Sudan University for Science and Technology
Faculty of Engineering

College of Post Graduate
Mechanical Engineering (power)

Simulation of Ejector Flow Behavior Which Produce Vacuum in Power Plants Condenser

(Case Study Kosti Thermal power Station)

نمذجة سلوك سريان نازعة منتجة للفراغ في مكثفات محطات القدرة

(دراسة حالة محطة كوستي الحرارية)

Thesis submitted in partial fulfilment of requirement for the degree of M.Sc.

in Mechanical Engineering (Power)

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<u>الآية الكريمة</u>

بسم الله الرحمن الرحيم

فال تعالى :

((يا ايها الذين آمنوا اذا قبل لكم تفسحوا في المجالس فافسحوا يفسح الله لكم واذا قيل انشزوا فانشزوا يرفع الله الذين آمنوا منكم والذين أتوا العلم درجات)) المجادلة الآَية 11

وقال ايضاً :

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الى الاصدقاء والاحباب

الى المعلمين والمعلمات في كافة المراحل التعليمية

اهدي اليكم هذا العمل المتواضع عسى ان ينال رضاكم

Abstract

The ejector is a device that been used to release the vapor and gases, or used to suctioned fluid from place to another been pumped to.

Power plants condenser need this equipment to release gases and uncondensate steam, the ejector available in Kosti Thermal Power Station (KTPS) is not sufficient for produce the required vacuum for releasing gases and vapor.

In this research the ejector was studied and some modifications to it like change is done the in throat length and divergent length, also motive steam pressure also was changed.

The results were obtained by using Computational Fluid Dynamics analysis software (ANSYS) was found acceptable. The ratio between throat length, divergent length and motive steam pressure found to effects on suction pressure and performance of the ejector.

المستخلص

النازعة هي المعدة التي بواسطتها يتم التخلص من الغازات و الابخرة او يمكن استخدامها في سحب موائع من والى المنطقة المراد ضخها فيه . مكثفات محطات القدرة تحتاج لمثل هذه المعدة لتخليصها من الغازات والابخرة الغير مكثفة , فنازعة البخار الواحدة المستخدمة في محطة توليد كهرباء كوستي غير قادرة على انتاج الضغط الكافي لتفريغ المكثف وتخليصه من هذه الغازات والابخرة . في هذا البحث تم دراسة النازعة وعمل بعض التعديلات عليها كتغييرطول الخانق وطول الناشرة وكذلك تم تغيير قيمة ضغط البخار الداخل الى النازعة وتم الحصول على نتائج مقبولة عن طريق عمل نمذجة بواسطة برنامج محاكاة سريان الموائع (الأنسيس) .

وجد ان نسبة طول الخانق الى طول الناشرة (divergent) والضغط الداخل يؤثران على ضغط السحب وأداء النازعة.

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List of Symbols :

Symbols	Title
LPT	low pressure turbine
KTPS	kosti thermal power station
STOL	short take-off and landing
VTOL	vertical take-off and landing
CFD	Computational Fluid Dynamics
S	distance from nozzle outlet to discharge
L	throat length
R	divergence length
D_n	diameter of nozzle
D_t	diameter of ejector throat
D_O	diameter of divergence
D_P	diameter of convergence
P_d	discharge pressure
P_s	suction pressure
M_P	primary mach number
M_S	secondary mach number
α	convergence angle
θ	divergence angle
p_m	motive steam pressure
p_s	suction pressure
m_a	mass of air
m_{st}	mass of steam
d_t	diameter of nozzle throat

CHAPTER I

1-1 Background:

Ejector devices are distinguished by their simple mechanical construction derived from a simple working principle which consist in transferring momentum from one filed to another by fluid shear in mixing process. This apparent simplicity, however, does not extend to the process analyses of the devices [1].

This research reports a part investigation of ejector systems in which used two fluids as working medium, the primary fluid is steam (with high pressure) and the secondary fluid is air (which is suctioned by steam).

Ejectors has been widely used in different cycles for refrigeration purposes and power plants. Ejector is expansion device, simple to construct and provide robust operation without moving parts while still yielding significant performance improvements [2] .

Significant simulation analyses have been devoted to understanding the variety of flow regimes present in steady flow within the ejector and then to improve performance of ejector, in another words best values of vacuum.

Steam Jet Ejectors are based on the ejector-venture principle. In operation, steam issuing through an expanding nozzle has its pressure energy converted to velocity energy, a vacuum is created, air or gas is entrained and the mixture of gas and steam enters the venture diffuser where its velocity energy is converted into pressure sufficient to discharge against a predetermined back pressure.

Also we have a jet pump as a simple device applied widely in the fields of civil engineering to dewater foundation excavations in fine soils and dredging. It is also used in several mechanical, chemical, and industrial engineering applications for evacuating gases, lifting of liquids, and solid particles.

The principle of the jet pump is to convert the pressure energy of the motive (primary) fluid into the velocity energy through driving nozzle. The resultant jet of high velocity creates a low pressure area in the suction chamber causing the pumped (secondary) fluid to flow into this chamber. Consequently, there is an exchange of momentum between the two streams in the mixing chamber resulting in a uniformly mixed stream traveling at an intermediate velocity between the motive and pumped fluid velocities. The diffuser is shaped to convert the kinetic energy of the mixture to pressure rise at the discharge flange with a minimum energy loss. The absence of moving mechanical parts eliminates the operational problems associated with bearings seals and lubrication. Therefore, such pumps are widely used because of their simplicity and high reliability (as a consequence of no moving parts), in this thesis the ejector will be studied as equipment for creating vacuum [3].

The condensate system of thermal power plants need device called ejector to provide a negative pressure to let all incondensable steam flow through low pressure turbine (LPT) and the other to be condensed (to be in water phase) in a condenser by reject its heat using cooling water coming from cooling towers or rivers.

1-2 Importance of the study:

To select a satisfy ejector which can be use for evacuation the uncondensed steam in condenser**,** or provide an enough vacuum on condenser.

1-3 Problem Statement:

The ejector available in Kosti Thermal Power Station (KTPS) is not sufficient for vacuum required except two ejectors working together (no standby), so a new one should be selected to provide required specification alone (other one in standby condition) to help the system in some cases like maintenance.

1-4 Research Objectives:

i. To obtain the specification of ejector needed.

ii. To obtain motive steam pressure needed.

iii. To obtain the best vacuum.

1-5 Methodology:

By using computer simulation software the flow parameters (velocity and pressure) would be studied to obtain the optimum value of vacuum when changing the motive steam pressure and dimensions of ejector (divergent length and throat length) .

1-6 Thesis layout:

The thesis consist of five chapters, chapter one is introduction containing the background, problem statement and methodology. While chapter two illustrate the theory and literature review, chapter three discussed the methodology of the thesis, when chapter four operates the calculation and show results of study. At last chapter five abstract the conclusions and afford some recommendation.

CHAPTER II

2-1 Background:

Electricity is the only form of energy which is easy to produce, easy to transport, easy to use and easy to control. Electricity in bulk quantities is produce in power plants, which can be of the following types:

i. Thermal power plants.
ii. Nuclear power plant.
iii. Hydraulic power plant.
iv. Geothermal power plant.

Thermal, nuclear and geothermal forms use steam as the working fluid and have many similarities in their cycle and structure. Gas turbine as the type of thermal plants are often used as peaking units. They run for short period in a day to meet the peak load demand. Hydraulic power plants are essentially multipurpose. Besides generating power, they also cater for irrigation, flood control, fisheries, afforestation, and navigation. They are however, expensive and take more time to build (high initial cost). Geothermal power plants can be built only in certain geographical locations.

2-2 Thermal Power Plants:

The basic components of the thermal power plants cycles which are shown in Figure (2-1) are:

• Boiler or Steam Generator

• Turbine

• Condenser

• Pump

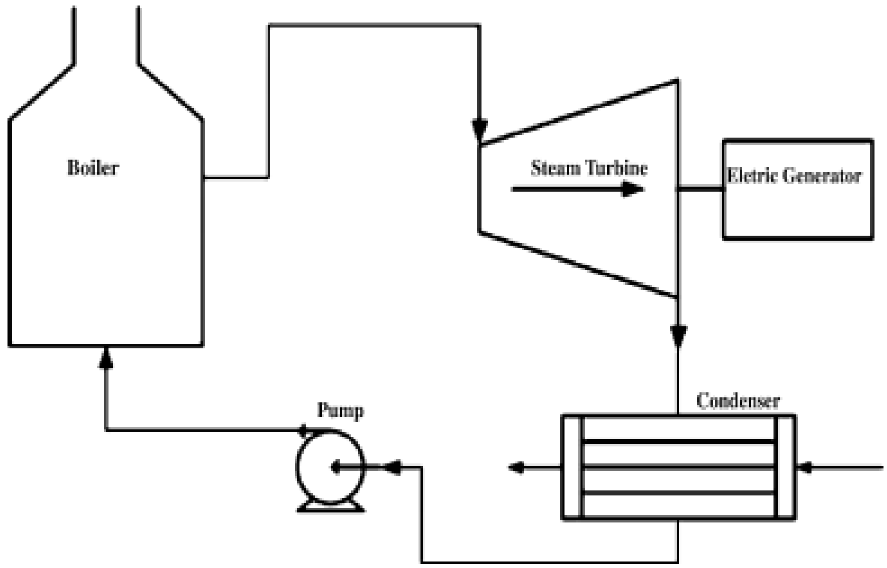

Figure (2-1): Components of Thermal Power Plant

A steam power plant continuously converts the energy stored in the fossil fuels (coal, oil, natural gas) or fissile fuels (uranium, thorium) into shaft work and ultimately into electricity. The working fluid is water which is sometimes in the liquid phase and sometimes in the vapour phase during its cycle of operation. The Figure (2-2) illustrates a fossil fuelled power plant as a bulk energy converter from fuel to electricity using water as working medium. Energy released by the burning of fuel is transferred to water in the boiler (B) to generate steam at high pressure and temperature which then expand in the turbine (T) to a low pressure to produce a shaft work. The steam leaving the turbine is condensed into water in the condenser (C) where cooling water from river or sea (cooling towers) carrying a way the heat released due condensation. The water (condensate) is then fed back to the boiler by the pump (P), and the cycle goes repeating itself [4].

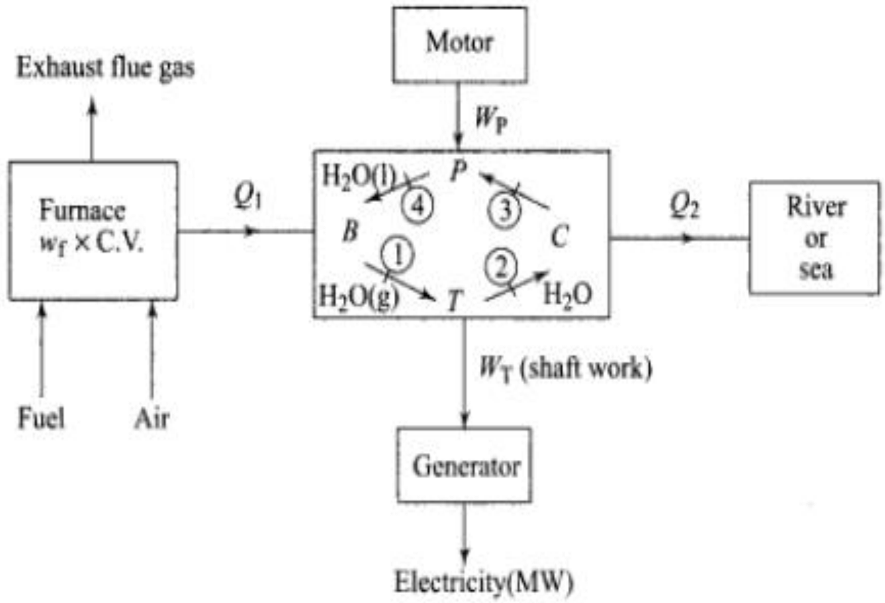

Figure (2-2): Steam Power Plant Bulk Energy Converter

The governing equations are given by:

$$\sum_{cycle} Q_{net} = \sum_{cycle} W_{net} \qquad (2\text{-}1)$$

or

$$Q_1 - Q_2 = W_T - W_P \qquad (2\text{-}2)$$

Where:

Q_1 = heat transferred to the working fluid $^{KJ}/_{kg}$

Q_2 = heat rejected from the working fluid $^{KJ}/_{kg}$

W_T = work transferred to the working fluid $^{KJ}/_{kg}$

W_P = work rejected from the working fluid $^{KJ}/_{kg}$

The simple way to calculate power plant overall efficiency is given by the following equations:

$$\eta_{cycle} = \frac{W_{net}}{Q_1} = \frac{W_T - W_P}{Q_1} = \frac{Q_1 - Q_2}{Q_1} = 1 - \frac{Q_2}{Q_1} \qquad (2\text{-}3)$$

2-3 Effect of Condenser Pressure on Work Output:

Condenser in Rankine cycle is kept below atmospheric pressure to extract more of amount of work from condensing steam, the TS curve in Figure (2-3) explains this.

A normal cycle is 1- 2- 3 - 4 -1, where suppose x is network output. Here the condenser pressure is P_4. Now, the condenser pressure is reduced to$P_{4'}$, the cycle will now become 1' - 2' - 3 - 4'-1', and suppose here x' is the network. Here it can observing that the steam expanded more till point 4' and thus produced more work as compared to expansion till point 4, so the shaded portion in the curve is the increase in net work.

2-4 Limits of Condenser Pressure Lowering:

The following are some factors which effect on condenser pressure such as:

2-4-1 Quality of Exhaust Steam: as shown in Figure (2-3), that quality of steam dryness fraction at point 4' is lower as compared to point 4 where it is in saturated condition. so that steam pits the turbine blades at supersonic speeds and if the steam has more amount of water, i.e. if it’s wet, it will cause pitting in turbine blades causing damage to it and also decrease the turbine efficiency, which is economically not viable. So, it shouldn’t allow the quality of steam to reach below 85%[5].

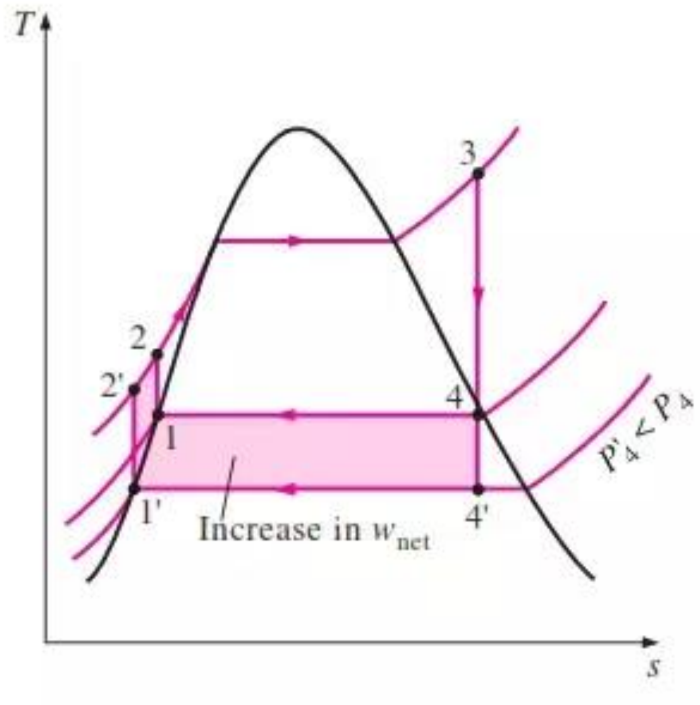

Figure (2-3) T-S Diagram of Rankin Cycle

2-4-2 Temperature of Cooling Water:

The more will be done to make the condenser pressure (vacuum) lesser, will be reduce the temperature of condensing steam, and for effective heat transfer there should be a substantial temperature difference between cooling water and the condensing steam, practically there is a limit on cooling towers also, they cannot cool the cooling water of condenser below a certain temperature. Thus, for maintaining a certain temperature difference, it cannot go on decreasing condenser pressure.

2-4-3 Condenser Losses:

Maintaining an equipment under vacuum condition is also a difficult task, perfect sealing of condenser is not possible, there is always a chance of air ingression, which reduces the condenser efficiency. So, keeping all these things in perspective, engineers calculate an optimal condenser pressure which will not harm the turbine blades, nor increase the cost of sealing the condenser, along with an improved efficiency of overall cycle.

2-5 Evacuation Equipment:

2-5-1 Jet pumps:

Jet pumps have been used for many years in industrial applications where a high-pressure gas such as steam is used to pump a lower-pressure gas. The jet Pump is a simple low-cost device with no moving parts and is particularly convenient for use with troublesome fluids such as two-phase flows, high-temperature gases, or corrosive gases. Jet pumps are usually employed as low-pressure-rise devices and their thermodynamic efficiency is low, i.e., under 20%. Because they are low-cost devices of limited performance potential, there has not been a strong incentive for research and development work on industrial jet pumps [6].

In recent years, applications of jet pumps to boundary layer control systems have become of increasing interest for short take-off and landing aircraft (STOL). Systems have been proposed which use jet pumps to entrain a large flow of secondary air which is then directed over a deflected flap for lift augmentation. In a configuration patented, a jet pump is used to entrain air from one section of the trailing edge of a wing (boundary layer suction upstream of a deflected flap) and then to discharge it over a deflected flap.

In this way, the inherent inefficiency of the jet pump is partially balanced by the double employment of the entrained air for boundary layer control. Jet pumps may also have application in vertical take-off and landing aircraft (VTOL) for direct lift or control force augmentation. The primary, high-pressure flow for the jet pumps can be provided by a bleed from the main engine compressors or by an auxiliary power unit [6].

The use of jet pumps as primary components of VTOL or STOL aircraft systems places new emphasis upon development of design techniques for these devices.

It is essential to be able to minimize the size of jet pumps for particular primary and secondary flow conditions, and to be able to predict the performance of jet pumps over a broad range of operating conditions. However, systematic design and analysis procedures are not available for high-entrainment-ratio compressible-flow jet pumps [6].

2-5-2 Ejector:

Ejectors are supersonic flow induction devices employed for the generation of a vacuum or pumping a fluid. or it is a simple supersonic mechanical devices which can be used to pump without any moving part [7].

High pressure motive fluid enters a converging diverging nozzle and is accelerated to a supersonic Mach number. The pressure at inlet section of mixing chamber is below that of the entrained fluid at its inlet. Consequently, the entrained fluid is drawn into the ejector. The motive and entrained fluids mix between sections of mixing chamber and the uniform mixture is diffused to reach the discharge pressure [7].

In the past, when engineers designed jet ejectors, either a “rule-of-thumb” or “trial-and-error” approach was used. Both approaches may provide unsatisfactory performance, and thus consume too much power, material, and labour.

It was discovered that the constant- pressure configuration provides a better performance than the constant-area configuration, because turbulent mixing in the jet-ejector is achieved more actively under an adverse pressure gradient, which occurs in the constant-area jet ejector, rather than under constant pressure (Kim et al., 1999). Stronger turbulent mixing dissipates the ejector performance. Figure (2-4) illustrate parts of steam jet ejector [8].

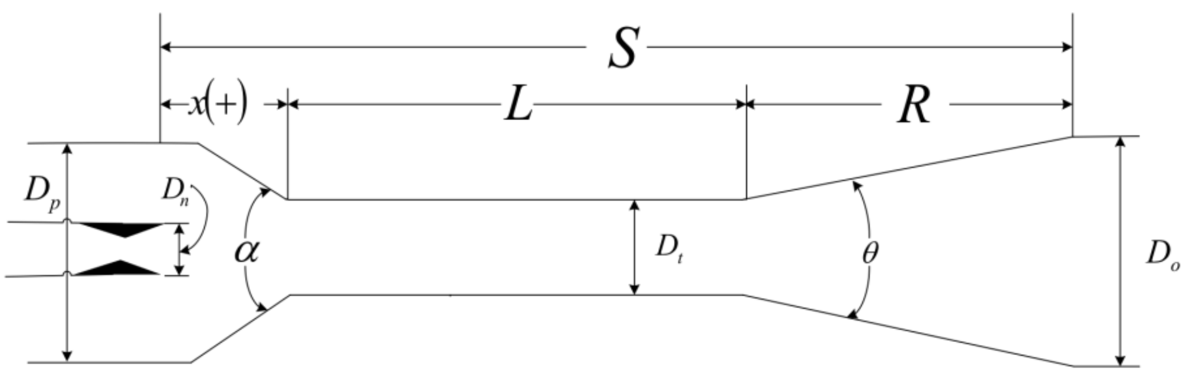

Figure (2-4) Jet Ejector Parts

Some researches provided the mathematical functions, which are valid for both configurations. The mathematical functions are used to calculate:

i. Optimum motive- and propelled-stream velocity as a function of expansion ratio for an arbitrary molecular weight and temperature
ii. Area ratio as a function of entrainment ratio (D_n/D_t)

The jet ejector is classified into two types depending on its convergence configuration:

i. Constant pressure jet ejector
ii. Constant-area jet ejector

The different between both types is shown in Figure (2-5).

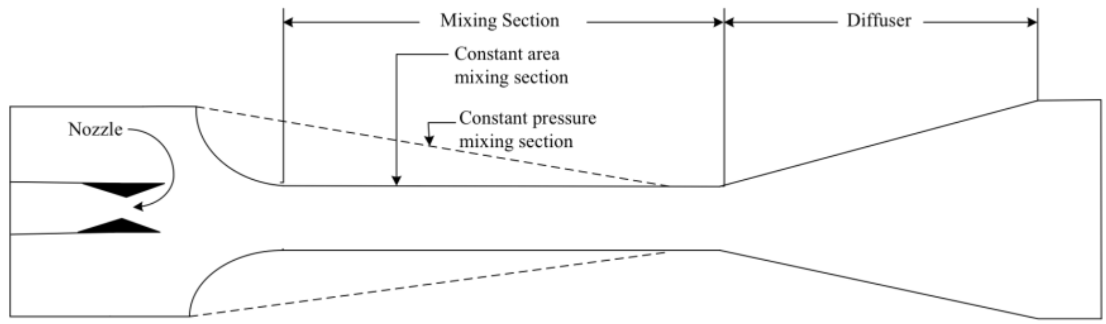

Figure (2-5) Jet Ejector Types

The jet ejector performance is mainly affected by mixing, turbulence, friction, separation, and energy consumption in the suction of the propelled stream. To maximize jet ejector performance, enhancing turbulent mixing should be a major consideration. The literatures indicate that the nozzle geometry should be well designed to boost the tangential shear interaction between the propelled and motive stream. Also both streams should blend completely inside the throat. The jet ejector should be designed properly to diminish turbulence effects. Each part of a jet ejector is explained in the Figure (2-4) [8].

2-5-2-1 Entrainment Ratio:

An experiment conducted by Mellanby concluded that for all practical purposes, the entrainment ratio is independent of the inlet position of the propelled stream. Holton discovered that the entrainment ration is a function of the molecular weight of the fluid, but independent of pressure, and jet ejector design. Figure (2-6) shows the correlation between the entrainment ratio and molecular weight. Furthermore, Holton and Schulz discovered that the entrainment ratio is a linear function of operating temperature, but independent of pressure and jet ejector design. Figure (2-7) displays the effect of the operating temperature on the entrainment ratio.

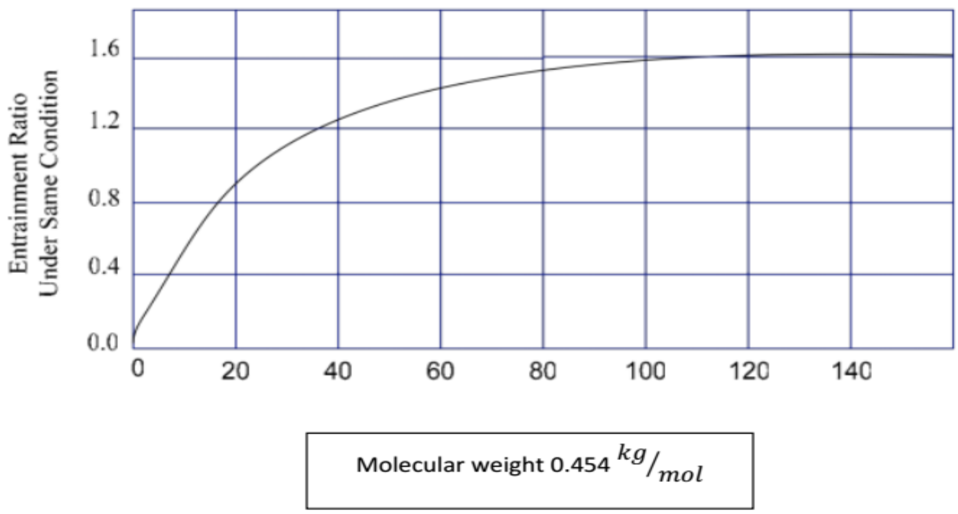

Figure (2-6) The Relationship Between Molecular and Entrainment Ratio

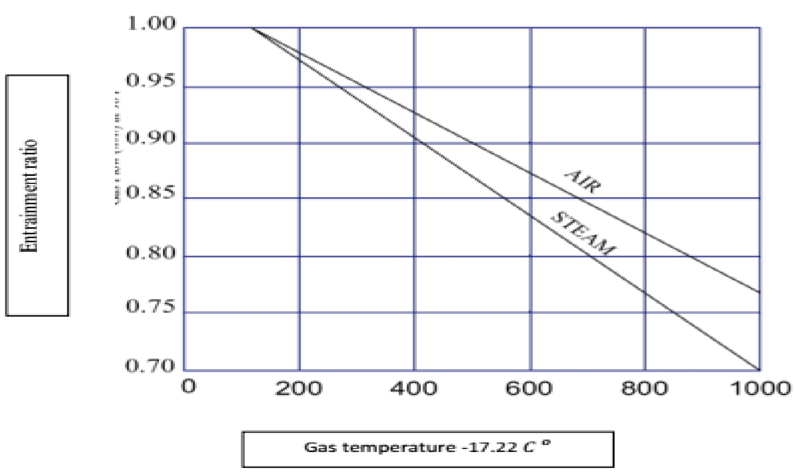

Figure (2-7) The Relationship Between Gas Temperature and Entrainment Ratio

2-5-2-2 Applications of Steam Ejectors:

In general, ejectors, exhausters or jet pumps, as they are called sometimes, are widely used in many industrial fields. Steam ejectors are employed in the power generation, chemical processing, nuclear industries, exhausting air from condensers, vacuum evaporation, distillation, and crystallization, refrigeration, drying, air conditioning, and for pumping large volumes of vapour and gas at low pressure. The main advantage of steam ejectors is that they have no pistons, valves, rotors, or any other moving parts, no lubrication or oil problems, no extremely close tolerances and hence require little maintenance. They are, mechanically, the simplest of all of the present-day type of vacuum pumps. Figure (2-8) illustrates a steam ejector [8].

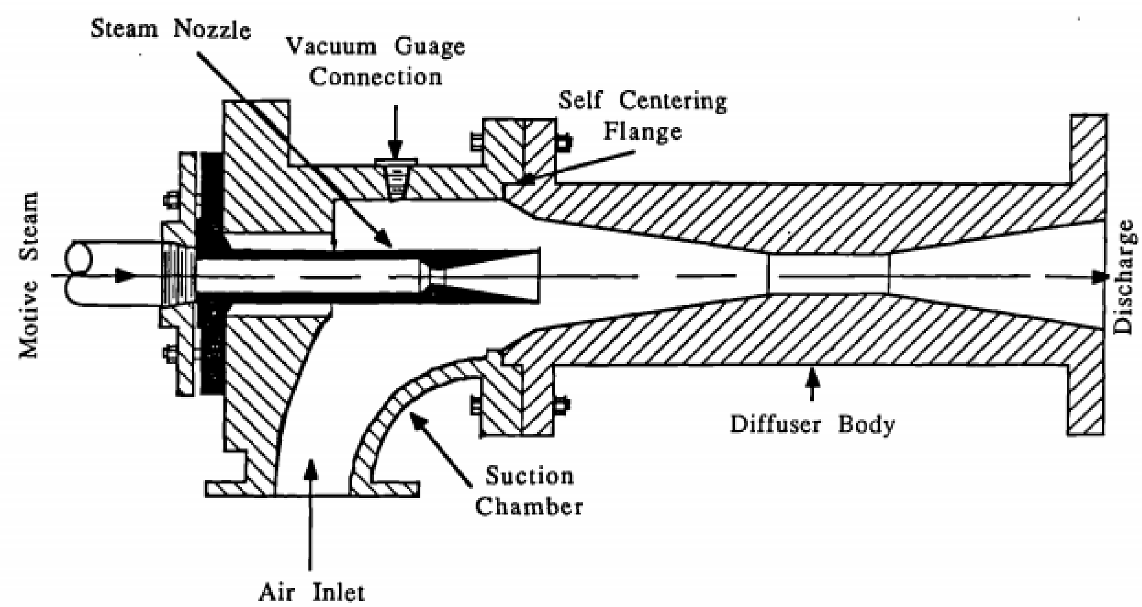

Figure (2-8) Typical Single-Stage Steam Ejector

Also due to their simplicity, jet ejectors have been used for various purposes. A number of the principle applications as follows:

i. **Extraction**: suction of the induced fluid.

ii. **Compression**: compression of the induced fluid discharged at the expansion pressure of the driving fluid.

iii. **Ventilation and air conditioning**: extraction and discharge of gas with small differences in compression near atmospheric pressure.

iv. **Propulsion or lifting**: intermediate compression of the fluid discharged at a certain adaptation velocity.

v. **Uniform mixing of two streams**: providing a uniform concentration or temperature in a chemical reaction.

vi. **Conveyance**: pneumatic or hydraulic transport of products in powder form or fractions [8].

Compared with mechanical pumps, steam ejectors have very low efficiencies when used in normal pumping applications but when a source of waste or low grade steam is available, a steam ejector may be cheaper to operate than a mechanical pump. Steam ejectors have many applications, such as heating, humidifying and pumping toxic and solids-bearing fluids, where a mechanical pump may be unsuitable [9].

2-5-2-3 Principles of Operation:

A single stage steam ejector depends for its pumping action on an expanding motive nozzle discharging a supersonic velocity jet at low pressure across a converging chamber, which is connected to the equipment to be evacuated, and so bringing the suction fluid into intimate contact with the high-velocity motive steam. The suction fluid is then entrained by the steam and mixed with it in a parallel section. Then passing through a diffuser the velocity is reduced and discharge pressure recovered [9].

2-5-2-4 Multi-Stage Steam Ejectors:

A single stage ejector has an operating limit to the attainable compression ratio which is usually about 8:1. The compression ratio is the ratio of the discharge pressure to the suction pressure ($\frac{P_d}{P_s}$) so, attempting to produce a vacuum more than about 100 unit, means that the ejector is using steam uneconomically. Thus, to have progressively higher vacuum more stages are added as indicated in Figure (2-9), a satisfactory number being chosen to improve the work stability and to use the steam more economically. In other words, the number of stages required is the more dependent upon the vacuum required, and upon which is the more important to the user, the operational cost or the equipment cost.

One must bear in mind that, when using a two-stage steam ejector, the second stage has to be large enough in capacity to handle the initial suction load plus the motive steam from the first stage unless an interstage condenser is used [8].

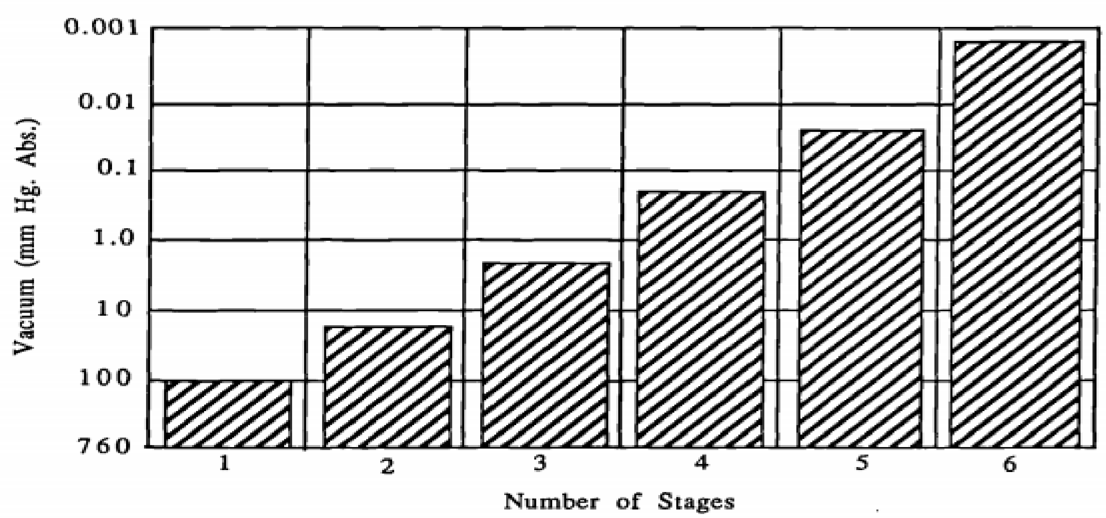

Figure (2-9) Range of Ejector Suction Pressure

2-1-5-2-5 Ejector Flow Scheme:

Figure (2-10) shows the ejector flow scheme for constant area mixing. Three essential cross sections enter the analysis with the following specifications:

i. At the entrance to the mixing section the driver-gas and the driven gas which are referred to as primary and secondary medium, respectively, enter with the Mach numbers M_P and M_S. The static pressure for each medium is assumed to be the same at this cross section.
ii. At the end of the mixing section both media are assumed to be completely mixed. In case of supersonic inlet conditions it is assumed that due to a normal or pseudo-shock the exit Mach number is always subsonic.
iii. At the exit of the diffuser, conditions vary significantly with each ejector application. For the ejector pump a maximum of static pressure should in general, be obtained [1].

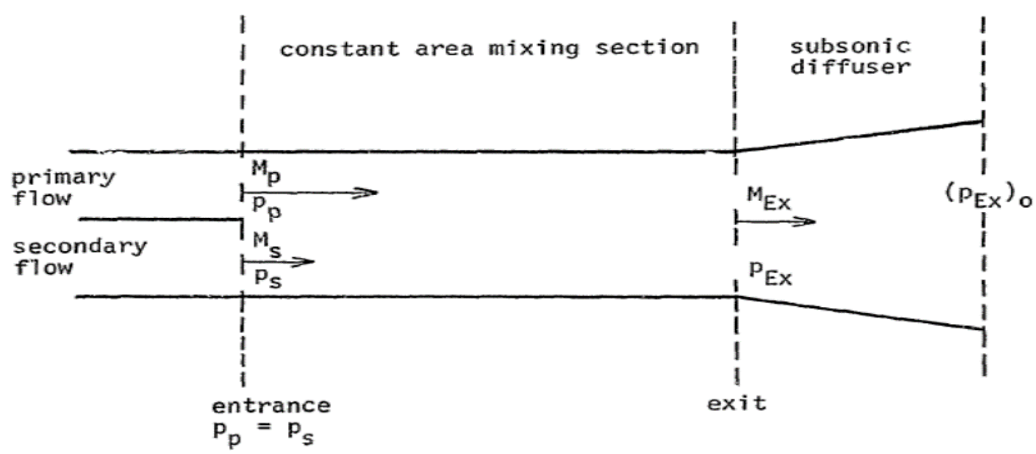

Figure (2-10) Basic Ejector Flow Scheme for Constant Area Mixing

2-6 Previous Study:

Significant numerical and experimental analyses have been devoted to understanding the variety of flow regimes present in steady flow ejectors. By practise certain regimes are more conducive to achieving high performance (i.e. high entrainment ratios).

In particular, the entrainment ratio is seen to be highest when the entrained fluid reaches a choked condition in the mixing region. In addition, the expansion regime of the motive nozzle (under-, perfectly- or over-expanded) appears to influence performance. A method was proposed to model an ejector of optimal geometry, designed for a favourable flow regime.

Then, rather than focusing upon the maximization of efficiency, it seek operational conditions that maximise ejector efficiency, specifically the reversible entrainment ratio efficiency. Ejector efficiency is found to be highest at low compression ratios and at low driving pressure ratios. However, at lower compression ratios, the optimal area of the mixing chamber becomes large relative to the motive nozzle throat area.

Conclusively when using a simple model of an optimal ejector, a parametric study has been performed to identify operating conditions conducive to high ejector efficiency. This optimal ejector is designed such that the motive fluid nozzle is perfectly expanded and the entrained fluid is choked at the entrance to the constant area mixing section. The research is concluded that:

i. The efficiency of an ideal gas ejector is highest when design for conditions of low compression ratio and low driving pressure ratio.

ii. At low compression ratios the area ratio of the mixing section to the motive nozzle throat increases, indicating that the required size of the ejector increases.

iii. As the driving pressure ratio increases, the required size of the ejector increases

iv. An optimal inlet temperature ratio of the inlet fluids appears to exist for fixed inlet and discharge pressures, although the ejector area ratio appears insensitive to the inlet temperature ratio [8]

The efficiency of the ejector cycle is very sensitive to the ejector efficiency. This research provides a literature review on ejector efficiencies in various ejector systems, such as vapour compression systems, solar driven ejector systems.

The definitions of overall ejector efficiency and ejector component efficiencies in literature are summarized.

The assumed constant ejector component efficiencies used in the ejector modelling, and the empirical correlations of the ejector efficiencies developed based on the external measured parameters are summarized and compared; the methods of determining energy efficiencies are summarized. The effects of ejector geometries, operating conditions and working fluid characteristics on ejector efficiencies are discussed. This review will be useful for further research on ejector efficiency, optimal design and control of ejectors and ejector systems[2].

Conclusively describes a number of research studies on overall ejector efficiency and ejector component efficiencies. The different definitions of overall ejector efficiencies were reviewed and discussed. The investigation about the effects of ejector geometrics, operating conditions and working fluid characteristics on overall ejector efficiencies. The assumed constant ejector component efficiencies are summarized. The methods of determining ejector component efficiencies were reviewed, such as CFD simulation method, a method combining experimental data and simulation modelling. It is hoped that this contribution will be useful for the future research on the optimal design and control of ejectors and ejector systems. Though a large amount of works have been conducted on ejector efficiencies, further efforts are still needed:

i. To improve the methods of determining actual ejector component efficiencies.
ii. To make a comprehensive study about the effects of operating conditions, ejector geometric and working fluid characteristic of ejector component efficiencies [2].

In the study of a two part investigation of single- and two-stage ejector systems in which the primary fluid is steam and the secondary fluid is air.

The first part is an experimental investigation. The vacuum created by the ejector is strongly affected by the distance between the steam nozzle outlet and the diffuser throat section. The relation between this distance, which is called here "the nozzle optimum distance (Lo)", and the geometrical and operating parameters of the nozzle and the diffuser were investigated and forms the object of this part of the thesis.

The second part is a theoretical approach. The exit Mach number for the nozzle was found by using the one-dimensional gas dynamic equations together with the first law of thermodynamics.

Also a two-dimensional approach using the Method of Characteristics was used to find the exit Mach number and the characteristic net of the flow from the throat to the outlet of the nozzle. Two computer programmes were written on the basis of these two different theoretical techniques and the comparison between the results for the exit Mach number found to be 95% in agreement over the pressure range of the experimental work.

A computer programme was also written using the Method of Characteristics to find the shape i.e. the characteristic net and the constant density lines within the flow of the steam jet leaving the nozzle and entering the diffuser. It is believed that the jet diameter at the point where it meets the diffuser wall, which is called in this work "the optimum jet diameter (Do)". Is strongly related to the nozzle optimum distance (Lo).

When the characteristic net for the jet is drawn, its point of interception with the diffuser wall can be found and then (Do) can be measured. This diameter (Do) was then related to the ejector dimensional parameters and the ejector operating conditions; an equation was found to predict the optimum jet diameter from this equation (Dop)e. Then the predicted optimum nozzles distance (Lop)e was determined by using a computer program where the characteristic net meets the diffuser wall at the calculated optimum jet diameter (Dop)e.

Finally, the experimentally determined value of the nozzle optimum distance (Lo) was compared to the theoretically predicted value, and the average error was found to be 1.23%.

The equation can be used in multi-stage ejector design. For instance, in a two stage ejector after the choice of a suitable intermediate pressure then each stage can be treated independently so that for the:

- First stage (steam expansion ratio) E = $\frac{P_m}{P_s}$, (the pressure ratio) C = $\frac{P_i}{P_s}$
- Second stage E=$\frac{P_m}{P_i}$, C=$\frac{P_d}{P_s}$.

The load to the second stage will be the sum of the ejector load plus the steam to the first stage if the ejector set has no interstage condenser. However if an intercondenser is fitted the load to the second stage will be the non condensables plus vapour carryover calculated using Dalton's law, at the temperature of the load. This temperature is typically 3-5°C above the cooling water inlet temperature when operating in a counter current mode [10].

Another experimental observations for the performance of a jet pump are presented with two different suction configurations and designs. The experimental rig was constructed in such a way it can be used with up feed (negative suction head) or down feed (positive suction head).

During experimental study water is used in both motive and pumped sides. The effect of nozzle-to-throat spacing to nozzle diameter ratio on the jet pump performance was also tested, with different flow rates and motive pressures, in both cases (up feed and down feed). It was found that the best efficiency for the jet pump is attained with the up feeding configuration [3].

A recent research with numerical investigation of axisymmetric subsonic air to air ejector. An analysis of flow and mixing processes in cylindrical mixing chamber are made. Several modes with different velocity and ejection ratio are presented. The mixing processes are described and differences between flow in the initial region of mixing and the main region of mixing are described. The lengths of both regions are evaluated. Transition point and point where the mixing processes are finished are identified. It was found that the length of the initial region of mixing is strongly dependent on the velocity ratio, while the length of the main region of mixing is slightly dependent on velocity ratio [11].

Research was performed to optimize high-efficiency jet ejector geometry by varying nozzle diameter ratios from 0.03 to 0.21, and motive velocities from Mach 0.39 to 1.97. The high-efficiency jet ejector was simulated by Fluent (CFD) software. A conventional finite-volume scheme was utilized to solve two-dimensional transport equations with the standard k-! turbulence model .

In this study of a constant-area jet ejector, all parameters were expressed in dimensionless terms. The objective of this study was to investigate the optimum length, throat diameter, nozzle position, and inlet curvature of the convergence section. Also, the optimum compression ratio and efficiency were determined. By comparing simulation results to an experiment, Computational Fluid Dynamics (CFD) modelling has shown high-quality results. The overall deviation was 8.19%, thus confirming the model accuracy. Dimensionless analysis was performed to make the research results applicable to any fluid, operating pressure, and geometric scale. A multi-stage jet ejector system with a total 1.2 compression ratio was analysed to present how the research results may be used to solve an actual design problem.

The results from the optimization study indicate that the jet ejector efficiency improves significantly compared to a conventional jet-ejector design. In cases with a subsonic motive velocity, the efficiency of the jet ejector is greater than 90%. A high compression ratio can be achieved with a large nozzle diameter ratio. Dimensionless group analysis reveals that the research results are valid for any fluid, operating pressure, and geometric scale for a given motive-stream Mach number and Reynolds ratio between the motive and propelled streams.

A multi-stage jet ejector system with a total 1.2 compression ratio was analysed based on the optimization results. The result indicates that the system requires a lot of high-pressure motive steam, which is uneconomic. A high-efficiency jet ejector with mixing vanes is proposed to reduce the motive-steam consumption and is recommended for further study [9].

CHAPTER III

3-1 Background:

By using the geometrical diamention of the ejector other diamentions will be optimized by standart expremental formulers, therefore to chose the optimum values that will give a good vacccum on air suction line then by make a simulation used fluid analised computer programming or computitional fluid dyanamic (CFD) software to compare this values (given by simulation) with that which recorded under operation condition to estmate modifications must be act on desing or initional condition.

The simulation will be execution many times, frist one with optimum values from standard formula and the second simulations with modifications happened (include change in throat length and divergence length) and at last the simulation with changing in motive steam pressure (inlet pressure).

3-2 Criteria for Modification:

The modification on ejector to improve vacuum besides to get good efficency depending on:

i. Dependence on lowest value of suction pressure (in frist simulation) whitch has best vacuum value.
ii. Second simulation will depend on difference in length values of throat and divergence section.
iii. Changing the initinal condition sufficiently to improve the vaccum, like changing in motive steam pressure.

Here is the typical ejector in Figure (3-1) which used in kosti thermal power station (KTPS) with outer diamentions without the iner diamentions such as motive steam nozzle, throat diameter,throat length , convergence angle and divergence angle because its not to scale drawing (NTS).

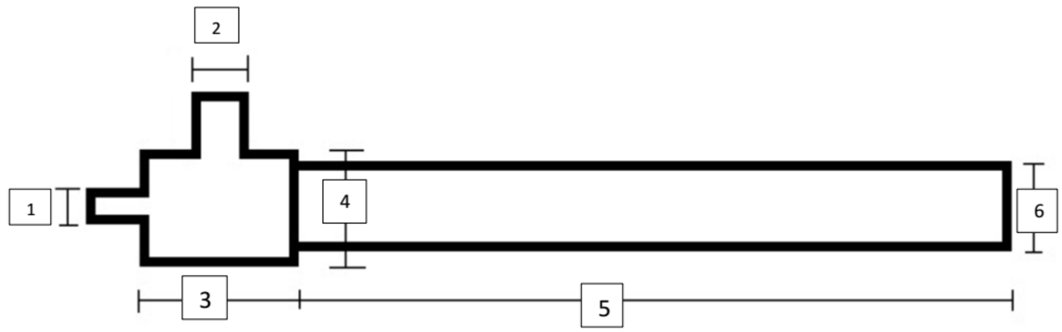

Figure (3-1) Typical ejector used in KTPS

1- Motive steam inlet diameter (0.064m)
2- Air inlet diameter (0.121m)
3- Mixing chamber length (0.37m)
4- Mixing chamber high (0.223m)
5- Diffuser length (1.3m)
6- Divergent outlet diameter (0.21m)

3-3 Design characteristic:

The constraints may be related to the required performance of the ejector or may be dimensional, restricting the size of the ejector. For most applications, the main design constraint falls into the following categories:

a) The motive steam pressure (p_m) is specified,

b) The suction pressure (p_s), is specified,

c) The secondary mass flow rate i.e. the mass of air (m_a), is specified [10].

The each section of the ejector parts was design with a number of relations depend on standard formulas which gives a wide range of these sections characteristic.

3-3-1 Convergence Section:

It is the first part of diffuser which the propelled-stream entered and passing throw the diffuser throat, According to Kroll , Engdahl and Holton , Mellanby and Watson found that the best design for the convergence section is a well-rounded or bell-mouthed entry. A conical or tapered entry is recommended to have an entry angle (α) greater than 20 degrees, because the nozzle jet which has a general angle of about 20 degrees will not create objectionable shock and eddy losses at the convergence inlet.

Watson did an experiment and stated that 25 degrees is about the best convergence angle.

Regarding the well-rounded geometry, a conical entry reduces the flow 2%, whereas a coupling and sharp entry reduce the flow 4 and 11%, respectively [10].

3-3-2 Throat Section:

Kroll also discusses that Mellanby and Watson reported that diffusers with a throat section created a greater vacuum than diffusers without a throat section. Mellanby also showed that a parallel throat throughout is inferior, but still much better than no parallel throat at all.

The length of the throat section must be designed properly. It should be sufficiently long to create a uniform velocity profile before the entrance of the divergence section. The uniform velocity decreases the total energy losses in the divergence section, thus obtaining better high-pressure recovery .

Two literature sources cited in Kroll (Duperow and Bossart,; and Keenan and Neumann, reported that an optimum throat length is about 7 times the throat diameter, whereas Engdahl came across with another optimum value of 7.5 times the throat diameter. Additionally, lengths of 5 to 10 times the throat diameter provided within 3% of optimum performance. Although the optimum length increased slightly with pressure and throat diameter, the increase was less than 1 diameter even when these factors were doubled (Keenan and Newmann). Engdahl reported that any length between 4 and 14 throat diameters will give within 4% of optimum performance. According to many literature sources, the length should be 7 to 9 times the throat diameter for the best performance.

The optimal throat diameter is sensitive to jet ejector parameters, especially the entrainment ratio. A small change in throat diameter creates a huge change in the entrainment ratio. If the throat area is too large, fluid leaks back into system; if it is too small, choking occurs. So, the throat diameter must be designed properly to obtain the best performance [8].

Vil'der found a simplified method of calculating the throat diameter (D_t) in relation with the suction air and motive steam mass flow rates (m_a, m_{st}) and the output pressure (discharge pressure) of the ejector, by the use of the equation (3-1) the diameter of throat can be calculated as:

$$D_t = \quad 1.6 * \left(\frac{\left(\frac{18}{29}\right)m_a + m_s \ ({}^{kg}/_{hr})}{p_d \ ({}^{kgf}/_{cm^2})}\right)^{\frac{1}{2}} \tag{3-1}$$

3-3-3 Divergence Section:

Kroll (1947) indicated that the angle of the divergent section, θ, is usually 4 to 10 degrees. Too rapid a divergence immediately after the throat is not recommended (Kroll). The divergent length, say from 4 to 10 times the throat diameter, is desired for pressure recovery. The length, however, may be as short as twice the throat diameter if necessary. It was discovered that eliminating the divergence section reduced the entrainment ratio by about 20% [8].

3-3-4 The Nozzle:

Two factors of the nozzle influence jet ejector performance:

1. Nozzle design

2. Nozzle position

Fewer researchers have studied the effect of nozzle design on jet ejector performance than nozzle position. Hill and Hedges (1974) studied the influence of nozzle design on jet ejector performance. In their experiment, two conically diverging nozzles were tested, but differing in the divergence angle. The exit and throat diameters of the nozzle were fixed in both cases. The experimental results show that the overall jet ejector performance was not influenced by the nozzle design. According to Kroll (1947), a study done by Engdahl and Holton confirms the above statement. They found that the nozzle, which was designed by conventional methods for a specific pressure, performed only slightly better than a simple straight-hole nozzle at pressure up to 170 psig (11.721 bar) [8].

Also, a machined nozzle with a convergence section and a 10 degree angle of divergence was only 3 to 6%better than a 100-psig (6.895 bar) small pipe-cap nozzle made by drilling a hole in a standard pipe cap. However, altering the nozzle design affects the motive stream velocity. This was studied explicitly by Berkeley. He also found that under normal circumstances, the expansion of motive stream in the ejector of a well-designed nozzle is almost always a fairy efficient part of the overall flow process. Therefore, very little energy is lost in the nozzle. But the task of efficiently converting velocity back into pressure is very difficult because energy is lost in this process. Additionally, Kroll reported that a poorly shaped nozzle causes unnecessary shock losses and useless lateral expansion, which decrease jet ejector efficiency tremendously.

The position of the nozzle has a greater effect on jet ejector performance than its design. A number of researchers investigated the optimum position of the nozzle in a jet ejector. Croft and Lilley ; and Kim et al. report that turbulence in the mixing tube decreases when the nozzle is placed right at the entrance of the throat section; however, Croft and Lilley also discovered that when the nozzle moves closer to the mixing tube, the entrainment ratio decreases. ESDU recommends placing the nozzle exit between 0.5 and1.0 lengths of throat diameter upstream of the mixing chamber. Not only the jet ejector performance, but also the mixing distance of the motive and propelled streams is affected by the nozzle position. Kroll has suggested that nozzle position should be adjustable to obtain the best performance using field adjustments [9].

Further, it is important to have the nozzle centred with the throat tube. He also recommended that the nozzle should be cleaned as often as possible for best performance.

The nozzle throat diameter calculated from equation (3-2):

$$d_t = 1.6 * \left(\frac{m_s\ ({kg}/{hr})}{p_m\ ({kgf}/{cm^2})}\right)^{\frac{1}{2}} \qquad (3\text{-}2)$$

3-4 Relation between Weather and Vacuum:

The condenser vacuum is effected by ambient condition (temperature and humanity), the steam flows throw condenser from last stage turbine (low pressure turbine) need a specific cooling water.

In summer season the outside ambient temperature is to be very hot which will be unable to carry out enough sensible heat from steam to condensate it. But the opposite happened in winter season and it will be no need for two ejector be in service.

Also the Table (3-1),(3-2) illustrate the effect of load and season on condenser vacuum in peak and base load respectively . This readings taken from three months (may to july)

Table (3-1): Vacuum on Peak Load

NO.	**LOAD** (mw)	**CONDENSER PRESSURE** (bar)
1	108.8	0.123
2	109.0	0.126
3	113.8	0.119
4	117.3	0.122
5	118.6	0.120

6	119.3	0.121
7	119.6	0.119
8	119.6	0.125
9	120.7	0.123
10	120.3	0.131
11	120.8	0.126
12	121.6	0.123
13	122.2	0.118
14	123.3	0.119
15	124.1	0.122

Table (3-2): Vacuum on Base Load

NO.	**LOAD** (mw)	**CONDENSER PRESSURE** (bar)
1	83.6	0.086
2	84.1	0.091
3	84.1	0.094
4	84.7	0.090
5	84.9	0.100
6	85.1	0.087
7	85.2	0.093
8	85.2	0.097
9	85.3	0.103
10	85.5	0.100
11	85.9	0.096
12	86.3	0.090

13	86.6	0.098
14	86.9	0.084
15	95.0	0.108

Also vacuum can effects by the precipitation on condenser tubes witch lead to tube scaling if water using is not treated chemically enough, in this case the condenser need to clean by some chemical solution in order to release the scale on tubes. Table (3-3) show data of condenser vacuum after cleaning.

Table (3-3) : Vacuums after Condenser Cleaned

NO.	**LOAD** (mw)	**CONDENSER PRESSURE** (bar)
1	13.80	0.062
2	14.70	0.071
3	108.5	0.092
4	119.3	0.080
5	120.4	0.101
6	121.2	0.086
7	121.7	0.101
8	121.7	0.102
9	121.9	0.088
10	123.0	0.084
11	123.0	0.094
12	123.1	0.095
13	124.0	0.097
14	124.8	0.096

15	124.8	0.100

All those load effect on vacuum (peak or base load) have influence with different degree Figure (3-2) explain more all those effects by curves.

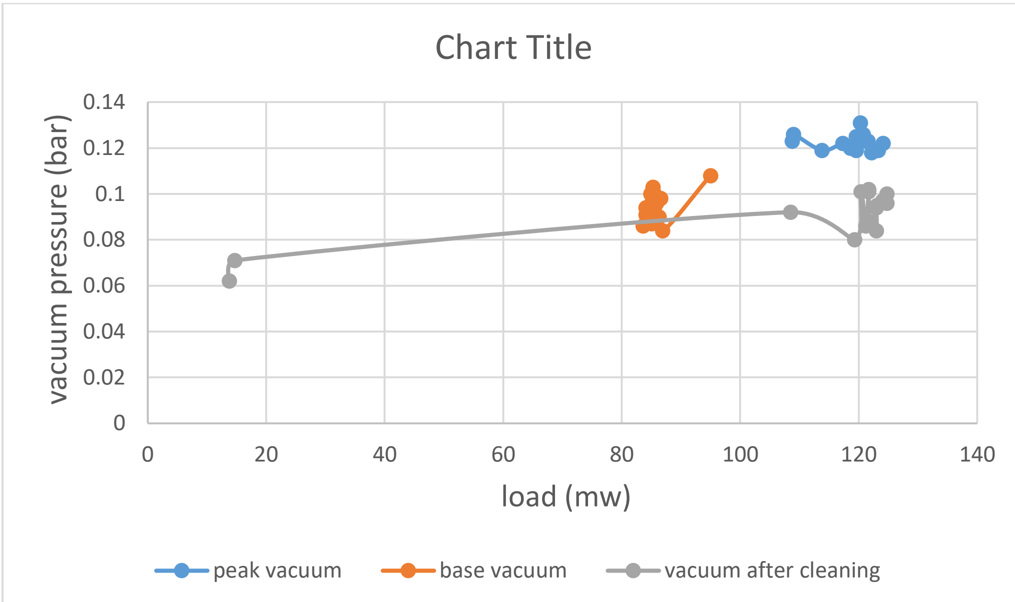

Figure (3-2): Relation between Condenser Vacuum and Load

CHAPTER IV

4-1 CALCULATIONS:

From design data:

m_a=25.5 $^{kg}/_{hr}$

m_s= 81.5 $^{kg}/_{hr}$

p_d= 3.5 bar

From operation data:

P_{HV} = - 0.8 bar

$P_{ej.} = \frac{-0.8}{2}$ = - 0.4 bar

4-1-1 Convergence Section:

The convergence section of ejector must be conical or tapered entry and have an entry angle (α) greater than 20 degrees let’s be 25 according to Watson (1933).

∴ α=25^o

4-1-2 Throat Section:

By use Vil'der (1964) formula the diameter of throat can be calculated by known of nozzle discharge pressure, suction air and motive steam mass flow rates. By using equation (3-1) :

$$D_t = 1.6 * \left(\frac{\left(\frac{18}{29}\right)m_a + m_s\ (^{kg}/_{hr})}{p_d\ (^{kgf}/_{cm^2})}\right)^{\frac{1}{2}}$$

$$D_t = 1.6 * \left(\frac{\left(\frac{18}{29}\right)25.5+81.5\ \left({kg}/{hr}\right)}{3.5\ \left({kgf}/{cm^2}\right)}\right)^{\frac{1}{2}} = 8.44 \text{ cm}$$

$$=0.0844\text{m}$$

For throat length and according two literature sources cited in Kroll (1947) (Duperow and Bossart (1927) and Keenan and Neumann, (1942)) reported that an optimum throat length is about 7 times the throat diameter.

There are wide range 4-14 times of throat diameter but its optimum one depending on.

- Optimum throat length (L_{th})= 7*0.0844 = 0.5908m

That mean there have throat lengths in a different ranges given in Table (4-1):

Table (4-1) Throat Lengths

No.	**TIMES**	**THROAT DIAMETER** (m)	**THROAT LENGTH** (m)
1	4	0.0844	0.3376
2	7	0.0844	0.5908
3	8	0.0844	0.6752
4	10	0.0844	0.8440
5	12	0.0844	1.0128
6	14	0.0844	1.1816

4-1-3 Divergence Section:

According to Kroll (1947) indicated that the angle of the divergent section θ, is usually 4 to 10 degrees. The divergent length from 4 to 10 times the throat diameter, is desired for pressure recovery. Let the one of midpoints of divergence length will be the optimum value then the optimum divergence length (L_{th})= 6*0.0844 = 0.5064m

That mean there have divergent lengths in a different ranges given in Table (4-2):

Table (4-2) Divergence Lengths

No.	TIMES	THROAT DIAMETER (m)	DIVERGENCE LENGTH (m)
1	4	0.0844	0.3376
2	5	0.0844	0.4220
3	6	0.0844	0.5064
4	7	0.0844	0.5908
5	8	0.0844	0.6752
6	10	0.0844	0.8440

4-1-4 The Motive Nozzle:

The nozzle throat diameter calculated from equation (3-2):

$$d_t = 1.6\sqrt{\frac{m_s\ ({}^{kg}/_{hr})}{p_m\ ({}^{kgf}/_{cm^2})}}$$

d_t=1.6 $\sqrt{\frac{81.5\ ({kg}/_{hr})}{12\ ({kgf}/_{cm^2})}}$

- d_t= 4.17c = 0.0417m

4-2 THE RESULT:

4-2-1 Simulation with optimum values:

By using ansys software (CFD) then have to simulate the flow at optimum throat length (0.5908) and optimum divergence length (0.5064) with different numbers of iterations and motive inlet pressure 12 bar which gives the suction pressures as given in Table (4-3) and Figure (4-1) explain the pressure dissipate on typical ejector with 12 bar inlet on suction.

Table (4-3): optimum value simulation result

No.	**No. of iteration**	**Suction pressure** (Pascal)
1	700	-33700.438
2	800	-4552.4175
3	900	-11847.525
4	1000	-18794.148
5	1100	-29567.787
6	1200	-32729.713
7	1300	-36645.711
8	1400	-41440.785
9	1500	-46857.953

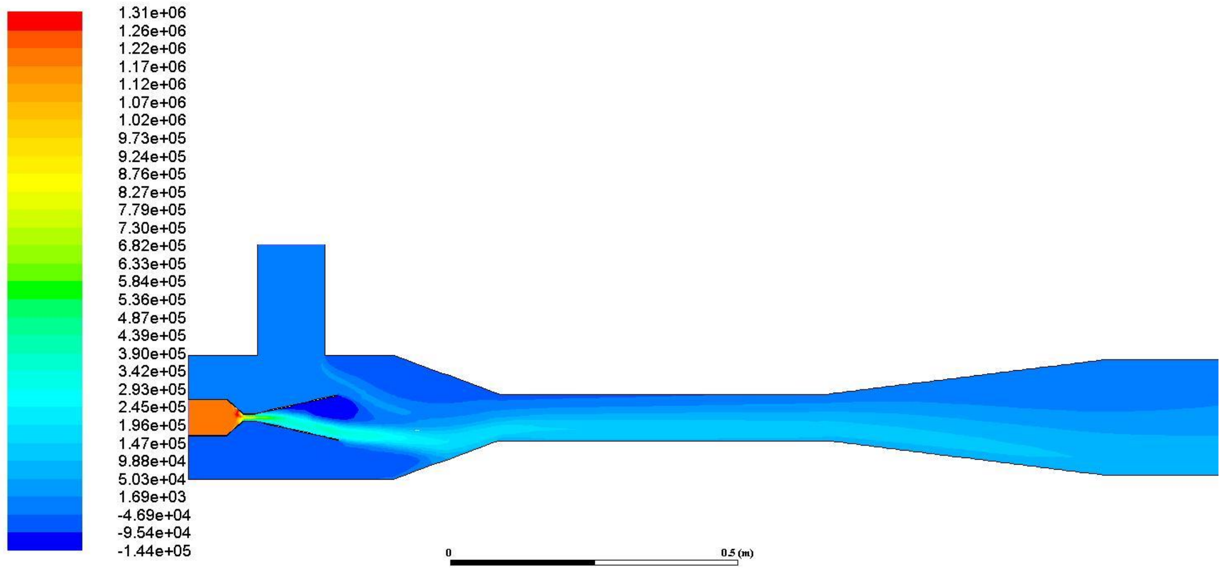

Figure 4-1: pressure dissipate on KTPS ejector

From table (4-3) with several numbers of iteration it's seem that the iteration 1500 times has less values recorded of suction pressure therefore it meet the lesser value of suction pressure recorded.

4-2-2 Simulation with changed in lengths:

Now by changing in throat length and divergence length one by one as inserted in table (4-4), the values of suction pressure with motive inlet pressure 12 bar depending on this changing are given also in Table (4-4), and Figure (4-2) explain the pressure dissipate on inlet of ejector when changing in lengths with 12 bar.

Table (4-4): different lengths simulation result

NO.	THROAT LENGTH (m)	DIVERGENCE LENGTH (m)	Suction pres. (Pascal)
1	0.3376	0.844	-76473.688
2	0.5908	0.6752	-43503.816
3	0.6752	0.5908	-52845.617
4	0.8440	0.5064	-28406.566
5	1.0128	0.4220	-58717.602
6	1.1816	0.3376	-75208.336

The ejector with throat length (0.3376) and divergence length (0.844) has lesser value of suction pressure recorded than others

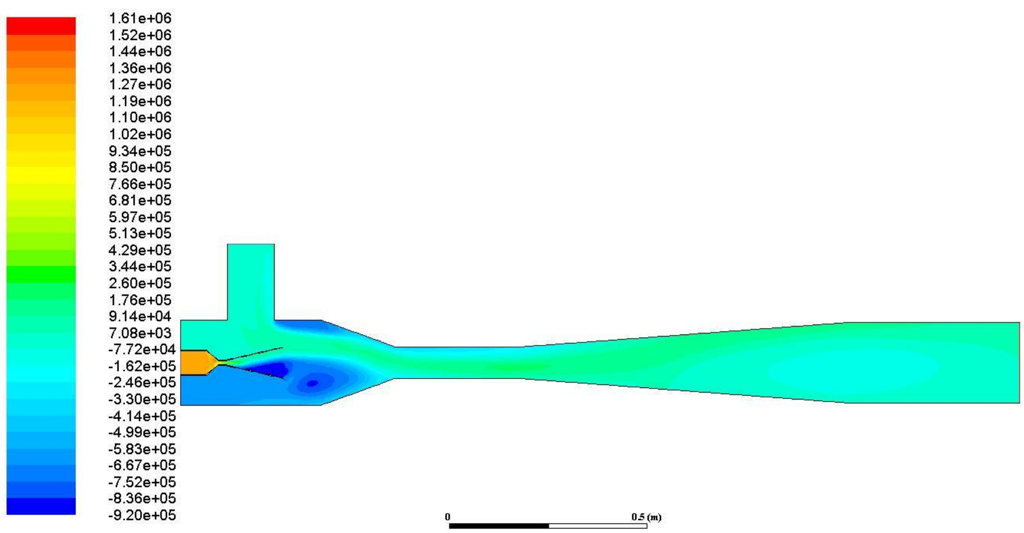

Figure 4-2: pressure dissipate on ejector when changing in lengths with 12 bar inlet

When the lengths changed the lowest suction produced is -76473.688 pas or -0.765 bar

4-2-3 Simulation with changing in motive steam pressure:

Here the motive steam pressure was changed between: (8, 10, and 15 bar) to see the changing in vacuum produced on suction air line,

i. **Motive Steam 8 Bar:** the values of suction pressure with motive inlet pressure 8 bar are given in Table (4-5) and Figure (4-3) explain the pressure dissipate on inlet of ejector when the motive pressure was 8 bar and less vacuum recorded.

Table (4-5): 8 bar motive steam simulation result

NO.	**THROAT LENGTH** (m)	**DIVERGENCE LENGTH** (m)	**Sum.** (m)	**Suction pres.** (Pascal)
1	0.3376	0.844	1.0972	-10235.682
2	0.5908	0.6752	1.1816	-51602.137
3	0.6752	0.5908	1.266	-20204.955
4	0.8440	0.5064	1.3504	-27401.15
5	1.0128	0.4220	1.4348	-38775.289
6	1.1816	0.3376	1.5183	-46202.031

The ejector with throat length (0.5908) and divergence length (0.6752) has lesser value of suction pressure recorded than others.

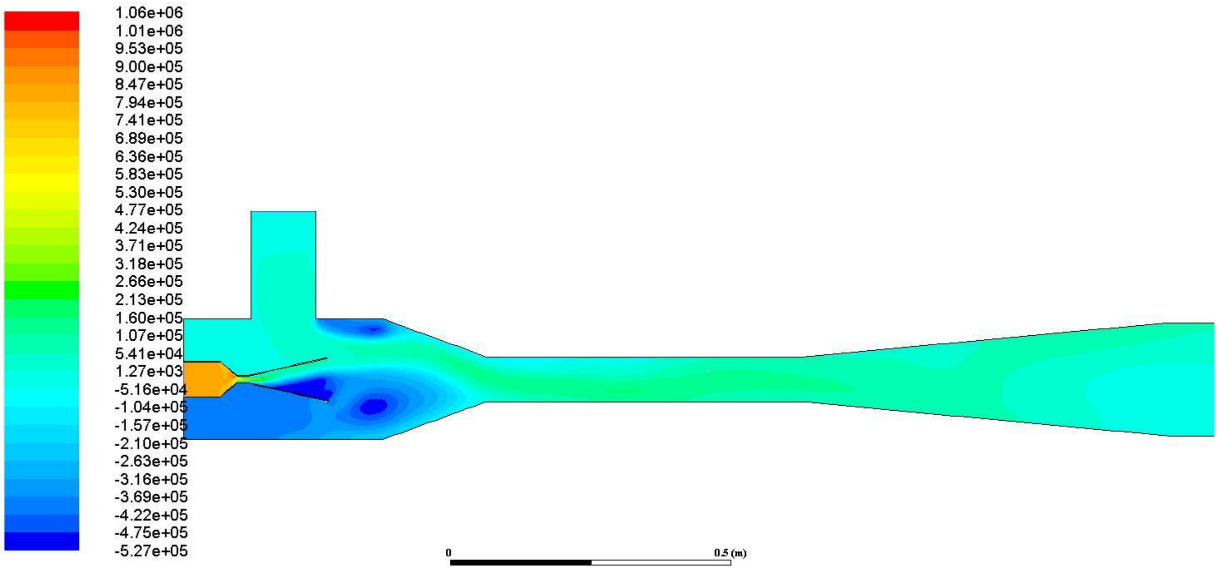

Figure 4-3: pressure dissipate on ejector with 8 bar inlet

The lowest suction produced is -51602.137 pas or -0.516 bar

ii. **Motive Steam 10 Bar:** the values of suction pressure with motive inlet pressure 10 bar are given in Table (4-6) and Figure (4-4) explain the pressure dissipate on inlet of ejector when the motive pressure was 10 bar and less vacuum recorded.

Table (4-6): 10 bar motive steam simulation result

NO.	**THROAT LENGTH** (m)	**DIVERGENCE LENGTH** (m)	**Sum.** (m)	**Suction pres.** (Pascal)
1	0.3376	0.844	1.0972	-14860.917
2	0.5908	0.6752	1.1816	-31157.666
3	0.6752	0.5908	1.266	-19302.729
4	0.8440	0.5064	1.3504	-22858.732
5	1.0128	0.4220	1.4348	-8851.1621
6	1.1816	0.3376	1.5183	-45147.563

The ejector with throat length (1.1816) and divergence length (0.3376) has lesser value of suction pressure recorded than others.

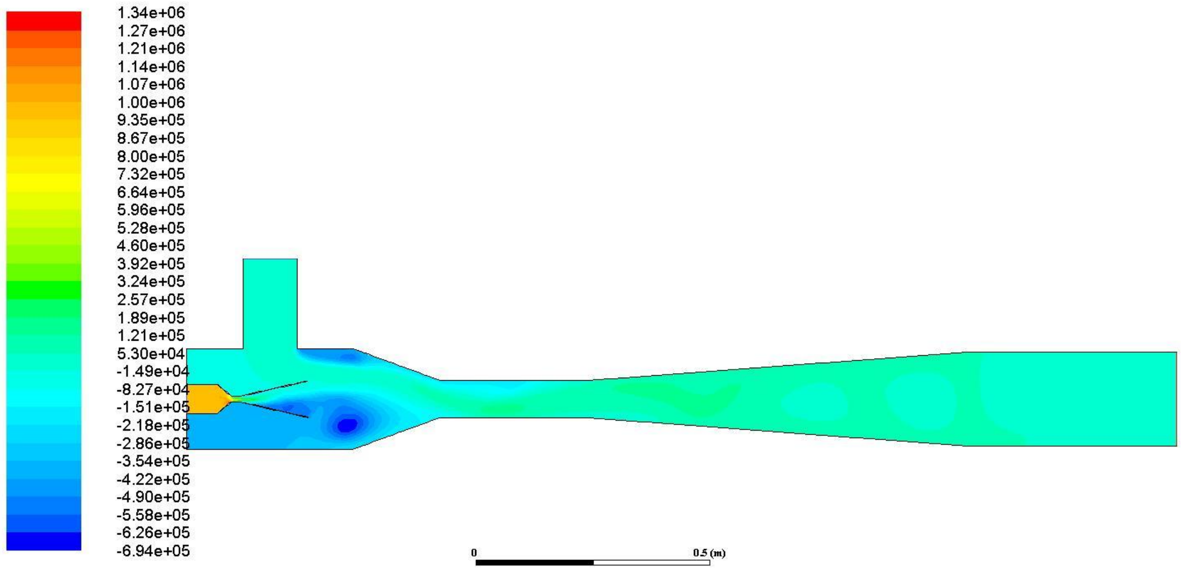

Figure 4-4: pressure dissipate on ejector with 10 bar inlet

The lowest suction produced is -45147.563 pas or -0.4515 bar

iii. **Motive Steam 15 Bar:** the values of suction pressure with motive inlet pressure 15 bar are given in Table (4-7) and Figure (4-5) explain the pressure dissipate on inlet of ejector when the motive pressure was 8 bar and less vacuum recorded.

Table (4-7): 15 bar motive steam simulation result

NO.	**THROAT LENGTH** (m)	**DIVERGENCE LENGTH** (m)	**Sum.** (m)	**Suction pres.** (Pascal)
1	0.3376	0.844	1.0972	-100288.66
2	0.5908	0.6752	1.1816	-49733.379

3	0.6752	0.5908	1.266	-26526.199
4	0.8440	0.5064	1.3504	-89615.797
5	1.0128	0.4220	1.4348	-60683.355
6	1.1816	0.3376	1.5183	-25609.387

The ejector with throat length (0.3376) and divergence length (0.844) has lesser value of suction pressure recorded than others.

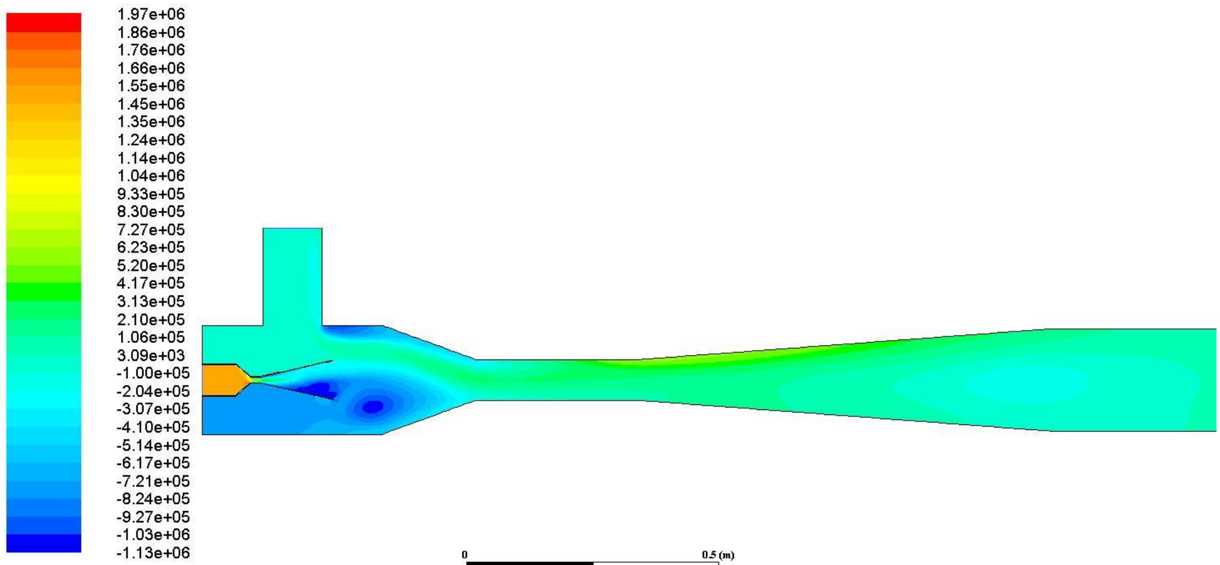

Figure 4-5: pressure dissipate on ejector with 15 bar inlet

The lowest suction produced is -100288.81 pas or -1.003 bar

CHAPTER V

5-1 Conclusion:

After studied the ejector and make modification need to get a satisfy vacuum, the flowing results are attained:

i. The specification of ejector found to be with throat length (0.3376m) and divergence length (0.844m).
ii. The optimum motive steam pressure for ejector motioned is 15 bar.
iii. The best vacuum for study found to be -100288.81 pas or -1.003 bar lesser than both typical ejector.

5-2 Recommendation:

After attain the results and to maintain or improve performance it recommend that:

i. To study the effect of rate of throat length to divergence length.
ii. To study the effects on evacuation to avoid it, such as cleanness of condenser tubes, condenser sealing and weather status.
iii. To study the effect of cooling water system such as chemical dosing.

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