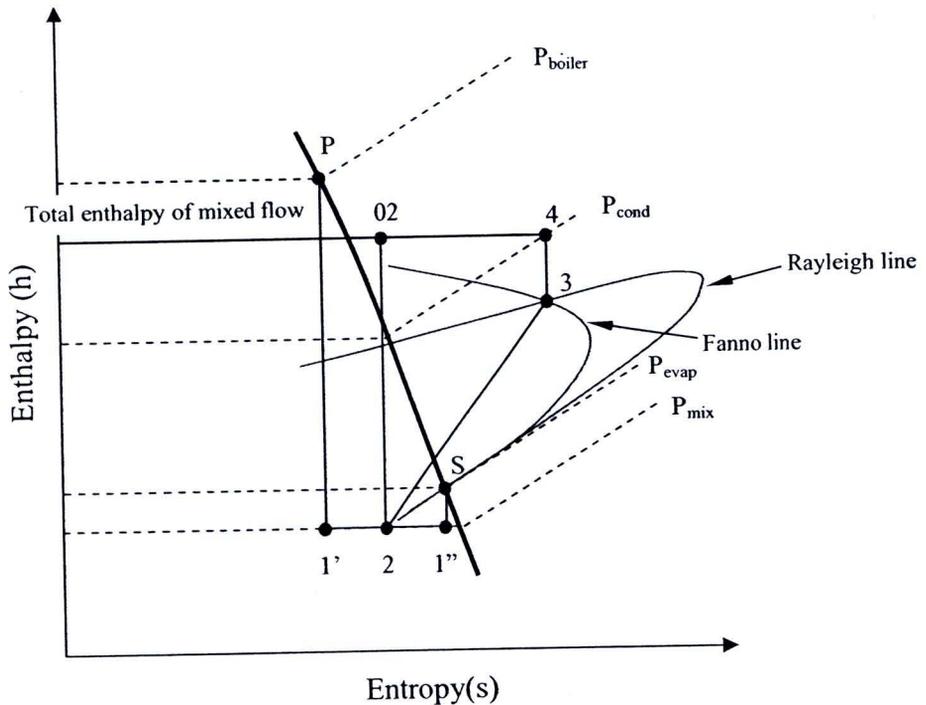


Figure 2.3: The Mollier's chart represents the flow process inside the ejector [1].

With reference to the figures, as a high pressure fluid vapour known as a *primary fluid* enters the primary nozzle (P), it is accelerated and fans out with supersonic speed at exit plane of the nozzle (1'). This causes the low pressure region to be produced in the mixing chamber. As a result, a low pressure fluid vapour known as a *secondary fluid* (S) can be entrained into the mixing chamber (1''). Inside the conical section of the mixing chamber, a converging duct for the secondary fluid is formed by the primary fluid jet core and the mixing chamber's wall. In this duct, the secondary fluid is accelerated (without mixing with the primary fluid) to sonic level at some section called an *effective area* [9]. Then the mixing process for the two fluids is begun. During the mixing process, the primary flow is retarded while the secondary flow is accelerated. At the end of mixing chamber (2), the two fluids are assumed to be completely mixed. The mixing process is assumed to be occurred at a constant pressure. Within the throat section, due to the pressure difference between the mixed streams and the ejector's downstream, a normal shock is induced (2-3). Based on this 1-D theory, the shock phenomenon is classified as a normal shock [7, 8]). This normal shock causes a major compression effect to be created within the ejector. Since, the shock is highly irreversible, the flow is not isentropic. This results in the major loss in the total pressure of mixed stream [18]. Across the shock, static pressure of the mixed stream is increased sharply and the flow is changed from supersonic to subsonic. The properties of fluid after shock can be defined by intersection between Fanno line and Rayleigh line [1, 18]. A further compression of the flow is achieved (4) as it is brought to stagnation through a subsonic diffuser. Performance of the ejector is defined in the term of entrainment ratio which can be defined as:

$$R_m = \frac{\text{mass flow of the secondary fluid}}{\text{mass flow of the primary fluid}} \quad (2.1)$$

2.2 Application of ejector in refrigeration technologies

A schematic diagram of a jet refrigeration system is presented in figure 2.4. The stage of the working fluid is plotted on the temperature-entropy diagram (T-s diagram) as shown in figure 2.5. When this system is compared with the commonly used vapour compression refrigeration cycle, the mechanical compressor is replaced with the boiler, the ejector, and the liquid pump. The ejector is used to elevate the working fluid (refrigerant) pressure. In others words, the main purpose of the ejector is similar to the mechanical compressor.

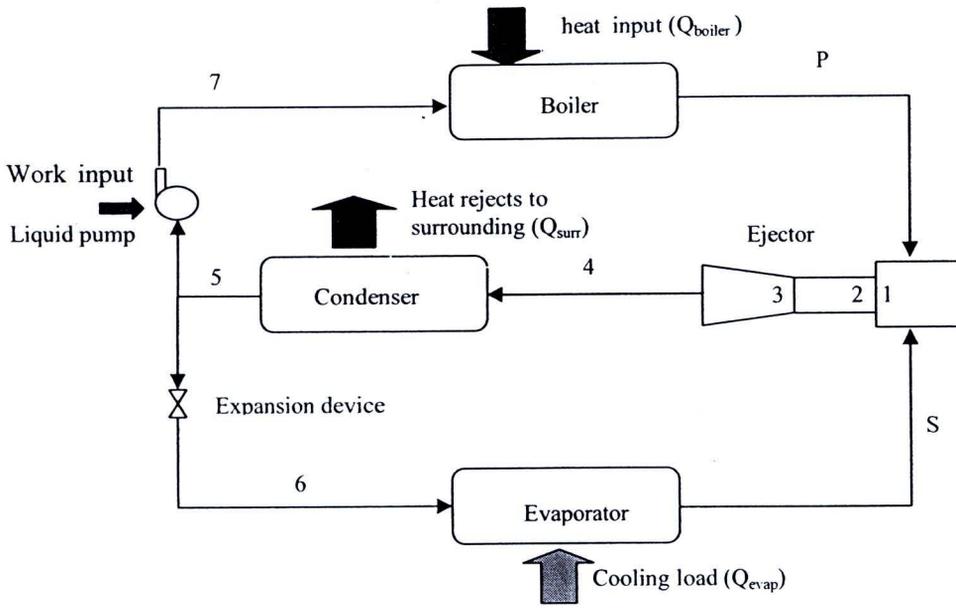


Figure 2.4: A typical jet refrigeration cycle.

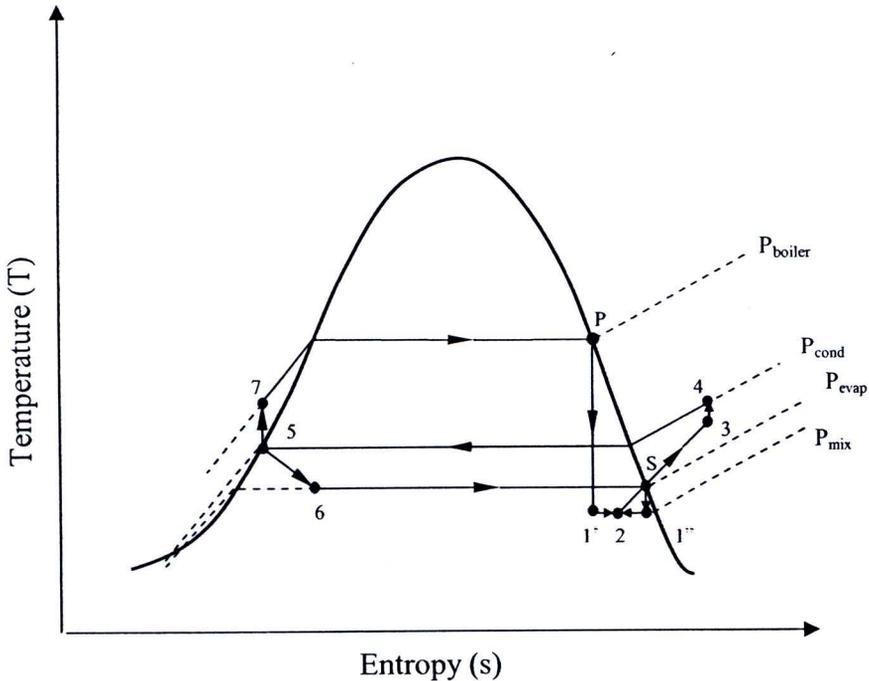


Figure 2.5: The process of jet refrigeration cycle.

From the figure 2.4, as heat is added the boiler (Q_{boiler}), the high pressure and high temperature (P) fluid is produced. It is used as the primary fluid for the ejector. The primary fluid is accelerated through the primary nozzle to produce a low pressure zone in

the mixing chamber (1'). As a result, low pressure and low temperature fluid known as secondary fluid (S) from the evaporator is entrained in to the mixing chamber (1''). As a result, useful refrigeration (\dot{Q}_{evap}) is produced at the evaporator. The two fluids are then mixed in the mixing chamber (2). Within the ejector's throat section, the shock is induced to create the major compression effect (3). Further compression effect is achieved as the mixed fluid is passed through the subsonic diffuser before being discharged to the condenser (4). At the condenser, the ejector's exhaust is cooled and condensed to liquid (5) as it rejects heat (\dot{Q}_{surr}) to the surroundings. Part of the condensate accumulated in the condenser is returned to the evaporator via expansion device (6) and the remainder is pumped back to the boiler via the liquid pump (7), and therefore, the cycle is completed.

Often, the operating condition of boiler, evaporator and condenser of a jet refrigeration cycle are defined by heat source, refrigerated purpose, and local climate respectively. The saturation properties of boiler and evaporator are considered as upstream condition of ejector while the saturation properties of condenser is considered as downstream condition of the ejector.

Performance of a jet refrigeration system is usually defined in terms of Coefficient of Performance (COP) which is related to the energy efficiency of a refrigeration cycle. The COP is defined as:

$$\text{COP} = \frac{\dot{Q}_{\text{evap}}}{\dot{Q}_{\text{boiler}} + \dot{W}_{\text{pump}}} \quad (2.2)$$

Where \dot{Q}_{evap} is the refrigeration effect produced at the evaporator (W)

\dot{Q}_{boiler} is the heat input require by the boiler (W)

\dot{W}_{pump} is the energy required by the pump (W)

The energy input required for the pump is very small and less than 1% of the heat input required at the boiler. Therefore, it is usually neglected from the calculation. The COP is then simplified to:

$$\text{COP} = \frac{\dot{Q}_{\text{evap}}}{\dot{Q}_{\text{boiler}}} \quad (2.3)$$

The refrigeration effect produced at the evaporator is:

$$\dot{Q}_{\text{evap}} = \dot{m}_s \cdot (h_s - h_6) \quad (2.4)$$

Where \dot{m}_s is the mass of the of the secondary fluid entrained by the ejector (kg/sec)

h_s is the enthalpy of the secondary fluid which is a saturated vapour at the evaporator saturation temperature (kJ/kg.K)

h_6 is the enthalpy of the fluid at the evaporator inlet which is equal to that of the saturated liquid accumulated in the condenser (kJ/kg.K)

The heat input to the boiler is:

$$\dot{Q}_{\text{boiler}} = \dot{m}_p \cdot (h_p - h_7)$$

Where \dot{m}_p is the mass of the of the primary fluid required by the ejector (kg/sec)

h_p is the enthalpy of the primary fluid which is a saturated vapour at the boiler saturation temperature (kJ/kg.K)

h_7 is the enthalpy of the fluid at the boiler inlet which is equal to that of the saturated liquid accumulated in the condenser (kJ/kg.K)

The COP is then:

$$\text{COP} = Rm \cdot \frac{h_{g\text{-evap}} - h_{f\text{-con}}}{h_{g\text{-boiler}} - h_{f\text{-con}}} \quad (2.5)$$

Where Rm is the ratio between the mass of the secondary fluid entrained by the ejector and the mass of the primary fluid required by the ejector.

Since the ratio of enthalpy change at the boiler and the evaporator is approximate to be equal to 1 [11]. Thus the COP can be assumed that:

$$\text{COP} \approx Rm \quad (2.6)$$

2.3 Performance characteristic of a steam ejector

Figure 2.6 shows example of a performance curve for a steam ejector. The primary fluid pressure or the boiler saturation temperature was fixed at 130°C. The secondary fluid pressure or the evaporator saturation temperature was fixed at 7.5°C. The ejector's backpressure or the condenser pressure was increased until the ejector was failed to operate.

From the figure, the performance curve can be classified into three regions, choked flow, unchoked flow, and reversed flow [8]. For the *choked flow region*, the ejector is operated with the pressure below the critical value (critical back pressure), it can be seen that the same amount of the secondary fluid is entrained into the mixing chamber. This is

resulted from the secondary fluid flow choking in the mixing chamber. As a result, the entrainment ratio remains constant within this region. In addition, a transverse shock, which creates a compression effect, is thought to appear in either the throat or diffuser section. The location of the shock is dependent on the back pressure. When the back pressure is increased, the shock will move upstream (toward the primary nozzle) without disturbing the mixing process.

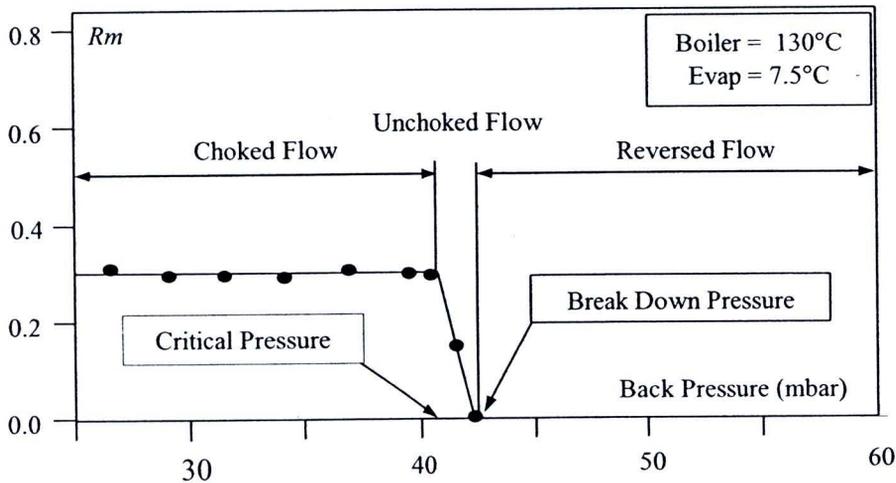


Figure 2.6: Performance characteristic of steam ejector based on experimental data [1].

For the *unchoked flow region*, the ejector is operated with back pressure higher than the critical value. The entrainment ratio drops sharply when the back pressure is increased. The secondary fluid is no longer choked in the mixing chamber. The transverse shock is believed to move upstream into the mixing chamber (converging duct section) and disturbs the mixing process between the primary fluid and the secondary fluid.

For the *reversed flow region*, the ejector is operated with back pressure higher than the point called *break down back pressure*. In this region the mixed stream is reversed back to the secondary flow inlet, therefore the ejector finally malfunctions.

Figure 2.7 shows the effect of the operating pressures on the performance of a steam ejector provided by Aphornratana [7] which was experimentally obtained. When the primary fluid saturation temperature is decreased from 130 to 120°C, the critical mass flow rate of the primary fluid through the primary nozzle will be reduced. As the flow area in the mixing chamber is fixed, more secondary fluid is entrained in to the mixing chamber. As a result, the entrainment ratio of ejector is increased. However, the momentum of the

mixed stream inside the mixing chamber is lower and therefore, the ejector can be operated with a lower critical condenser pressure.

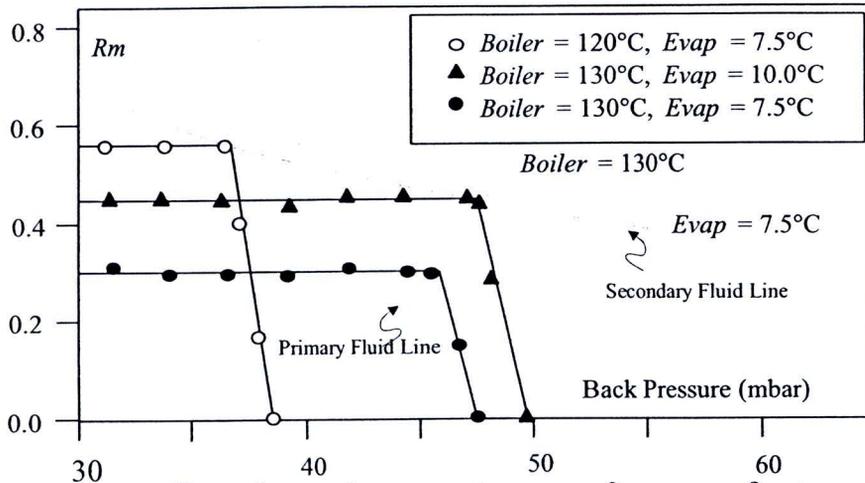


Figure 2.7: Effect of operating pressures on performance of a steam jet refrigerator based on experimental data provided by [7].

When the evaporator saturation temperature is increased from 7.5 to 10°C, the ejector allows more secondary fluid from the evaporator to be entrained into the mixing chamber. This is because the evaporator is the upstream pressure of the ejector. When the evaporator pressure is increased, more secondary fluid is pushed into the mixing chamber. Meanwhile, the critical mass flow rate of primary fluid is fixed (fixed boiler at 130°C). Thus, the entrainment ratio of ejector is increased. Moreover, the momentum of mixed stream is also increased (due to a higher upstream pressure). This results in the ejector being able to operate with a higher critical condenser pressure. It can be concluded that, when the evaporator saturation temperature is increased, both the entrainment ratio and the critical condenser pressure are increased simultaneously, and vice versa.

As described above, it can be seen that the performance of ejector is directly dependent on its operating condition (boiler saturation temperature, evaporator saturation temperature and condenser saturation pressure). However, many past studies [1, 2, 7, 8, 10, 11, and 12] showed that not only operating conditions, but ejector's geometries also affect the COP and critical condenser pressure of a jet refrigeration cycle. Several experimental studies of the effect of primary nozzle throat size and ejector geometry on the system performance were conducted [1, 2, 3, 4, 8, and 12]. It was found that the ejector performance, the cooling capacity, COP, and critical condenser pressure of the jet refrigeration could be varied by changing nozzle throat size and ejector geometry.



2.4 Using CFD analysis with an ejector

Using Keenan's theory alone, a constant capacity phenomenon of the ejector (the ejector entrains the same amount of the secondary fluid over a range of the back pressure) can not be explained. However, this phenomenon can be explained by the effective area concept proposed by Munday and Bagster [9]. They explained that the primary fluid and secondary fluid were not immediately mixed. The secondary fluid was accelerated through *converging duct* which resulted from the flow without the mixing of two fluids. At some cross section along this duct, the velocity of secondary fluid reached to sonic level and choked. The cross section area where secondary fluid choked is called an *effective area* (the two fluids was first mixed). The effective area was located at between supersonic jet core and ejector's wall (annular area between jet core and ejector's wall) as shown in figure 2.8.

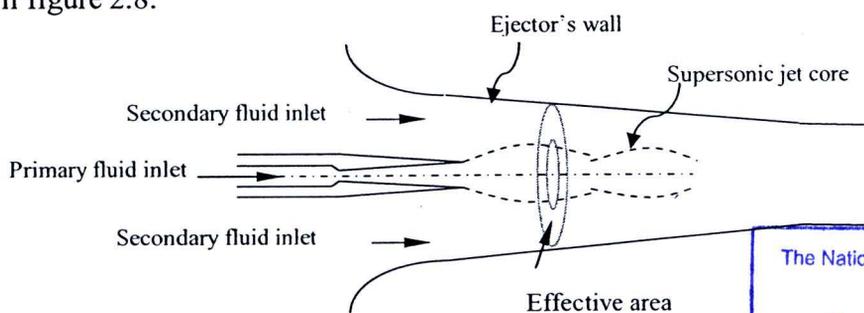
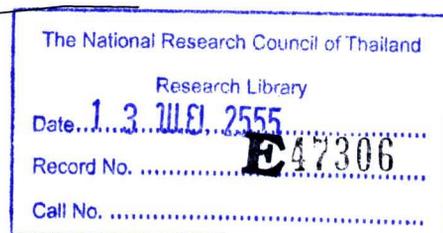


Figure 2.8: The effective area.



From experimental investigations [10, 11], it was found that the classical Keenan 1-D theory could only be used to predict an ejector performance at its design condition only (at critical condenser pressure). Moreover, effects of the ejector's geometries to the performance were not included. Therefore, it may be concluded that the used 1-D theory may not be adequate in order to improve, analyze and design the ejector.

In order to improve the ejector performance, the understanding of the flow behavior and mixing process inside the ejector is necessary. However, by using only the classical 1-D theory, the mixing process inside the ejector may not be clearly explained. Therefore, many researchers have attempted to explain and present the mixing process inside the ejector by using others methods. Matsuo et. al. [23] investigated the mixing process in a rectangular supersonic ejector by using the Schlieren photograph technique. Tamaki et. al. [22] measured the pressure variation along the centerline of an ejector. Their results

indicated that the series of oblique shocks was induced and its static pressure after shock was gradually increased. This was in contrast to the 1-D theory proposed by Keenan of which a single normal shock was induced and pressure recovery after shock was extremely raised. This helps to support the fact that the 1-D theory is inadequate. However, the mixing process was not explained clearly and still remained as imaginary process.

Over the past two decades, computational fluid dynamics (CFD) technique has been developed. During the 90s [1], the CFD technique was first applied to analyze the mixing process of an ejector. This technique was applied to some specific parts of the ejector due to the limitation of the numerical method and the computer technology. Therefore, at that time, the use of CFD technique to analyze the mixing process within an ejector was still incomplete.

Due to the large progress in computer technology and the numerical method, the CFD technique has been rapidly developed. Many commercial CFD software packages can be used with personal computers and are friendly to the users available. It gains popularity in the field of the fluid dynamics problem. Most of the commercial CFD packages have been developed to present the results graphically. This is the distinctive point in order to develop the ejector's performance.

In 1996, Riffat et. al. [13] employed the CFD technique to analyze the ejector's performance with various working fluids and various shapes of the primary nozzles. Three dimensional ejector model was created with a grid size around 37,000 elements. The fluid was assumed to be incompressible in order to avoid convergence difficulty with compressible fluid. This contradicted to the fact that high speed fluid is compressible. Moreover, using the incompressible flow assumption, the effects of shock were discounted.

In 1999, the compressible flow model was begun in order to govern the flow inside the ejector. The turbulence viscosity model, *reliable* $k-\varepsilon$ [3, 4], was used to investigate with 3-D model. This turbulence viscosity model was suitable with supersonic flow field [16]. The result was well in agreement with the experimental result. In addition, the flow inside the ejector was more realistic than that incompressible flow assumption used previous. However, it required lengthening of time for calculation. Sriveerakul et. al. [3, 4] has investigated the 2-D axisymmetric model of steam ejector by using CFD technique. The result showed that, three parameters from CFD technique, entrainment ratio, critical condenser pressure and pressure profiles along the ejector's wall were provided accurately when compared with the experimental result. In addition, the literatures [20] also showed

that the CFD technique could provide accurate results when various working fluids were used.

The advantage of using CFD technique was that the flow inside the ejector can be presented. The past studies [4] revealed that using the graphic filled contour of Mach number, the flow behavior and mixing process inside the ejector could be explained. The filled contour of Mach number showed that the converging duct as proposed by Munday et. al. [9] was found. Meanwhile, the shock was classified as *two-dimensional shock* or *oblique shock* [18] which was similar to that proposed by Matsuo et. al. [23]. In addition, it was also found that the shock occurred at two locations. Firstly, it suddenly occurred at mixing chamber as it leaved the primary nozzle. Secondly, it occurred at throat or subsonic diffuser of ejector. During investigation, it indicated that the location of second shock could be moved by variation's operating-condition and geometry's ejector [4]. It was also found that location of the effective area was difficult to define. It could be located at anywhere within mixing chamber and throat section which depended on operating-condition and geometries of ejector [24, 25 and 26].

2.5 Conclusions

From the literatures, they found that, the performance of jet refrigeration cycle is directly dependent on the equipped ejector. Thus, to improve the COP of cycle, the performance of ejector should be first considered. However, even though, the ejector was invented more than a century ago, very few of past studies concerned with ejector's improvement were found. This is because the researchers emphasized the overall performance of cycle at specified operating condition over the ejector's improvement.

In order to improve the ejector performance, the clear understanding of flow behavior inside the ejector and mixing process are first needed. However, it is very complex and is difficult to propose. Therefore, some researchers have attempted to explain by various methods. However, it was still inadequate in order to clear understanding of the flow inside the ejector. Since, the CFD was applied to the field of ejector's improvement; the flow behavior inside the ejector was then presented, graphically. Thus, a better understanding of the mixing process was proposed. However, there are still some areas where experimental data are insufficient and where CFD techniques are rarely applied.