

# Applications of Jet Ejectors for efficient refrigeration and modelling of Multi Phase Multi Fluid flow in Ejectors

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**Abstract.** The uses of ejector for efficient refrigeration are manifold – it has been used, among other applications, in the VCRS to reduce the compression ratio, in the combined ejector absorption cycle to enhance the refrigeration capacity and in the ejector absorber cycle to obtain lower evaporator pressures, higher absorber pressures and pre-absorption of the refrigerant in the ejector. Hence, modeling of flow which may be two phase two fluid as in ejector absorber cycle or two phase single fluid as in VCRS in an ejector assumes utmost importance. However, much work has not been done in this field. The primary objective of the present work is to discuss about the role of ejectors in various refrigeration systems and to model the two phase two fluid flow in the nozzle and the diffuser of an ejector under suitable assumptions. The equations of conservation of mass, momentum and energy have been solved to find the different flow properties like pressure, temperature and velocity of the two phases as function of the length in the diffuser. Different cases pertaining to different flows have been taken care of by appreciating what type of phenomena can actually occur at the interface of the two phases. Higher pressure rise was obtained for a given diffuser length with higher diffuser angles, smaller droplet diameter, higher inlet velocity of the gaseous phase and higher drag coefficients. Among other results, it was also seen that the two phases reached thermal equilibrium faster with higher diffuser angle, smaller droplet diameter and higher heat transfer coefficient.

**KEYWORDS:** Vapour Absorption Refrigeration Systems, Ejector, Multiphase-flow.

## NOMENCLATURE

COP	Coefficient of Performance
$\eta$	2 <sup>nd</sup> Law efficiency of the system
$\rho$	Density( $Kgm^{-3}$ )
$u, v, w$	Velocites in the x,y and z direction( $ms^{-1}$ )
A	Cross-sectional area( $m^2$ )
$\dot{m}$	Mass flow rate( $Kgs^{-1}$ )
P	Pressure( $Pa$ )

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$F$	Force ( $N$ )
$Q$	Heat ( $KJ$ )
$W$	Work ( $KJ$ )
$H$	Enthalpy ( $KJ$ )
$\mu$	Coefficient of viscosity ( $Kgm^{-1}s^{-1}$ )
$K$	Turbulent Kinetic energy ( $KJ$ )
$\varepsilon$	Turbulent dissipation
Re	Reynold's number
D	Diameter ( $m$ )
$C_D$	Non-dimensional Drag coefficient
$C_{vm}$	Virtual mass coefficient
$\xi$	Concentration ( $Kgm^{-3}$ )
$h$	Convective heat transfer coefficient ( $Wm^{-2}K^{-1}$ )
$h_m$	mass transfer coefficient
$u_r$	Relative velocity ( $ms^{-1}$ )
T	Temperature ( $K$ )
$N_d$	Number of droplets per unit volume ( $m^{-3}$ )
$t$	time ( $s$ )
$f, \alpha$	volume fraction
$\nu$	Kinematic viscosity ( $m^2s^{-1}$ )
$\gamma$	interfacial tension ( $Nm^{-1}$ )
$h_l$	head loss ( $m$ )
$\varepsilon$	roughness of pipe inner surface ( $m$ )
$f$	Darcy Weisbach friction factor
<b>Subscripts</b>	
e	Evaporator
g	Generator/Gaseous phase
a	Absorber
p	Pump
c	Condenser
$\mu_t$	Coefficient of turbulent or eddy viscosity
$l$	Liquid phase

## 1. INTRODUCTION

The importance of conservation of energy in the context of growing global population, increasing per capita power consumption and depleting fossil fuel resources cannot be overemphasized. Energy can be conserved by using it more efficiently which depends not only on the amount of the energy used for the given application but also on the form of the energy. In other words, as far as possible the energy used for a given application should be of correct amount and type. As an example, it would make more sense thermodynamically to spend thermal energy for heating rather than using high grade energy such as electricity. The same holds good for refrigeration and air-conditioning, which can be obtained by thermal energy, whose quality is far inferior to mechanical or electrical energy.

### 1.1 VAPOR ABSORPTION SYSTEMS

Refrigeration by mechanical vapor compression system uses shaft work which is a high grade of energy and therefore expensive. Vapor absorption system however makes use of waste heat which is a low grade of energy and readily available. The amount of work required is also less and so is the maintenance cost owing to very few moving parts. Since the compressor gets hot we have to have means of cooling it and this puts a constraint on the refrigerant we can use. Conventional VCRS unlike VARS use CFCs which are facing opposition owing to high GWP & ODP. Hence, studies on VARS assumes utmost importance and methods to increase their performance has been an area of concern.

In vapor absorption refrigeration system, the function of the compressor is accomplished by a 3 step process by the use of absorber, pump and generator. In absorber, absorption of the refrigerant vapor coming from the evaporator by its weak or poor solution in a suitable absorbent occurs forming a rich solution of the refrigerant in the absorbent. The rich solution is pumped to the generator raising its pressure to the condenser pressure. In the generator or desorber, distillation of the vapor from the rich solution occurs leaving the poor solution for recycling through an expansion valve back to the absorber where it is again ready to absorb fresh refrigerant. A simple vapor absorption system thus consists of a condenser, an expansion device and an evaporator as in the vapor compression system and in addition an absorber, a pump, a generator and a pressure reducing valve to replace the compressor.

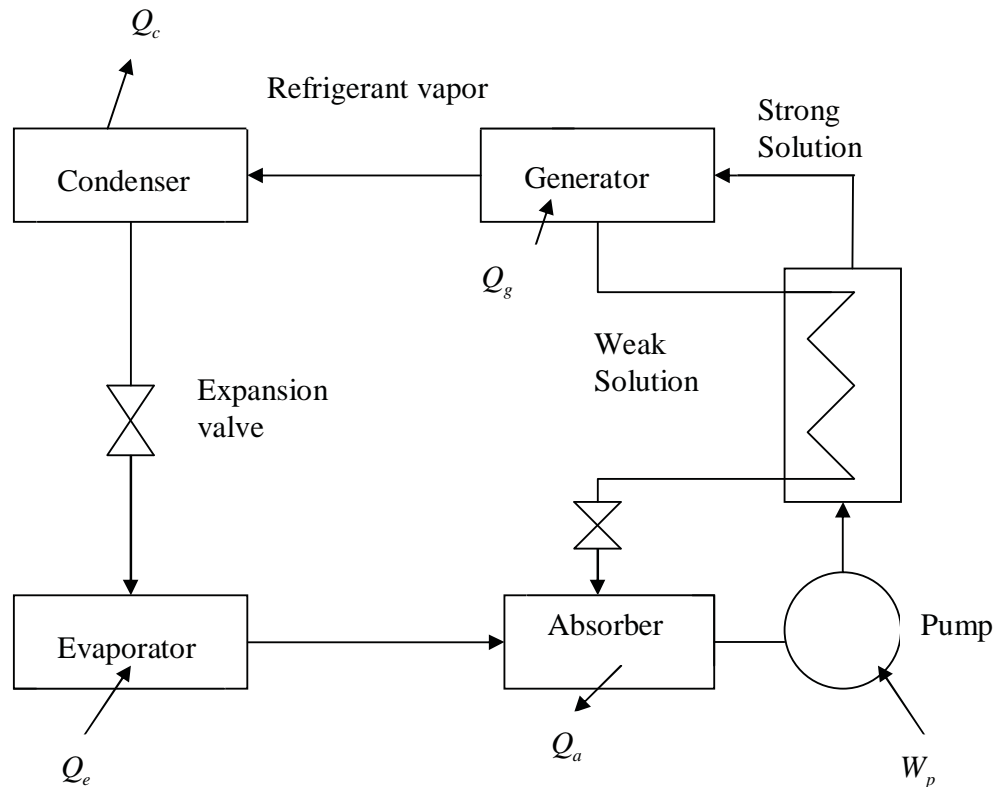


Fig 1 A simple vapor absorption cycle

An ejector is a device where a motive stream expands through a nozzle creating a region of low pressure which entrains the secondary stream which may evaporate in the process thus causing refrigerating effect. The primary objective of the present work is twofold – to discuss the role of ejectors in improving the performance of absorption systems and to model the flow in the nozzle and the diffuser of an ejector and find the different flow properties like pressure, temperature and velocity as function of the length in the diffuser. Different literature surveys have been made in this connection. An attempt to model various one dimensional flows through ejectors like two phase two fluid which is present in different refrigerating systems has been made. For this an one dimensional model has been developed and the different cases pertaining to different flows has been taken care of by appreciating what type of phenomena can actually occur at the interface of the two phases. The equations for conservation of mass, momentum and energy have been solved and flow properties like pressure, temperature and velocity have been plotted as function of length in the diffuser.

## 2. APPLICATIONS OF EJECTORS IN REFRIGERATION SYSTEMS

The applications of ejector and their role in various systems has been discussed in this section.

### 2.1.1 VAPOR COMPRESSION REFRIGERATION SYSTEM

An experimentally validated approach to recover some of the expansion losses of the transcritical carbon dioxide cycle is based on replacing the expansion valve by an ejector as shown below.

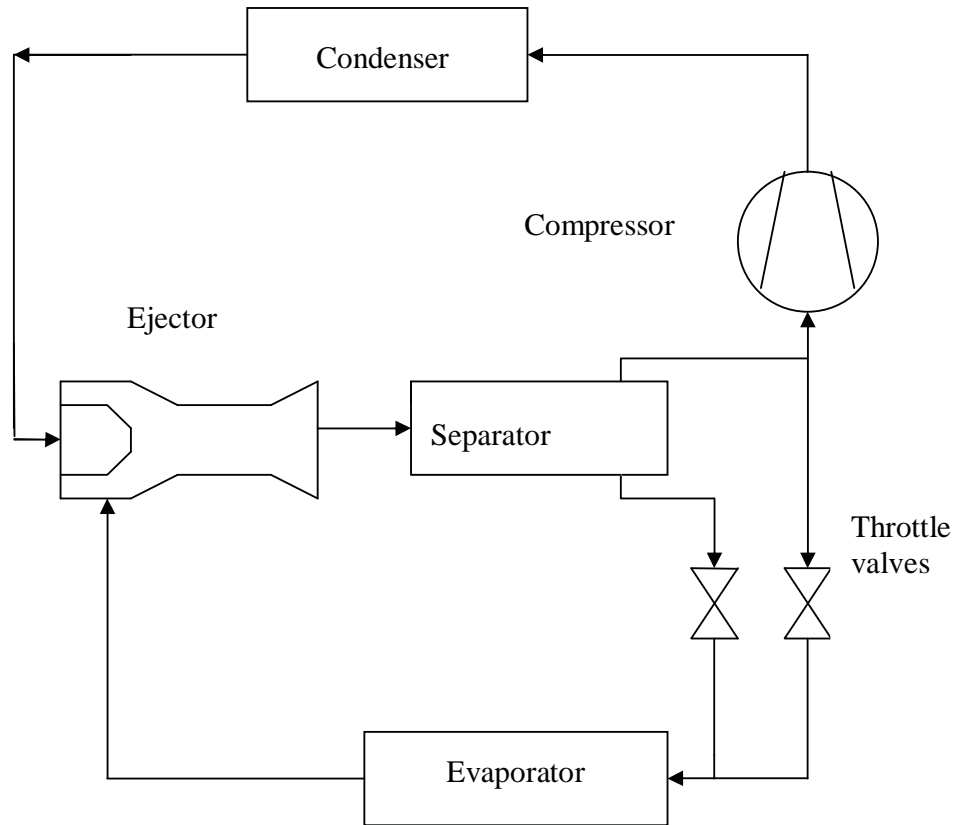


Fig 2 Schematic of trans-critical  $CO_2$  cycle with ejector expansion device

The working processes of the ejector can be explained as follows. The motive stream in the form of high-pressure liquid leaves the condenser and enters the ejector. The motive stream expands in the ejector nozzle from the high pressure to the receiving chamber pressure. The enthalpy of the motive stream reduces and the velocity increases. At the same time the suction stream is drawn from the evaporator at the evaporator pressure and enters the ejector in the receiving chamber where its velocity increases. The suction stream mixes with the motive stream in the mixing section and become a single stream at an intermediate mixing pressure. This stream further increases its pressure in the diffuser by converting its kinetic energy into potential energy. The stream leaves the ejector at a pressure slightly higher than the evaporator pressure and enters a separator to separate saturated liquid from saturated vapor. The saturated liquid expands to the evaporator pressure and enters the evaporator while the saturated vapor enters the compressor. The cycle efficiency increases because the compressor operates at a reduced pressure ratio compared to the cycle without the ejector. In the above system the ejector operates with a 2 phase single fluid.

### 2.1.2 COMBINED EJECTOR ABSORPTION CYCLE

Eames et al [1] investigated an ejector-absorption cycle both theoretically and experimentally. This cycle shown in the figure below consists of 2 sub-cycles: the steam ejector cycle and LiBr-water single-effect absorption cycle. The ejector cycle is driven by water vapor from the generator of the absorption cycle.

Because the steam ejector utilized the energy, otherwise lost in a conventional absorption cycle, to enhance the vaporization process in this novel cycle, a higher COP was expected. The computer simulation of this novel cycle was reported by Sun et al[2]. The experimentally-measured COP of this cycle was reported in the range between 0.8 to 1.04 for  $5^{\circ}\text{C}$  cooling temperature [3]. However, this cycle requires a generator temperature of at least  $200^{\circ}\text{C}$  and this may result in increased corrosion rates, which may be problematic[4]. In this system the ejector operates with a single phase single fluid.

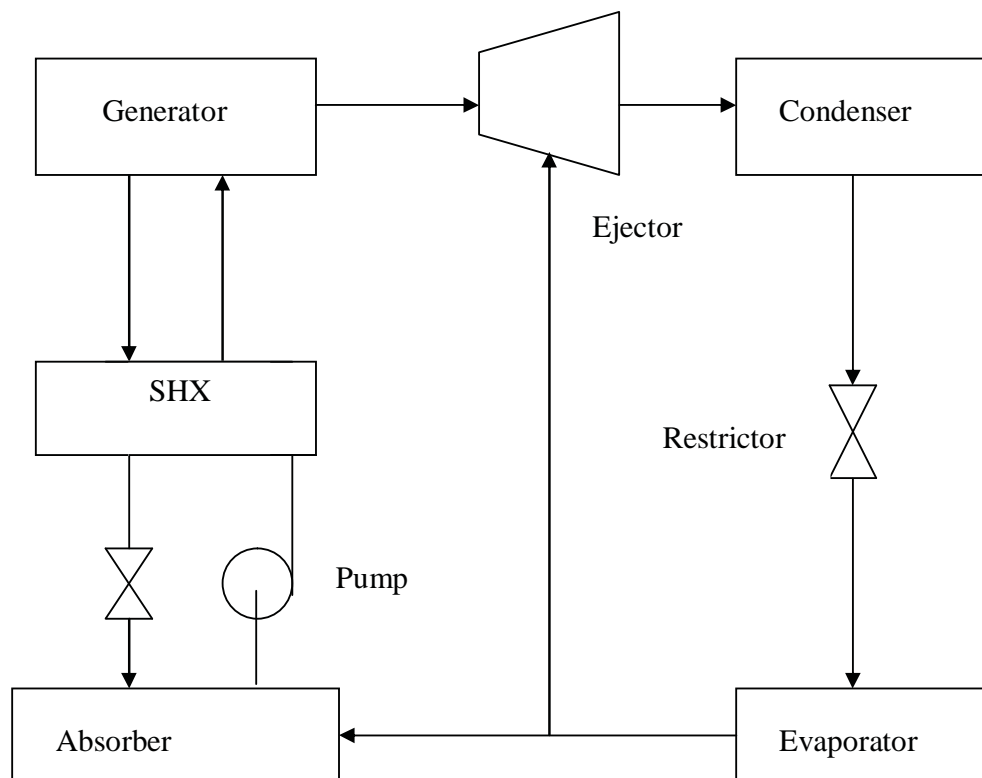


Fig 3 Combined ejector-absorption cycle

### 2.1.3 EJECTOR ABSORBER CYCLE

Chung et al[5] and Chen [6] used the returning solution from the generator as the primary fluid to entrain the refrigerant vapor from the evaporator. The mixture fluid was discharged into the absorber. Since this arrangement allowed the absorber pressure to be higher than the evaporator pressure, the circulation ratio of the solution could be reduced. Therefore, the COP was improved. However, this system can only be operated using high density refrigerant vapor, because a liquid driven ejector is not suitable for low density vapor such as water vapor (which would involve huge volumetric flow rates). The flow in the ejector is that of 2 phase 2 fluid.

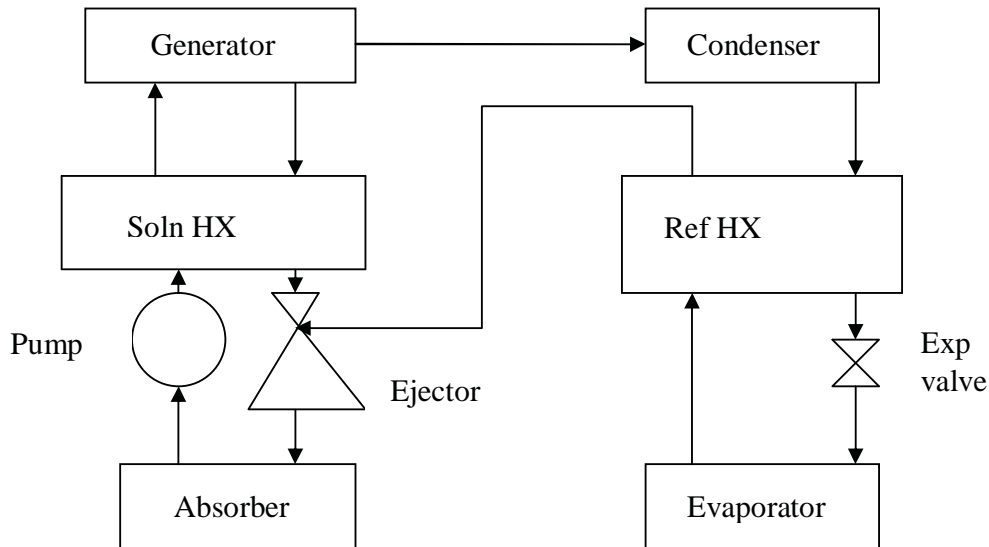


Fig 4 An ejector absorber cycle

The above cycle has the following advantages:

- (i) For a given absorber pressure it is possible to achieve a lower value of evaporator pressure, which would enable lower temperatures.
- (ii) For a given evaporator pressure we can have a higher value of absorber pressure and hence a higher temperature and hence the absorber can reject heat to a hotter sink.
- (iii) There is pre-absorption of the refrigerant in the ejector.

## 2.2 CONCLUSIONS FROM LITERATURE SURVEY

The application and the use of ejector in different systems has been looked into. In vapor compression refrigerating systems the introduction of the ejector reduces the work of compression by reducing the compression ratio. In combined ejector absorption cycle, the ejector by sucking vapor from the evaporator increases the refrigerating capacity subject to the condition that the generator has to be operated at a higher temperature. In ejector absorber cycle, the introduction of the ejector enables us to operate the absorber at a higher pressure for a given evaporator pressure which implies that the absorber can reject heat to a hotter sink. It also makes the evaporator work at a lower pressure for a given absorber pressure which means we can get lower temperatures. Besides, the ejector also improves pre-absorption which implies that the size of the absorber is reduced.

A generalized model for 3 dimensional flow in ejectors has been looked into. A model in which the liquid phase has turbulent flow has also been studied. The book on “1 dimensional 2 phase Flow” by Wallis gives different modeling of flows namely the homogeneous model, the separated flow model, the drift-flux model which is applicable to flow of different fluids under different operating conditions. From the studies it is apparent that not much work has been done in the field of two phase flows. Hence, there is large scope for work in this field which is relevant considering that there is so much application of such flows in different systems.

## 3. MODELING OF 1-D TWO PHASE FLOW IN AN EJECTOR

Levy et al (9) models two phase two fluid flow in ejector which is present in vapor absorption systems as discussed in the literature survey under a set of assumptions given below.

### 3.1 DESCRIPTION

The gaseous phase is the pure refrigerant which comes from the evaporator after passing through the refrigerant heat exchanger. The dispersed phase is the solution which comes from the generator after passing through the solution heat exchanger and is injected with the help of a nozzle into the diverging section. The two phases pass through the diverging section of the ejector whose other end is connected to the absorber. The flow in the diverging section assumed 1 dimensional has been modeled in this section.

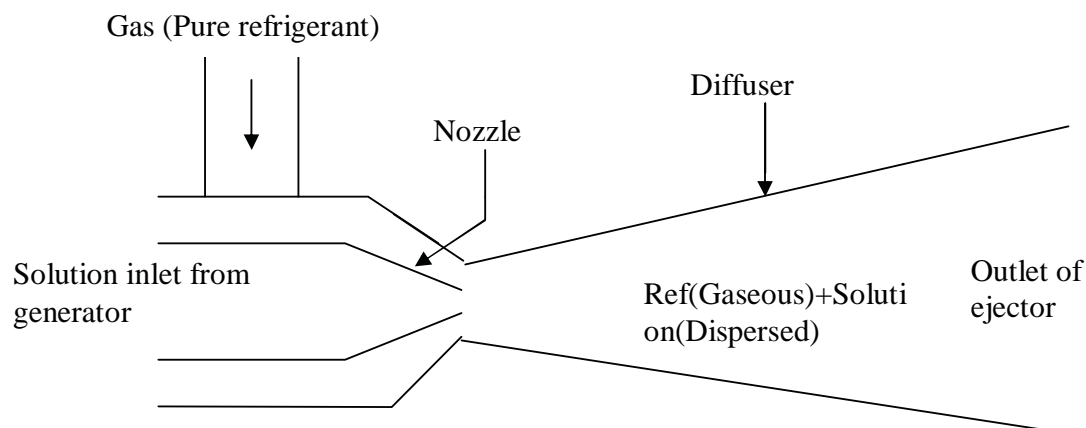


Fig 5 Schematic of the ejector

### 3.2 ASSUMPTIONS

- (i) Steady state.
- (ii) One Dimensional flow with  $u=u(x)$ .
- (iii) Pressure  $P=P(x)$  which is reasonably good assumption as flow is in X-direction.
- (iv) Properties at a particular section are constant.
- (v) Mass, momentum and heat transfer only between the 2 phases which implies adiabatic diffuser and frictional work with walls of diffuser negligible.
- (vi) Dilute phase flow
- (vii) Droplets consist of a binary mixture of absorbent and refrigerant
- (viii) Constant droplet diameter
- (ix) Electrical forces and surface tension neglected.
- (x) All relevant functions are continuous.
- (xi) The dispersed phase is fully dispersed i.e. there is no coagulation of any sort.
- (xii) The two phases share the same pressure field i.e. at a particular section  $P_g = P_d = P$ .

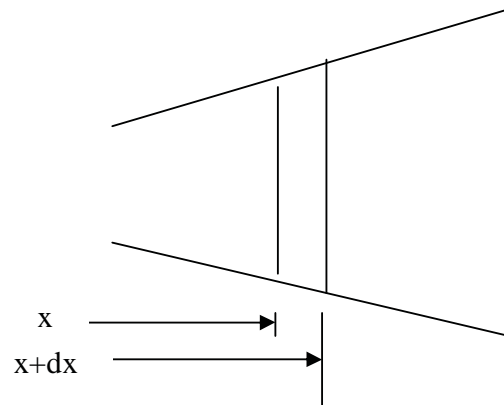
### 3.3 THE MODELLING OF 2 PHASE FLOW IN THE DIVERGING SECTION ASSUMING ABSORPTION

We apply the Reynold's Transport theorem to a control volume of length  $dx$  and cross sectional area same as that of the diffuser at that particular section. We can apply Reynold's Transport theorem to both the dispersed and liquid phase individually (separated flow model) or as a whole to the entire fluid flowing through the diffuser (homogeneous model) which obviously depends on the conditions prevalent in the



diffuser. But it should be seen that the number of differential equations obtained by the conservation laws equal the number of unknowns.

### CONSERVATION OF MASS



Applied to the gaseous phase,  $(\rho_g u_g A_g)|_x - (\rho_g u_g A_g)|_{x+dx} = d\dot{m}_{gd}$  where  $\dot{m}_{gd}$  is the rate of absorption of gaseous phase to dispersed phase which is assumed positive i.e. mass is absorbed into the dispersed phase. Here  $A_d, A_g$  and  $A_{tot}$  are the areas of cross-section occupied by the dispersed phase, gaseous phase and total diffusion cross-sectional area respectively.

$$\frac{d(\rho_g u_g A_g)}{dx} = -\frac{d\dot{m}_{gd}}{dx} \dots\dots\dots(3.1)$$

Applied to the dispersed phase,  $(\rho_d u_d A_d)|_x - (\rho_d u_d A_d)|_{x+dx} = -d\dot{m}_{gd}$

$$\frac{d(\rho_d u_d A_d)}{dx} = \frac{d\dot{m}_{gd}}{dx} \dots\dots\dots(3.2)$$

### CONSERVATION OF MOMENTUM

Applied to the gaseous phase,

$\rho_g u_g^2 A_g|_{x+dx} - \rho_g u_g^2 A_g|_x = dF_{gd} - A_g P|_{x+dx} + A_g P|_x - d\dot{m}_{gd} u_d$  where  $F_{gd}$  is the force on the gaseous phase by the dispersed phase which is a surface force.

$$\frac{d(\rho_g u_g^2 A_g)}{dx} = \frac{dF_{gd}}{dx} - \frac{d(A_g P)}{dx} - \frac{d\dot{m}_{gd}}{dx} u_d \dots\dots\dots(3.3)$$

Applied to the dispersed phase,

$$\rho_d u_d^2 A_d|_{x+dx} - \rho_d u_d^2 A_d|_x = -dF_{gd} - A_d P|_{x+dx} + A_d P|_x + d\dot{m}_{gd} u_d$$

$$\frac{d(\rho_d u_d^2 A_d)}{dx} = -\frac{dF_{gd}}{dx} - \frac{d(A_d P)}{dx} + \frac{d\dot{m}_{gd}}{dx} u_d \dots\dots\dots(3.4)$$

where the last term of the above 2 equations implies that the momentum that is lost by the gaseous phase is gained by the dispersed phase.

**CONSERVATION OF ENERGY**

Applying it for gaseous phase,

$$\left\{ \rho_g u_g A_g \left( h_g + \frac{u_g^2}{2} \right) \right\} \Big|_x + dQ_{gd} - dW_{gd} + d\dot{m}_{gd} \left\{ h_{gd} + \frac{u_d^2}{2} \right\} = \left\{ \rho_g u_g A_g \left( h_g + \frac{u_g^2}{2} \right) \right\} \Big|_{x+dx}$$

$$\Rightarrow + \frac{dQ_{gd}}{dx} - \frac{dW_{gd}}{dx} + \frac{d\dot{m}_{gd}}{dx} \left\{ h_{gd} + \frac{u_d^2}{2} \right\} = \frac{d}{dx} \left\{ \rho_g u_g A_g \left( h_g + \frac{u_g^2}{2} \right) \right\} \dots\dots\dots(3.5)$$

where  $h_{gd}$  is the enthalpy of the refrigerant at the droplet surface and the last term on the L.H.S denotes the energy transfer from the gaseous to dispersed phase due to absorption.

Applying it for dispersed phase,

$$\left\{ \rho_d u_d A_d \left( h_d + \frac{u_d^2}{2} \right) \right\} \Big|_x - dQ_{gd} + dW_{gd} - d\dot{m}_{gd} \left\{ h_{gd} + \frac{u_d^2}{2} \right\} = \left\{ \rho_d u_d A_d \left( h_d + \frac{u_d^2}{2} \right) \right\} \Big|_{x+dx}$$

$$- \frac{dQ_{gd}}{dx} + \frac{dW_{gd}}{dx} - \frac{d\dot{m}_{gd}}{dx} \left\{ h_{gd} + \frac{u_d^2}{2} \right\} = \frac{d}{dx} \left\{ \rho_d u_d A_d \left( h_d + \frac{u_d^2}{2} \right) \right\} \dots\dots\dots(3.6)$$

The last term on the L.H.S of the above two equations denotes that the energy lost by the gaseous phase due to absorption is gained by the dispersed phase. Since we assume the ejector to be adiabatic, we have only internal heat exchange. We will use different empirical relations relating heat transfer, work transfer and force to other properties of the gas or solution and other unknowns like

transfer, work transfer and force to other properties of the gas or solution and velocities so that we have same number unknowns as the number of equations.

**THE MODELLING OF 2 PHASE FLOW IN THE DIVERGING SECTION ASSUMING NO ABSORPTION e.g. AIR-WATER**

This can be easily obtained from the previous set of equations by putting  $\dot{m}_{gd}=0$  i.e. the masses of the dispersed and gaseous phase throughout the entire diffuser remain constant. Several relations to close the system of equations have been developed. These include equations for number of droplets per unit volume, interface forces including forces due to drag and forces due to virtual mass, work transfer at the interface, heat transfer at the interface and mass transfer. The correlation for estimating the overall mass transfer was obtained from an approximation of the theoretical solution of Kronig and Brink for one drop which fits the experimental results of Garner for a cloud of drops (*Clift et al (1978)*)

**4. RESULTS AND DISCUSSION**

A sample set of input includes the following: velocity of air at inlet to diffuser = 45  $ms^{-1}$ ; velocity of water at inlet to diffuser = 60  $ms^{-1}$ ; mass flow rate of water = 24  $Kgs^{-1}$ ; density of water = 1000  $Kgm^{-3}$ ; density of air = 1  $Kgm^{-3}$ ; non-dimensional drag coefficient = .44; convective heat transfer coefficient = 250  $Wm^{-2}K^{-1}$ ; diameter of droplets = 0.0002 m; specific heat of water = 4180  $JKg^{-1}K^{-1}$ ;

specific heat of air =  $1005 \text{ J Kg}^{-1} \text{ K}^{-1}$ ; radius of diffuser at exit=0.2 m; radius of diffuser at inlet=0.1295 m; length of the diffuser=.4 m

Equations 3.7 to 3.12 for air as the continuous phase and water as the dispersed phase have been solved with the above input data using MATLAB 6.5 and the results are shown.

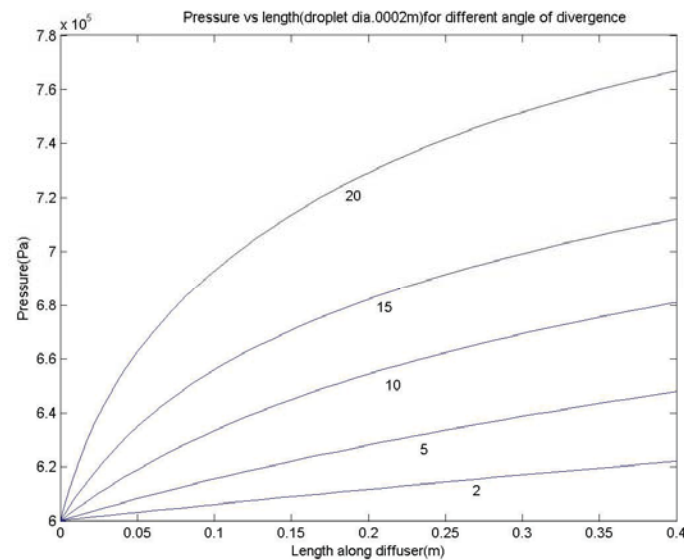


Figure 6 Pressure vs length (droplet diameter=.0002m)for different diffuser angles

From the above plot(Fig 6), it is seen that higher pressure recovery was obtained for higher diffuser angles over a given diffuser length. Also, the rise in pressure progressively becomes more and more as the diffuser angle is increased. Beyond  $15^\circ$ , the diffuser pressure rises suddenly to high values and problems of flow separation may occur with consequent loss of energy. For this purpose, diffuser angles are generally kept much below this value. Apart from same conclusion which can be reached by the following plot (Fig 7), the later shows that for lesser droplet diameters we have higher pressure rise.

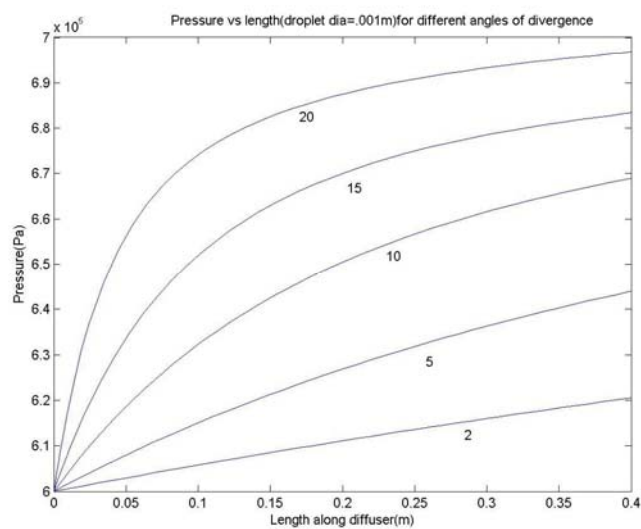


Figure 7 Pressure vs length (droplet diameter=.001m) for different diffuser angles

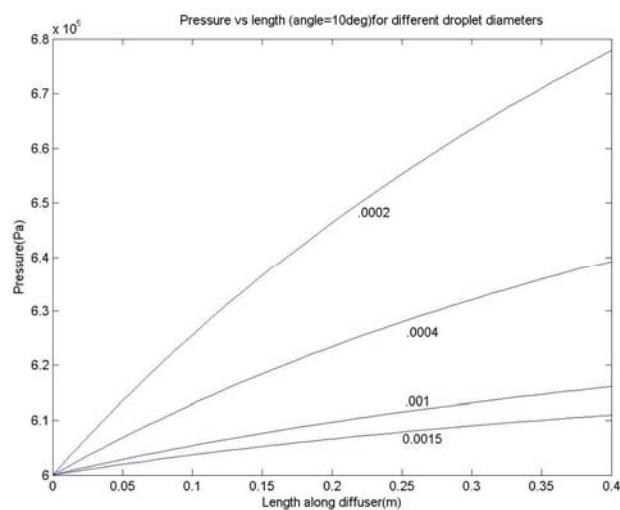


Figure 8 Pressure vs length (angle=10°) for different droplet diameters

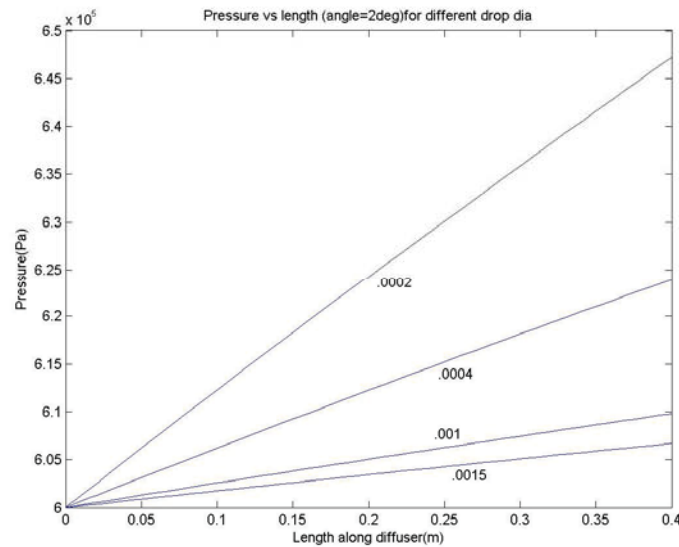


Figure 9 Pressure vs length (angle= 2° )for different droplet diameters

The plot in the previous page (Fig 8) as well as the plot (Fig 9) above shows that a higher pressure rise was obtained when the droplet diameter was reduced. This can be explained from the fact that when the droplet diameter was reduced a higher total surface area was obtained for the same volume and hence there was larger momentum transfer area.

Fig 10 shows that a higher value of pressure is reached if the inlet velocity of the gaseous phase is higher. This is obvious from the fact that with a higher velocity there is a higher velocity head which can be converted to pressure head.

Fig 11 shows that a higher value of pressure is reached if the drag coefficient between the two phases is higher. With more drag there will be more momentum transfer resulting in a higher exit pressure.

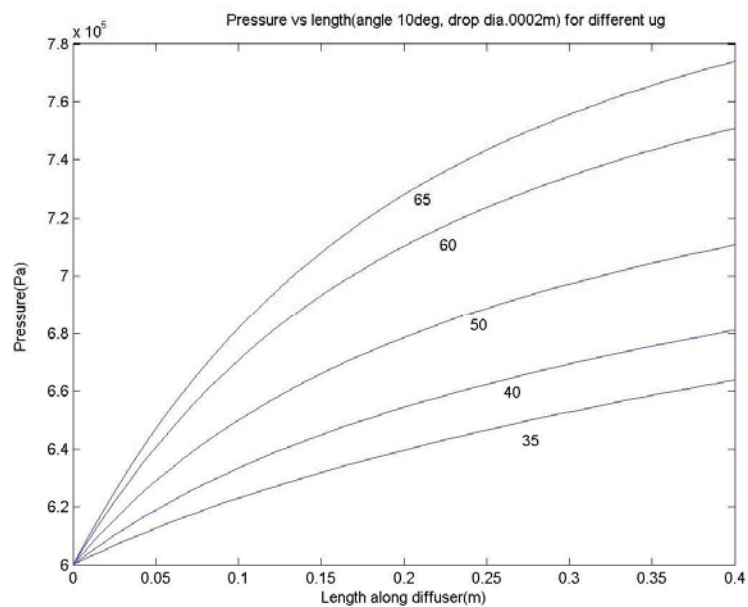


Figure 10 Pressure vs length for different inlet velocities ( $ms^{-1}$ ) of the gaseous phase.

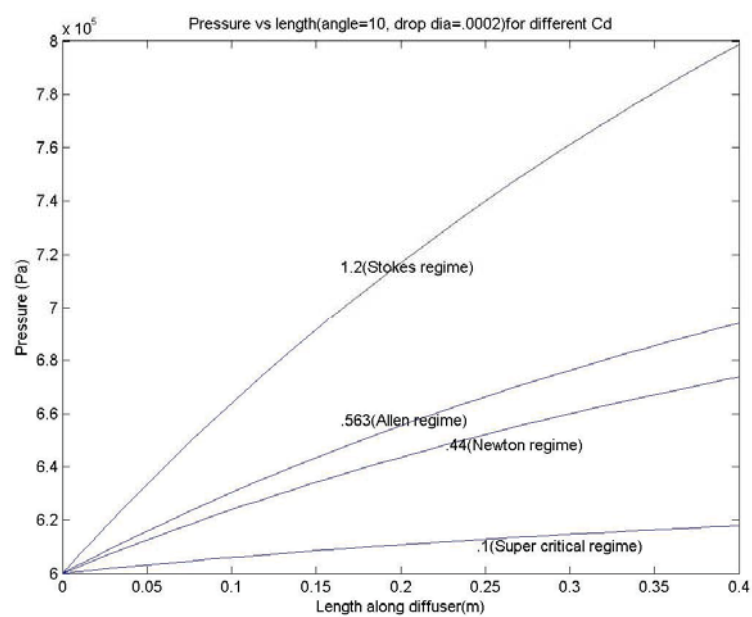


Figure 11 Pressure vs length for different drag coefficients

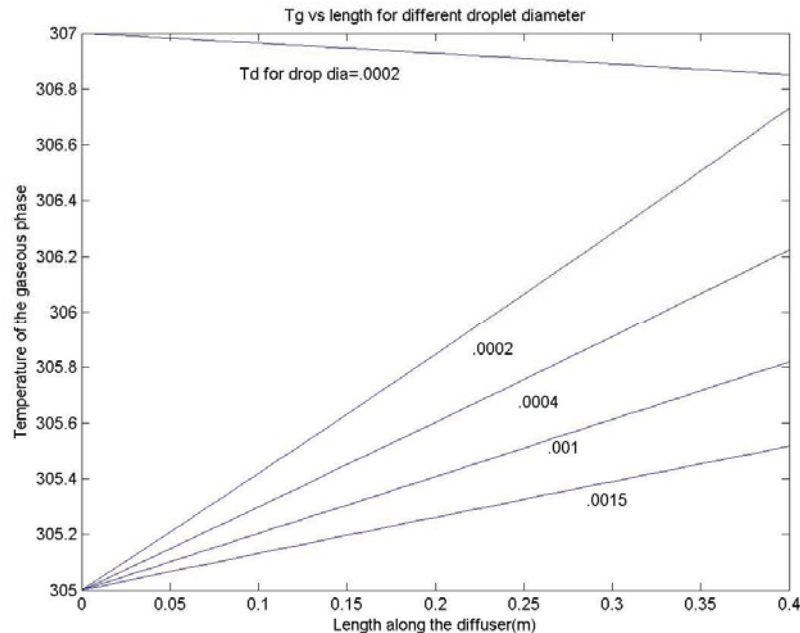


Figure 12 Temperature vs length for different droplet diameters

From Fig 12, it is seen that the temperature rise of the gaseous phase is more than the temperature fall in the dispersed phase. This is because the specific heat of the dispersed phase (water) is about four times more than that of the gaseous phase (air). Also as the droplet diameter is decreased the temperature of the gaseous phase increases implying more heat transfer which is obvious from the increased surface area for heat transfer available due to reduced droplet diameter. Also as the droplet diameter is reduced there is a tendency of the two phases to reach thermal equilibrium over a smaller length.

Figures 13 and 14 show that the tendency of the two phases to reach thermal equilibrium is higher for larger diffuser angles.

Figure 15 shows that the temperature rise of the gaseous phase and hence the tendency to reach thermal equilibrium is more for larger values of convective heat transfer coefficient.

Figure 16 shows that the velocity of the gaseous phase fell faster as the diffuser angle grew.

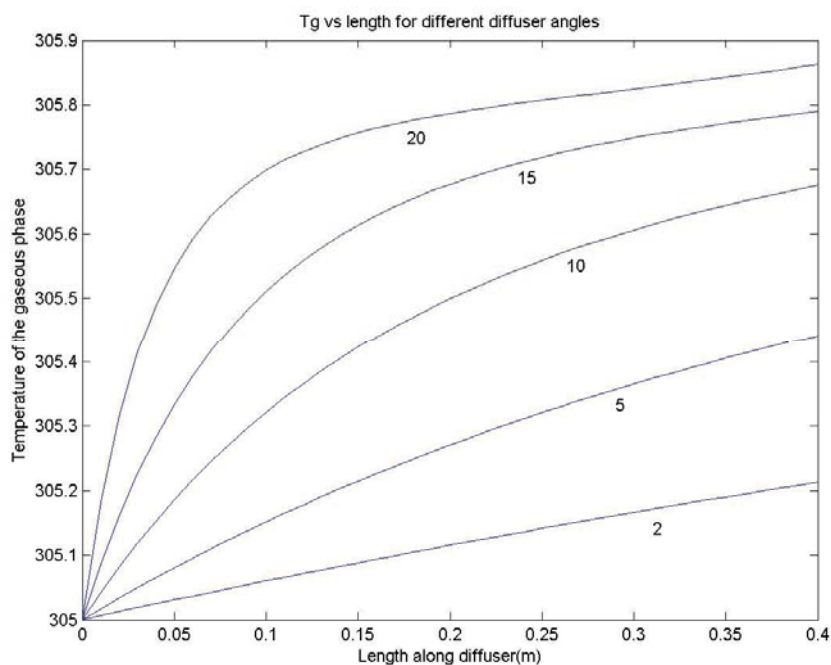


Figure 13 Temperature of the gaseous phase vs length for different diffuser angles

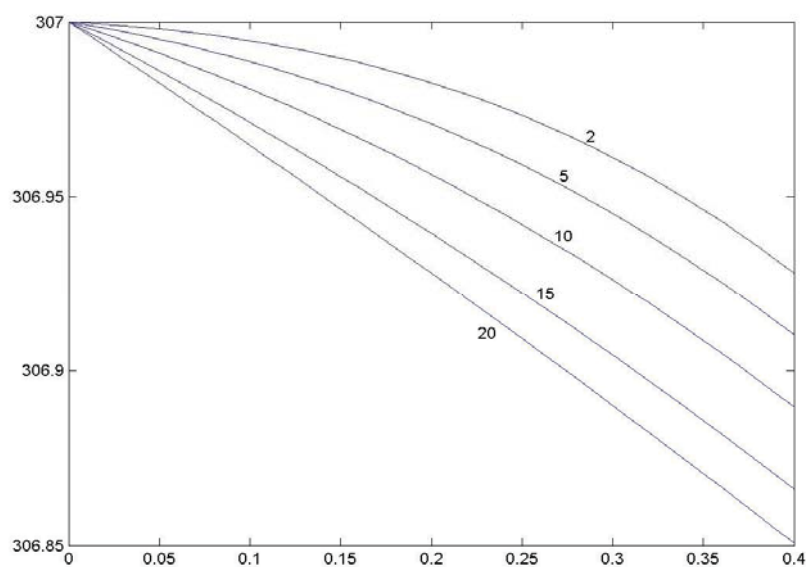


Figure 14 Temperature of the dispersed phase vs length for diff diffuser angles



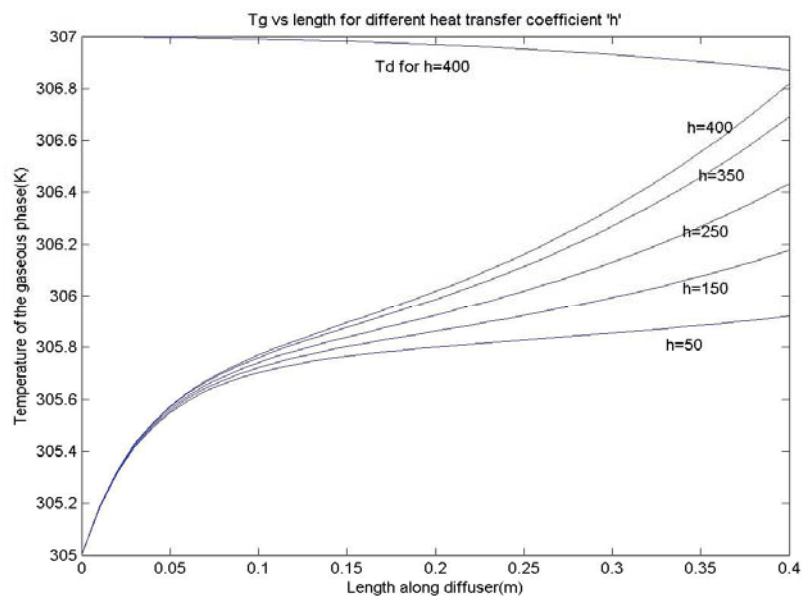


Figure 15 Temperature of the gaseous phase vs length for different heat transfer coefficient

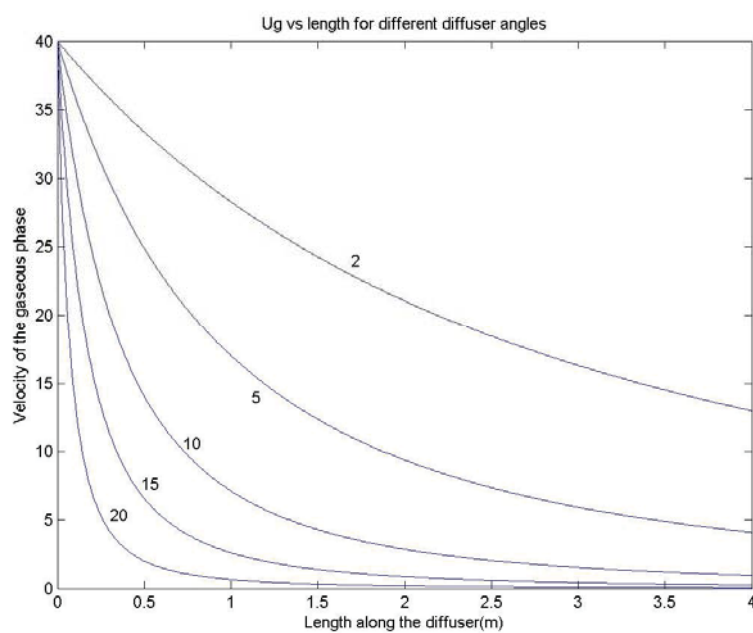


Figure 16 Velocity of the gaseous phase vs length for different diffuser angles

## 5. CONCLUSION

### 5.1 SUMMING UP

The results of the one dimensional flow model developed and solved using air water mixture are quite interesting. The trends of the basic model may be used to design diffuser for actual refrigerant absorber pairs. It is seen that the pressure rise is more for larger diffuser angles but beyond a certain value this may not be advisable for design purposes owing to problems of flow separation and consequent loss in energy. A larger pressure recovery is also observed for smaller droplet diameter and hence the nozzle has to be suitably designed to produce droplets of the required diameter depending on the required pressure at the diffuser exit. The pressure rise is also large if the drag coefficient is more and hence suitable substances may be added to the solution to increase the drag coefficient. A higher pressure rise is also noted for larger inlet velocities of the gaseous phase. It is seen that the temperature rise of the gaseous phase was higher than the temperature fall of the dispersed phase and the two phases tend to reach thermal equilibrium faster for smaller droplet diameters owing to larger surface area for heat transfer. The temperature rise is also more for larger diffuser angles and higher values of heat transfer coefficient. The fall in velocity is also faster for wider diffuser angle.

### 5.2 FUTURE SCOPE OF WORK

This work can be extended to practical refrigerant absorbent pairs and using the model developed, it can be found which pair is suitable for which application. For this the general one dimensional model developed assuming mass absorption can be used and studies have to be made for getting the appropriate constitutive relations at the interface and the properties of the respective mixture and the refrigerant. We can also develop a two dimensional model.

Though we have focused our attention on the flow in the diffuser part of ejector only, we can also analyze how the ejector improves system performance by comparing the COP with and without the ejector. The results of the present study can be used to predict the performance of complex systems having ejectors as one of the components.

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