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Performance of Venturi Scrubber

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Abstract:- In this article we have made an attempt to survey some results on performance of jet ejector. The literature is also reviewed to pursue further work in this area.
Key words- HEVS, EVS, jet ejector, performance of venture scrubber, characteristics of jet ejector, efficiency

I. INTRODUCTION

Jet ejectors have been successfully used for polluted gas cleaning application over last many decades due to their capability of handling gas containing pollutants such as vapor, gaseous, and solid/liquid aerosols up to 0.1 μm size. However they have inherent disadvantage of high pressure drop across the system which results in high fan/pump operating cost. But this disadvantage is compensated by their significantly less capital and maintenance costs compared to other wet scrubbers with comparable collection efficiencies. Since last six decades investigators have focused their attention to optimize the performance of venturi scrubbers. In this article we have made an attempt to describe the performance of scrubber based on literature.

Economopoulou and Harrison (2007), Viswanathan et al. (2005), Ravi et al. (2003), Gamisans et al. (2002), Ananthanarayanan and Viswanathan (1998), Singh et al. (1974) and Bhat et al. (1972) have investigated the performance of jet ejectors. A jet ejector when used as a scrubber is considered to have given optimum performance when its desired scrubbing efficiency is achieved at minimum pressure drop. Models to predict pressure drop and scrubbing efficiency are required for optimization of performance of jet ejector. Pressure drop and scrubbing efficiency are complex functions of gas velocity, liquid-to-gas ratio, ejector geometry (shape and number of nozzles, area ratio, throat diameter, throat length, projection ratio, angle of divergence and convergence), operating and suction pressure, properties of gas and liquid (temperature, concentration, diffusivity, viscosity, surface tension, etc.), reactivity of fluids, variation in composition of fluids, etc. Most of researchers have presented their data graphically in dimensionless form. The equations governing scrubbing efficiency are either empirical or based on dimensional analysis. Recently some investigators (Taheri and Mohebbi, 2008) tried to utilize modern technique like artificial neural networks using a genetic algorithm for predicting collection efficiency in venturi scrubbers. Many researchers applied CFD method to understand the hydrodynamics. It is common conclusion that CFD is an efficient tool for predicting the hydrodynamics and mass transfer characteristics of an ejector as it gives comparable result with experiments.

Venturi scrubbers are broadly classified into two groups viz. High Energy Venturi Scrubber (HEVS) and Ejector Venturi scrubber (EVS). As far as their performance is concerned HEVS may be differentiated from EVS as given in Table 1.

Table 1 : Performance of HEVS versus EVS

High Energy Venturi Scrubber (HEVS)	**Ejector Venturi Scrubber(EVS)**
Gas and liquid both are introduced in scrubber by external mechanical device.	Primary (Motive) fluid is ejected in venturi scrubber at high velocity by external mechanical device; another fluid is drawn in by kinetic energy of primary fluid.
L/G ratio is very low	L/G ratio is always high
Gas velocity in throat is dominant to break up liquid into droplets	Velocity of primary fluid at the discharge of nozzle/nozzles atomizes secondary fluid
Pressure drop and collection efficiency are studied as functions of operating conditions like L/G ratio, gas velocity at entry and at throat	Pressure drop and collection efficiency are studied as function of operating conditions like pressure ratio (ratio of operating pressure to suction pressure) and entrainment ratio (ratio of mass flow rates of entrained fluid to operating fluid)

Performance is studied with respect to design parameters like length, nozzle diameter, and throat aspect ratio (ratio of depth to width)	Performance is studied in terms of projection ratio(ratio of distance between nozzle end to commencement of throat) and area ratio (area of diffuser throat to area of nozzles)

II. PERFORMANCE OF HIGH ENERGY VENTURI SCRUBBER

The performance of a venturi scrubber is measured by consideration of its collection efficiency and pressure drop. There are number of models documented in the literature to predict the venturi scrubber efficiency. These models are used in optimizing and designing new scrubbers or predicting the effect of changes in operating conditions and dimensional variables of existing equipments on their performance. Models proposed by Crowder et al. (1981) and Goel and Hollands (1977) have reported the limitations of complex mathematical expressions and the need to estimate physical properties data. A summary of models that are more realistic and have utility in prediction of pressure drop and collection efficiency are reviewed and presented in Table 2.

Mathematical models to predict pressure drop

Several attempts have been made to predict pressure drop across venturi scrubbers theoretically. However, none of these models accurately predict pressure drops for a wide range of operating conditions. The main models reported in the literature are:

- Calvert's Model (1970)
- Boll's model (1973)
- Annular flow model (AFM) (Viswanathan et al., 1985)
- Boundary layer growth model (BLM) (Azzopardi et al., 1991)
- Full boundary layer model (Sun et al., 2003)

Many researchers tried to predict pressure drop separately for atomization section, throat section and diffuser section. Almost all have presented their findings graphically on the plot either pressure or pressure drop vs. axial distance. The nature of plots is found to be almost similar qualitatively but they differ quantitatively. The pressure drop increases slowly till the entry of throat and then it suddenly rises in throat. In the diffuser some pressure is recovered and curve starts falling. Typical plots are presented in Figure 1, 2 and 3.

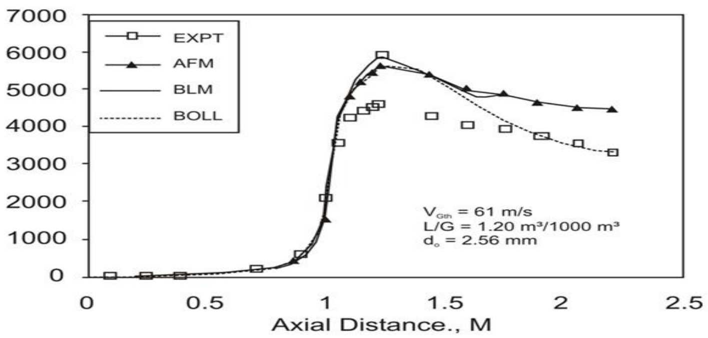


Figure 1: Comparison of axial pressure drop predicted by different models with experimental data (Vishwanathan et al., 2005)

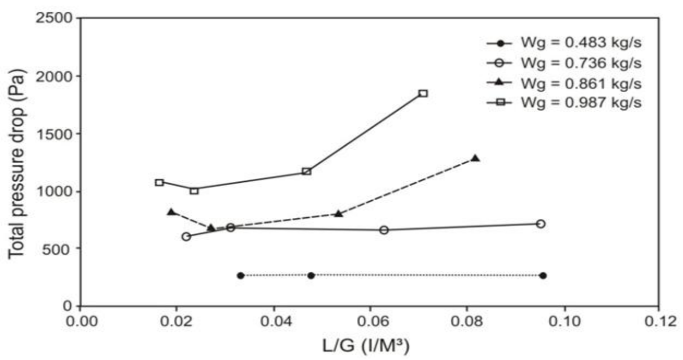


Figure 2 : Variation of the total pressure drop in the venturi with liquid to gas ratio and throat velocity. Liquid injected as a spray (Silva et al., 2009)

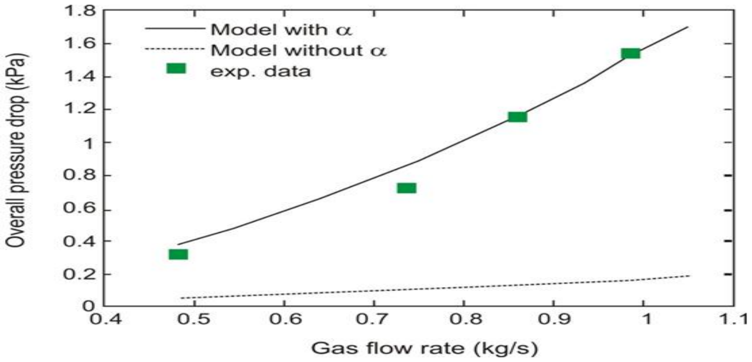


Figure 3 : Comparison of overall pressure drop predicted with and without correction factor α, experimental data of Silva et al. (2009) (Rahimi et al., 2011)

Collection efficiency

Jet ejector efficiency has been defined by researchers in different ways, like target efficiency, collection efficiency, overall efficiency and fractional efficiency (Mohebbi et al., 2003; Pulley 1997; Yung et al., 1977; Leith and Cooper, 1980; Boll 1973; Calvert 1970). The overall collection efficiency is defined as

- For particulate matter

$$Collection\ efficiency\ = \frac{the\ mass\ of\ the\ removed\ particulate}{inlet\ of\ the\ mass\ of\ total\ flow\ of\ particulate\ matter}$$

For gaseous pollutant: Taheri et al. (2008) defined collection efficiency (the extent of absorption) as

$$Collection\ efficiency\ (in\%) = \frac{P_i - P_f}{P_i - P_e} x100$$

where P_i P_f and P_e are the initial, final, and equilibrium partial pressure of gaseous pollutant in mm of H_g, respectively Collection efficiency have been reported with respect to gas/liquid ratio, gas and liquid flow

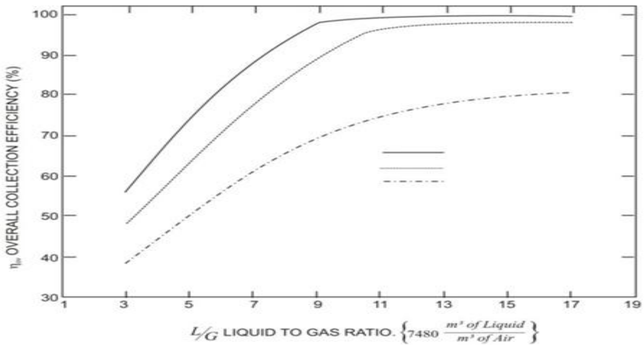


Figure 4 : Dependence of the overall collection efficiency of liquid gas ratio (Vishwanath et al., 1997).

rates , geometry of venturi scrubber like projection ratio P_R, length of throat, angle and length of convergent diffuser section and property of particulate/gas pollutants. Researchers have reported different empirical correlations to predict efficiency on the basis of different assumptions they have considered. The vast literature has been published on the subject. Table 2 is the summery of some of the earlier research. Typical graphical presentations are shown in Figure 4, 5, 6 and 7.

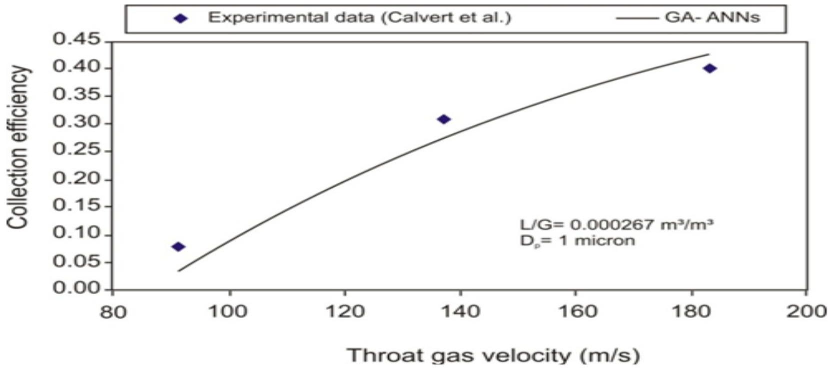


Figure 5 : The effect of throat gas velocity on the collection efficiency in venturi scrubber (GA–ANN no. 1). (Taheri et al., 2008)

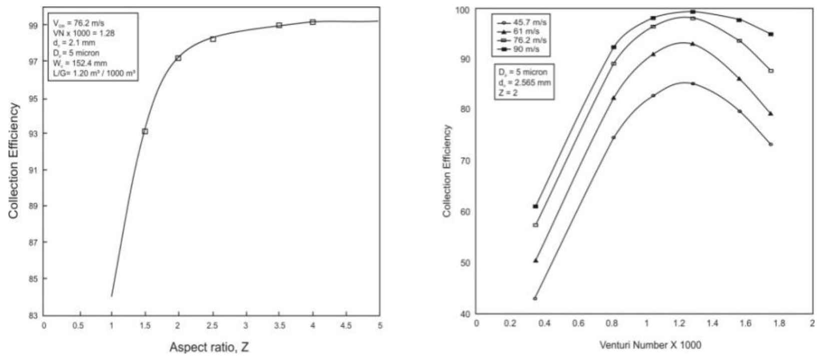
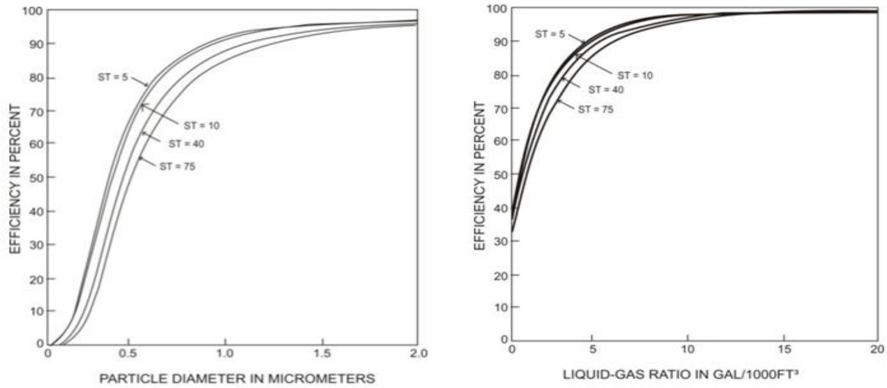


Figure 6 : Effect of variation in venturi number and aspect ratio on collection efficiency for a constant venturi number. (Ananthanarayanan and Vishwanathan, 1998)



(A) (B)

Figure 7 : Efficiency as a function of (A) particle diameter (B) liquid to gas ratio with liquid surface tension as a variable. (Ott el al., 1987)

Ott et al. (1987) developed a model studying the effect of surface tension on performance of venturi scrubber. They examined the effect of liquid surface tension on droplet size and on particle penetration into the droplet. (Figure 7A and B)

Economopoulou and Harrison (2007) developed graphical tools for estimating the overall collection efficiency of venturi scrubbers under the specified design and operating conditions based on the well-established theoretical formulations of Calvert (1970) and Yung et al. (1978).

Taheri et al. (2010) simulated gas absorption in a venturi scrubber and developed a three-dimensional mathematical model, based on a non uniform droplet concentration distribution. They validated their model with the experimental data reported by Johnstone et al. (1954) and Wen and Fan, (1975) for SO_2 removal by using alkaline solution and H_2O. They used

Lagrangian approach for water droplet movement. Yung et al. (1978) and Crowder et al. (1981) have developed mathematical models to study different parameters of high energy venturi scrubbers

Table 2 : References for pressure drop and collection efficiency of HEVS

Sr. No	**Reference**	**Type of scrubber studied**	**performance in terms of Δp / η**	**Parameters studied having effect on scrubber perf.**	**Property of Pollutant(particle diameters)**	**Ventury scrubber Geometry**	**Specific findings**
1	Boll R.H. (1973) (1974)	R, H	p, Δp,η	G,L/G,Vg, Vgth	diameter of particle, drag coefficient, separation number	Diameter and length of throat	presented math. model that can be used to optimise design and operating conditions in specific applications and to predict drop size.

2	Yung et al (1978)	HEVS	η	drop mdiameter	_	throat length	model to predict η.
3	Crowder, J.W. et al.,(1981) (1982)	HEVS	p, Δp, angle of conv./div. throat length	L,G,L/G, V_{gth}	_	_	to optimise converging angle, throat length and diverging angle.
4	Crowder, J.W. et al.,(1982)	HEVS/PA	η	_	_	contactor length	prediction of minimum contacactor length
5	Ott Robert M. el al. (1987)	HEVS	η	G,L/G,	diameter of particle	surface tension	new model presented
6	Monabbati et al.(1989)	HEVS	η	L,G	diameter of particle		new model presented
7	Viswanathan (1997)	HEVS/R	η	G,L/G,V_g, V_{gth}	diameter of particle	nozzel dia	The two-phase, two-component,annular flow unit was predicted.
8	R.A.Pulley (1997)	HEVS/PA /WA	η,Δp	L,G,L/G, V_{gth},	particle size	throat length	new model based on inertial mechanism.
9	Ananthanarayanan N V et al. (1998)	HEVS/R	η	G,L/G,V_g	diameter of particle	V_N, d_J, throat aspect ratio	V_N 1.0-1.5X 10^{-3} offers maximum efficiency.
10	Viswanathan (1998)	HEVS/PA /R	p, Δp	G,L/G,V_{gth}, liquid film		orifice diameter	a correlation has been developed to predict the liquid film thickness throughout the scrubber length.
11	Ananthanarayanan et al. (1999)	HEVS/PA /C	η	G,L/G,V_{gth}	diameter of particle	V_N	as $V_N$is independent of G, it is desirable to operate the scrubber in the range of 70-100 m/s to achieve maximum liquid utilization and collection efficiency
12	H. Sun et al.(2003)	HEVS/PA /WA	Δp	L,G	initial drop zize	orifice diameter	full boundary layer model has been presented.
13	Ravi G. et al.(2003)	HEVS/PA /R	η	L/G,V_{gth}	_	nozzle configuration	three-dimensional model for the collection η with the NSGA algorithm
14	Mohebbi et al.(2003)	orifice scrubber	η,Δp	particle diameter	_		particle-source-in-cell model proposed

15	Rahimi et al. (2004) and (2011)	HEVS/PA /R/C	Δp	L,G,L/G, V_{gth}		entering gas temperature/humidity	new concept of presentation of Δp in terms of number of throat velocity heads
16	Viswanathan et al.(2005)	HEVS/PA /R/C	p, Δp	G,L/G,V_{gth}	–	V_N,nozzel dia,throat aspect ratio	proposed- improved, ease,versatile and comprehensive algorithm to optimize scrubber performance which takes into account non-uniform liquid distribution
17	M.Taheri et al.(2008)	HEVS/PA /R	η	L/G,$V_{gth,}$	diameter of particle	Diameter of throat	GA–ANN model is more efficient it has less AAPD
18	Silva et al. (2009)	HEVS/PA /WE/R/C	Δp	L,G,L/G, V_{gth}	–	–	model is inadequate for the prediction of pressure drop in the throat region
19	Nasseh et al.(2009)	HEVS/PA /R	Δp	L/G,$V_{gth,}$	–	throat length	a neural network optimized by GA for predicting pressure drop in venturi scrubber.

Venturi type-HEVS-High energy Venturi Scrubber, EV-Ejector venturi scrubbere, R-Rectangular cross section, C-circular crosssection, PA-Pease-Antony type,WA-Wetted Aproach G-gas velocity Vgth-Gas velocity at throat,L/G -Liquid to gas ratio (m3/m3),η-collection efficiency, R- Rectangular cross section,C-circular crosssection, V_N – Venturi number, AAPD - Average Absoulte Percent Deviation, GA = Genetic Algorithm, NSGA = Nondominated sorting genetic algorithm.

III. JET EJECTORS

The application of jet ejector as vacuum producing device and as jet pump is well known (Gamisans et al., 2004; Govatos, 1981; Cunningham, 1974; Cunningham and Dopkin, 1974; Bonnington, 1956, 1960, 1964; Bonnington and King, 1972; Blenke et al., 1963; Kroll, 1947). With the fast growth of chemical process industry, their use as entraining and pumping device to handle corrosive fluids, slurries, fumes and dust laden gasses has increased. Their use as mass transfer equipment for liquid-liquid extraction, gas absorption, gas stripping, slurry reaction like hydrogenation, oxidation, chlorination, fermentation, etc. has increased. Due to increasing interest in the usage of jet ejectors, numbers of investigators have attempted to optimize their performance. (Das and Biswas, 2006; Gamisans et al., 2004; Gamisans et al., 2002; Dasappa et al., 1993; Mukharjee et al., 1988,1981; Radhakrishnan and Mitra, 1984; Pal et al., 1980; Biswas et al., 1977, 1975; Acharjee et al., 1975; Singh et al., 1974; Bhat et al., 1972; Davis et al., 1967; Mitra and Roy,1964; Mitra et al., 1963 ; Davis (1963).

Working of jet ejector

A jet ejector is a device in which suction, mixing and dispersion of secondary fluid is done by utilizing the kinetic energy of a motive (primary) fluid. Das and Biswas (2006) stated that when jet ejectors are used as a device for contacting gas–liquid , the secondary fluid may be dispersed by the shearing action of the high velocity motive fluid or the motive fluid itself may get dispersed when it is arrested by a secondary fluid. Figure 8 shows the typical ejector system in which the jet of primary fluid issuing out of a nozzle creates a low pressure region around it. The pressure differential between the entry point of the secondary fluid and the nozzle tip provides the driving force for entrainment of the secondary fluid. Two principal flow regimes in ejectors are coaxial-flow and froth-flow. The coaxial-flow constitutes a central core of primary fluid with secondary fluid flowing in the annular region formed between the jet of primary liquid and ejector. Froth-flow regime is a co-current flow of fluids with one phase completely dispersed in the other. Witte (1969) termed the phenomenon of change from coaxial-flow to froth-flow as mixing shock. Here a part of the kinetic energy of the flow is dissipated in the shock creating the

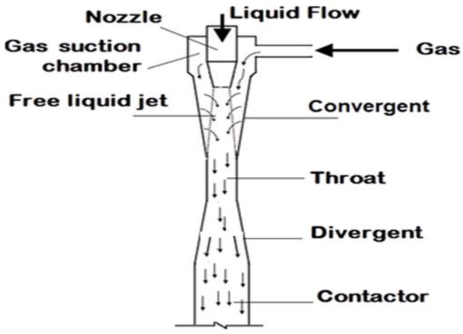


Figure 8 : Typical gas-liquid jet ejector

gas-liquid dispersion. The mixing shock results into generation of small bubbles and consequently creation of high interfacial area (~ 2000m^2/m^3). Ejectors thus, give superior gas-liquid mass transfer rates and higher rates of reaction as compared to conventional gas-liquid contacting equipments like stirred tanks, bubble columns, packed columns, etc. Yadav and Patwardhan (2008) stated that there could be diverse objectives for ejector design depending on application as follows:

(a) To get large entrainment of the secondary fluid.
(b) To produce intense mixing between the two fluids.
(c) To pump fluids from a region of low pressure to a region of high pressure.

Geometry of ejector

The significant parts of an ejector are (Refer Figure 9) primary fluid inlet, suction chamber, secondary fluid inlet, converging section, throat or mixing zone, diverging section or diffuser. The ejector may be specified by denoting nozzle diameter (D_N), throat diameter (D_T), diameter

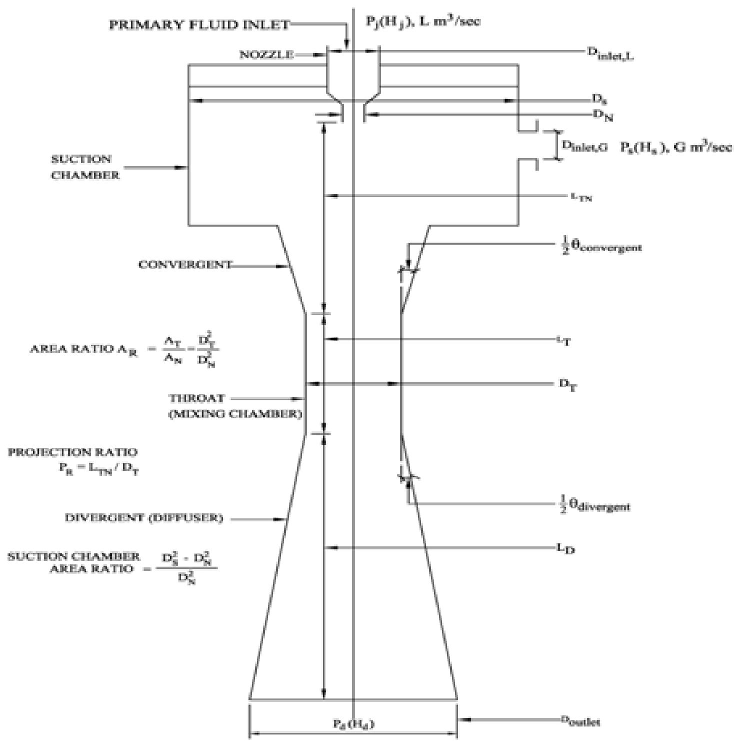


Figure 9 : Schematic diagram showing geometry of an ejector

of suction chamber (D_S), length of throat (L_T), length of diffuser (L_D), distance between nozzle to commencement of throat (L_{TN}), angle of converging sections $(\Theta_{convergent})$ and angle of diverging sections $(\Theta_{divergent})$. Performance of the ejectors has been studied in terms of (a) area ratio $(A_R = A_T/A_N)$, i.e., area of throat/area of nozzle, (b) throat aspect ratio (L_T/D_T), i.e., length of throat/diameter of throat, (c) projection ratio $(P_R = L_{TN}/D_T)$, i.e., distance between nozzle tip to the commencement of throat / diameter of throat and (e) suction chamber area ratio $[A_S/A_N = (D_S^2 - D_N^2)/D_N^2]$.

Dutta and Raghavan (1987) studied and compared the performance of jet ejectors with and without venturi and throat**.** Similarly Gamisans et al. (2002) studied jet ejector without diffuser. Both of them concluded that the jet ejectors without diffuser or throat are less effective compared to ejector with them.

Many researchers have studied the mass transfer characteristics and performance of the jet ejectors followed by contactors, draft tube, packed column or bubble column (Li and Li, 2011; Rahman et al., 2010; Balamurugan et al., 2008, 2007; Utomo et al., 2008; Mandal, 2010; Mandal et al., 2005; 2004, 2003a, 2003b; Havelka et al., 2000, 1997; Dutta and Raghavan, 1987; Ogawa et al., 1983; Mitchell, 1981; Biswas et al., 1977). All have similar conclusion that there is less mass transfer coefficient in the extended portion compared to that in the ejector itself.

Effect of ejector geometry

Das and Biswas (2006) reported that the efficient functioning of an ejector depends on the design of the suction chamber, the mixing throat, the divergent diffuser and the forcing nozzle. Besides, the relative dimensions of the various parts of the ejector, the factors such as shape of the entrance to the parallel throat, angle of divergence and the projection ratio are also important factors to be considered.

Different investigators have studied the effect of geometry of jet ejector like area ratio, angle of convergence and divergence, projection ratio, shape of entry of convergent section, length of throat which are compiled in Table 3.(Yadav and Patwardhan, 2008).

Area ratio (A_R**)**

The area ratio is defined as the ratio of area of throat (A_T) to area of the nozzle

$$A_R = \frac{A_T}{A_N} = \left(\frac{D_T}{D_N}\right)^2$$

Bonigton (1964) studied the effect of changing the diameter ratio i.e. ratio of nozzle diameter to throat diameter (D_N / D_T) instead of area ratio of the jet ejector performance. Acharjee et al. (1975), Singh et al. (1974), Bhat et al. (1972) and Mitra et al. (1963) studied the effect of area ratio on Mass ratio M_R (ratio of mass of driving fluid to the entrained fluid). It can be concluded from these studies that as the area ratio is increased the entrainment ratio also increases. But at the higher area ratio the increase in entrainment ratio becomes less. A typical correlation is shown in Figure 10.

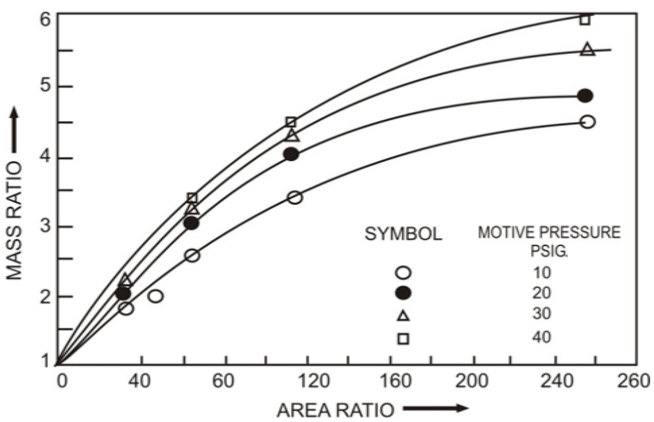


Figure 10 : Effect of area ratio on mass ratio for water-water system

(Singh et al., 1974)

Projection ratio

The projection ratio (P_R) is defined as the ratio of the distance between the injecting nozzle to the commencement of throat (L_{TN}) to diameter of throat (D_T)

$$P_R = \frac{L_{TN}}{D_T}$$

A typical plot of M_R vs. P_R is presented in Figure 11. It is observed that as P_R rises the entrainment ratio is not much effected but at definite value of P_R, the M_R, rises suddenly and again falls to previous value. Thus P_R at which it draws maximum entrained fluid is considered to be optimum. Biswas et al. (1975), Acharjee et al. (1975) and Devis et al. (1967) had similarly observed that at P_R around 1.10 is optimum. Singh et al. (1974) in their research study observed optimum P_R as around 0.5. It has been suggested that the optimum P_R is influenced by geometry of entrance to the mixing tube. Table 3 shows that the optimum value of P_R by the different investigators is different. Yadav et al., (2008) utilized computational fluid dynamics (CFD) to study the role of P_R , angle of converging section and diameter of suction chamber. They studied the effect of P_R (0, 1.5, 5, 10 and 14.5) on entrainment, pressure profile along the axis of ejector power

Table 3 : Geometrical parameters of ejectors used by deferent investigators
(As ascending order of throat aspect ratio)

Throat aspect ratio (L_T/D_T)	Area ratio $(D_T/D_N)^2$	Entry to the throat	Angle of converging section (deg)	Angle of diverging section (deg)	Projection ratio (L_{TN}/D_T)	References
0	2.37 – 2.66	Conical	--	--	Pitch 1.1, 1.5, 2*	Panchal et al. (1991)
0	4	Well rounded	2.5 – 9	4.85	0 – 14.5	Yadav & Patwardhan (2008)
0	3.7-25.1	Conical	28	10	8.9	Bhat et al. (1972)
0	33.8--113.8	Conical	--	6.4	--	Zahradnik et al. (1982)
0 – 4	1.1-6.45	Well rounded	--	--	2 – 4	Balamurugam et al. (2008)
0--16	1.8-10.2	Conical or bell shaped	12	5.0	3.0*	Bhutada and Pangarkar (1987)
0.5--1.3	-	Conical	--	--	--	Gamisans et al. (2004)
1-- 6	5.6--10.0	Well rounded.		10	1.1-6.8	Sriveerakul et al. (2007)
1.1	-	Conical	12	2	2.17	Li and Christofides (2005)
1.8	1.5--3.5	Conical	17.35	9.5	--	Moresi et al. (1983)
2--10	2.1--9.0	Conical	--	3.0	--	Cramers and Beenackers (2001)
2.8	7.66-16	Conical	--	15	--	Dutta & Raghwan (1987)
2.95 – 7	4.48 – 40	Well rounded	--	--	--	Appusamy et al. (2008)
3.5	4.0	Conical	--	2.0	--	Ben Ebrahim et al. (1984)
4-10	6.69	--	--	3.5	--	Utomo et al. (2008)
4.8	6.76 – 18.7	Conical	--	--	--	Rahman (2010)
5.0	2.5	Conical	10	7	5.0	Rusly et al. (2005)
5--20	3.24	Conical	--	7	--	Havelka et al. (1997)
6	1.4--12.8	Well rounded	--	7	1.9*	Biswas et al. (1975)
6	9.3	Well rounded	--	7	2	Agrawal (1999)
6	-	-	-	8	1.0	Fernandez (2001)
6.5	9.9 – 39	Well rounded	--	7	--	Mukherjee et al. (2007)
7	-	Well rounded	20-25	4--10	0.5—5	Kroll (1947)
7	21.6-247	Well rounded	--	10	1.9*	Davies et al. (1967)
7	29.3--169.8	Conical	-	7	--	Kundu et al. (1994)
7	1	--	--	9	--	Raghuram (2009)
7.5	2.0--25.5	Well rounded	--	5	0.4--0.9*	Henzler (1983)
7.76	15.5--59.5	--	--	8.6	--	Das and Biswas (2006)
8	5.4-50.4	Well rounded	--	10	2.0*	Acharjee et al. (1975)
9.6	7.4--22.5	Well rounded	--	7	--	Majumder et al. (2005)
9.7	5.6--14.4	Well rounded	--	9.1	--	Mandal et al. (2005b)
9.7	10.0	Well rounded	--	9.1	--	Mandal et al. (2005a)
9.7	29.3--169.8	Well rounded	--	7	--	Kundu et al. (1995)
10	-	Conical	--	7	2.72	Elgozali et al. (2002)
10.8	1-50.6	Well rounded	-	7	--	Mukherjee et al. (1988)
12.3—32.4	2.2-6.5	Well rounded	--	4.0	3.0*	Cunningham and Dopkin (1974b)

*investigators suggested as the optimum values

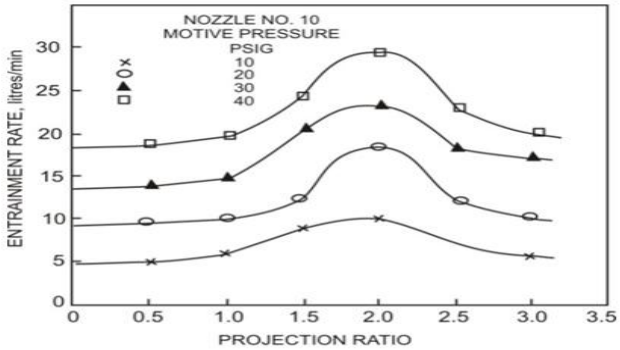


Figure11: Variation of entrainment of air with projection ratio of water-air system (Acharjee et al. 1975)

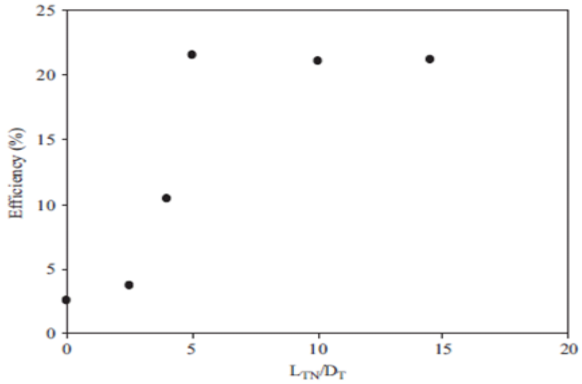


Figure12: Effect of projection ratio *(L_{TN}/D_T)* on energy efficiency (Yadav and Patwardhan, 2008)

efficiency. They concluded that the rate of entrainment and power efficiency increases as the projection ratio increases that is because of the fact when one increases the P_R it leads to the reduction in the generation of radial flow. However beyond $P_R > 5$ negligible amount of radial flow is generated and hence the rate of entrainment and energy efficiency remain constant. Hence it may be considered that the optimum projection ratio is 5 (Figure 12).

Diameter of suction chamber (D_S)

Though cross sectional area of the suction chamber is important parameter which effects the

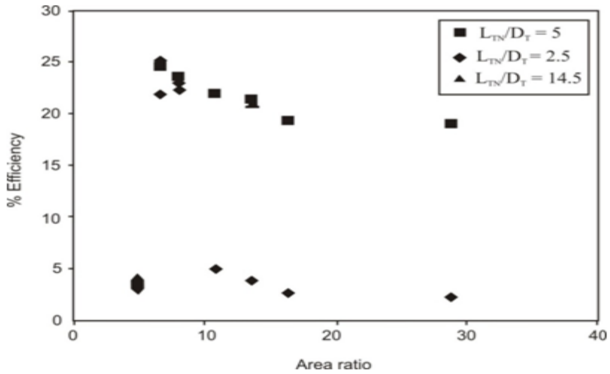


Figure 13: Effect of area ratio $(D_S^2 - D_N^2)/(D_N^2)$ on efficiency of ejectors for different values of projection ratio (Yadav and Patwardhan, 2008)

Performance of venturi, it has not been given the necessary attention. Yadav and Patwardhan (2008) studied the effect of diameter of suction chamber. To study the effect of suction chamber diameter they defined suction chamber area $ratio$ $(A_S \,/\, A_N)$ as

$$Suction\ chamber\ Area\ ratio = \left(\frac{A_S}{A_N}\right) = \frac{D_S^2 - D_N^2}{D_N^2}$$

They concluded that maximum power efficiency (20 to 25%) is obtained for $(D_S^2 - D_N^2)/(D_N^2)$ =6.6 and for $(D_S^2 - D_N^2)/(D_N^2)$ > 13.6 it remain constant. (Refer Figure 13)

Effect of angle of convergent section and divergent section

It can be seen from Table 3 that numbers of investigators have worked to find optimum angle of convergence and divergence. Yadav and Patwardhan (2008) studied the effect of angle of convergence on entrainment and efficiency. In Figure 14 entrainment for different angles: 2.5˚, 10˚, 30˚ and 90˚ has been shown. It can be seen that the rate of entrainment is low for θ = 2.5˚. It increases with increase in θ and attains a maximum value for θ =10˚. Further increase in θ results in decrease in the rate of entrainment of the secondary fluid. Similarly their study shows that the

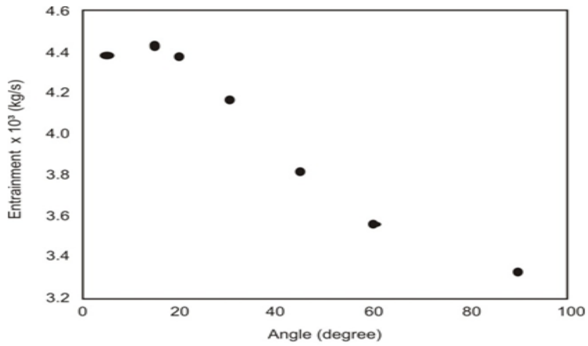


Figure 14 : Effect of angle of converging section (θ) on rate of entrainment (Yadav and Patwardhan, 2008)

largest pressure driving force is generated for θ = 10˚ and it results in the highest entrainment for this case. With increase in θ beyond 10˚ the pressure driving force was observed to reduce and it results in decrease in the rate of entrainment. They also showed that highest efficiency is obtained at θ =10˚and larger values of θ results in poor energy efficiency. Thus, they suggested for obtaining maximum entrainment the angle of convergent may be kept between 5˚–15˚. The angle of divergent section has been kept between 7˚ to maximum 10˚ by many of the investigators.

Mathematical models

Utomo et al. (2008) developed three dimensional CFD model to investigate mass transfer characteristics. They varied the gas-liquid flow ratio in the range of 0.2 to 1.2 and the length to diameter ratio of mixing tube (L_{TN} / D_{MT}) from 4 to 10. Their CFD studies show that at $L_{TN}/D_M = 5.5$, the volumetric mass transfer coefficient increases with respect to gas flow rate. They observed that at $L_{TN}/D_M = 4$, the graph of volumetric mass transfer coefficient vs gas-liquid flow rate ratio reaches the maximum at gas-liquid flow rate ratio of 0.6. A remarkable observation in their study was that volumetric mass transfer coefficient decreases with the increase of mixing tube length. They validated results obtained from CFD with the experimental result (configuration of ejector has a mixing tube diameter of 22 mm and diffuser outlet diameter of 40 mm, diffuser angle of 3.5 and a draft tube length of 100 mm.). The mixing tube lengths are varied between 88 and 220 mm with the nozzle diameter of 8.5 mm.

Kandakure et al. (2005) developed a CFD model to understand the hydrodynamic characteristics of ejectors. They varied the value of the slip velocity between the phases for validation keeping nozzle velocity constant (at different height to diameter ratio of throat) to validate the model. They found that when the slip velocity is made 13% of the axial water velocity, it matches the experimental data very well. They found that the predicted air entrainment is the maximum for the ejector with height to diameter ratio of throat equal to zero and the area ratio of 4. They justified that the CFD simulations eliminate all such empiricism.

Kim et al. (2007) studied rectangular bubble column ($0.22 \times 0.26 \times 1.3m$) with a horizontal flow ejector. They investigated the effect of the ejector geometry i.e. nozzle diameter and mixing chamber diameter and the operating conditions like liquid flow rate and liquid level in rectangular column, on the hydrodynamic characteristics. They established that the gas holdup increases with increasing liquid flow rate and decreases with increasing level of liquid in the rectangular column. They applied the multiphase CFD simulation with the mixture model and found that the gas entrainment rate increases with increasing liquid flow rate contrary to this the gas suction rate decreases with increasing nozzle diameter and the liquid level in the rectangular column. The predicted values obtained from CFD simulation were compared with the experimental data, which were well matching.

Li and Li (2011) investigated the entrainment behavior and performance of gas–liquid ejectors using different software and computational technique like Computational Fluid Dynamics (CFD) and validated with experimental data over a wide range of operating conditions for ejector with different configurations.

IV. PARAMETERS OTHER THAN GEOMETRY OF THE EJECTOR

Many investigators (Gamisans et al., 2004, Gamisans et al., 2002, Ebrahim et al.1984; Bhutada, and Pangarkar, 1987; Acharjee et al., 1975, Singh et al., 1974; Bhat et al., 1972; Davis et al., 1967; Mitra and Roy 1964; and Mitra et al., 1963) studied effect of mass ratio (M_R) as a function of motive pressure, suction pressure, separator pressure, pressure drop, A_R, P_R , Reynold's number, Euler's number etc. Some of investigators (Mitra et al., 1963; Bonington 1964) studied the effect of head ratio on ejector performance, head ratio is defined as:

$$Head\ ratio = \frac{Head\ generated\ by\ suction\ fluid}{Head\ lost\ by\ driving\ fluid} = \frac{H_D - H_s}{H_j - H_s}$$

where H_D = pressure head at discharge of ejector, m; H_s = pressure head at suction of ejector, m; and H_j = operating pressure, m.

The empirical equations to predict mass ratio (M_R) from dimensionless analysis given by various authors are summarized in Table 4. Many investigators (Ebrahim et al.1984; Acharjee et al., 1975; Bhat et al., 1972; Biswas and Mitra, 1981: ,Henzler, 1983) have developed correlations to determine mass ratio (M_R) by theoretical analysis.

Table 4 : Mass ratio correlations from dimensionless analysis given by various authors

Authors	System Primary-Secondary	Geometry and range investigated	Mass ratio correlation
UPWARD FLOW			
Davies et al. (1967)	Air-water	$D_N = 0.00808–0.002676\ m$, $D_T = 0.0127\ m,\ H_T = 0.0889\ m$, $(D_N/D_T) = 0.009–0.2107$, $D_C = 0.0635\ m, H_C = 1.219m$	$M_r = k\left(\frac{\mu_m}{D_{N\rho m}\,U_m}\right)^{0.76} (A_r)^{0.4} \left(\frac{g\mu_e^4}{\rho_e\sigma_e^3}\right)^{-0.04} \left(\frac{\rho_e-\rho}{\rho_e}\right)^{0.63}$
Acharjee et al. (1975)	Water/ glycerin/ kerosene-Air	Flow—upward: $D_N = 0.00178–0.0055\ m$, $D_T = 0.0127\ m, H_T = 0.1016\ m$, $D_N/D_T = 0.14–0.433$	$M_r = 5.2\times10^{-4}\left(\frac{\Delta P}{\rho_e U_e^2}\right)^{-0.305} (A_r)^{0.68}\left(\frac{g\mu_m^4}{\rho_m\sigma_m^3}\right)^{-0.305}$
DOWNWARD FLOW			
Ben Ebrahim et al. (1984)	Water/ mono ethylene glycol-Air	$D_N = 0.0025\ m$, $D_T = 0.005\ m$, $H_T = 0.0175, D_N/D_T = 0.5$, $H_C = 1\ m, D_C = 0.01m$	$M_r = 43.86\times10^{-3}\left(\frac{\Delta P}{\rho_e U_e^2}\right)^{-0.38}\left(\frac{g\mu_m^4}{\rho_m\sigma_m^3}\right)^{-0.01}$
Dutta & Raghvan (1987)	Water-Air	$D_N = 0.0045, 0.0065\ m$, $D_T = 0.018\ m$, $D_C = 0.040m$	$M_r = 2.4\times10^{-3}\left(\frac{\Delta P}{\rho_e U_e^2}\right)^{-0.82}\left(\frac{g\mu_m^4}{\rho_m\sigma_m^3}\right)^{-0.01}$
Bhutada & Pangarkar (1987)	Water-Air	$D_N = 0.005, 0.008, 0.01, 0.012\ m$, $D_T = 0.016, 0.0159\ m$, $D_N/D_T = 1.6–3.2$	$M_r = x\left(\frac{\Delta P}{\rho_e U_e^2}\right)^{y}(A_r)^z$; $x = 5.58\times10^{-4}\ to\ 9.67\times10^{-4}$; $y = -0.135\ to -0.202$; $z = 0.07 - 0.224$

HORIZONTAL FLOW			
Bhat et al. (1975)	Water/ glycerin/ kerosene-Air	$D_N = 0.0019–0.00449\ m,$ $D_T = 0.00925\ m,$ $D_N/D_T = 0.2–0.48,$ $H_T = 0, H_C = 1.1m$ $D_C = 0.0254\ m,$	$M_r = 8.5 \times 10^{-2} \left(\frac{\Delta P}{\rho_e U_e^2}\right)^{-0.3}$ $(A_r)^{0.46} \left(\frac{g\mu_m^4}{\rho_m \sigma_m^3}\right)^{-0.02}$
Singh et al. (1974)	Water-Water	$D_N = 0.00159, 0.00238,$ $0.003175, 0.00397,$ $0.00437\ m$ $D_T = 0.025\ m$ $D_N/D_T = 0.0625\ to\ 0.17$	$M_R = 3.2 \times 10^{-2}(Re_f)^{0.25}$ $(A_R)^{0.70}((g_c\Delta p)/(\rho_s u_s^2))^{-0.38}$
*M*r- mass ratio , μ- viscosity (kg/ms), *g*- acceleration due to gravity (m/s^2), *U*- velocity of fluid as denoted by subscript (m/s), *H*c- height of ejector (m),*H*D- height of diffuser (m),*H*T throat height (m), *D*C dia. of colum (m),ρ- density of mixture (kg/m^3),σ- surface tension (kg/s^2), (Δ*P*) pressure drop (N/m^2)			

Bonington (1964) published a plot of power efficiency vs head ratio with diameter ratio as parameter. As per their co relation the maximum efficiency achieved is around 33% at head ratio 4 and diameter ratio (ratio of diameter of nozzle to throat diameter) 0.52. Similar studies have also been done by Yadav and Patwardhan (2008), Gamisans et al. (2004), Cunningham (1974) and Blenke et al. (1963).
Yadav and Patwardhan (2008) defined Energy efficiency of ejector as

$$\%\eta = \frac{power\ imparted\ to\ the\ secondary\ fluid}{power\ of\ fluid\ coming\ out\ of\ the\ nozzle} \times 100$$

Where

$$Power\ of\ fluid\ coming\ out\ of\ the\ nozzel = (Power)_P = \frac{\pi}{8}\rho_P D_N^2 V_j^2$$

and

$$Power\ imparted\ to\ the\ secondary\ fluid = (Power)_S = (P_{outlet} - P_{throat})Q_S$$

where P_{outlet} is absolute pressure at diffuser outlet, Pa; P_{throat} is absolute pressure at throat, Pa; Q_S flowrate of secondary fluid, m^3/s; ρ_P is density of the primary fluid, kg/m^3 ; D_N, diameter of nozzle, m; V_j, velocity of primary fluid at outlet of nozzle.

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