A STUDY OF WATER JET PUMPS

by

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Notations - (General Theory)

P - Pressure

Z - Elevation

V - Velocity

Q - flow rate, gpm

Q_1 = Primary flow rate

Q_2 = Secondary flow rate

$Q_t = Q_1 + Q_2$

A - Area

h - Total head = $P/_{\gamma} + V^2/2g + Z$

f - Friction factor in $h_L = \frac{fL}{D} \cdot \frac{V^2}{2g} = K.\frac{V^2}{2g}$

η - Efficiency (η = MN)

γ - Unit weight of liquid

M - Q_2/Q_1

N - Head ratio $(h_d - h_2) \ / \ (h_1 - h_d)$

R - Ratio of area of nozzle tip to the mixing-tube area

A_j/A_t

K - Resistance factor in pipe lines

Subscripts

1 - a point in the drive line

2 - a point in the suction line

d - a point in the discharge line

j - the tip of the nozzle

s - the annular area surrounding the tip of the nozzle t - the throat of the mixing chamber

a - the transverse section at the tip of the nozzle

b - the discharge end of the mixing chamber

(Optimum Design Parameters)

d - nozzle tip diameter, in.

H - Head = $\frac{\text{Total Pressure}}{\text{Specific weight}}$, ft

H_1 = inlet primary head to ejector

H_2 = secondary flow inlet head

H_d = ejector discharge head

ℓ - mixing tube length, in.

s - spacing between nozzle tip and mixing tube, in.

t - mixing tube diameter, in.

α - angle of mixing tube entrance, deg.

β - diffuser cone angle, deg.

θ - nozzle external angle

(Jet Pump System Characteristics)

D_1 - inlet well-pipe diameter, inc.

D_2 - discharge well-pipe diameter, in.

H_{L_d} - friction head loss in discharge well pipe, ft

H_{L_i} - friction head loss in inlet well pipe, ft

$H_{p_2} - H_{p_1} + \Delta H_p$

ΔH_p - pump total head

L - depth of the well

N_o - value of N for M = 0

N_∞ - value of N for $M = \infty$

V_n - velocity of flow through nozzle-tip area

INTRODUCTION

The jet pump is a device for pumping fluids by means of a high velocity jet of the same or another kind of fluid. The principle phenomenon involved is the transfer of some momentum from the high velocity fluid to the low velocity fluid during the mixing process, with the result that all fluid leaving the mixing tube has about the same velocity. The general type of pump operating on this principle is classified as a momentum pump. Special names have been developed for jet pumps applied to specific services, the names being ejector, injector, hydraulic compressor, etc.,.

The jet pump is applied to many pumping problems due to its low first cost, simplicity in operation and ability to mix thoroughly two fluids. On the other hand, the usual installation is limited to small power inputs since the efficiency of the pump is inherently low. Development of theory has been slow because the fluid flow equations are complicated particularly in case of compressible fluids. High velocity jets of steam are used to pump air. Or sometimes, the jets of high velocity air are used to pump air.

As was pointed out earlier, the general class includes injectors and steam jet air pumps, as well as water jet pumps, but the latter possesses the decided advantage of not involving either compressibility or heat transfer.

The water jet pump was first used by James Thomson about 1852, and the theory of pumping by jet action was developed in 1870 by Rankine. The low efficiency of this pump has limited the field of application to conditions in which the absence of valves or working parts and the small size of the unit are sufficient to offset the greater power requirement of this than of other possible types. This unfavorable circumstance has caused the water jet pump to receive less attention than it deserves. A number of writers including Jesse (1904), Lorenz (1910), Gibson (1924), LeConte (1926), and Bergeron (1928) (Ref. 7) have developed theoretical equations, but almost no experimental data for checking these equations were available.

This report has been designed to summarize particularly the operation, theory, design characteristics and the application of "water jet pumps" in which the water is used as a primary as well as secondary fluid. Optimum design parameters, the jet pump system characteristics and their use in design have been reported and discussed. Various advantages of jet pumps have been mentioned. Experimental verification was made and the results reported.

OPERATION

Basically, high energy primary fluid expands through a nozzle, mixed with the driven or the secondary fluid and imparts energy to the secondary stream. The combined flows usually are discharged from the ejector through a diffusing section.

An auxiliary pump is used to circulate the primary fluid at a predetermined pressure. This pressurized or energized fluid expands through the ejector - a main component of the water jet pump. The primary fluid while expanding through a nozzle, has a large amount of momentum. With this momentum it has an ability to drag the secondary fluid which is in the immediate vicinity created through the existence of the inlet ports in the pump body. Thus, the fluids at different levels of energy are mixed thoroughly in the mixing tube provided in the pump body. Here the fluids are expected to have become just alike in the aspects of their energy levels. In order that the thorough mixing can be done with the minimum of losses, the length of the mixing tube is determined by the well-known Prandtl's theory of turbulent mixing (Ref. 12).

The combined flows then enter the diffusing section known as the diffuser. Here, the flow at uniform velocity gets to its normal condition prevailing in the pipe lines. Mostly, conical diffusers are preferred which provide adequate mixing and the pressure rise to suit the flow conditions in the pipe lines.

The Fig. 1 and 1a (Ref. 13) illustrates the essential elements of a jet pump. The nozzle converts the potential energy of the drive or primary fluid into kinetic energy, the resulting high speed jet entrains the slow speed fluid by means of the mixing process, the combined fluid flows into the diffuser and passes out of the pump. The action is simple and no moving parts are required.

The following are the advantages of the jet pump:

1. They have no moving parts -- nothing to break or wear. Because there are no packing glands, no lubrication is required. These contribute to the low maintenance cost.
2. They are practically noiseless in operation.
3. Since there is no foundation or wiring necessary, installation cost is low.
4. Operation is simple and trouble-free.
5. The lift of the centrifugal pumps is limited to 15 feet practically, whereas jet pumps could be installed at any depth in the well and can lift water from a greater depth.
6. Combination of jet pump and centrifugal pump allows the installation of the motor and centrifugal pump in a convenient place - such as the basement or a small house on the top of the well.

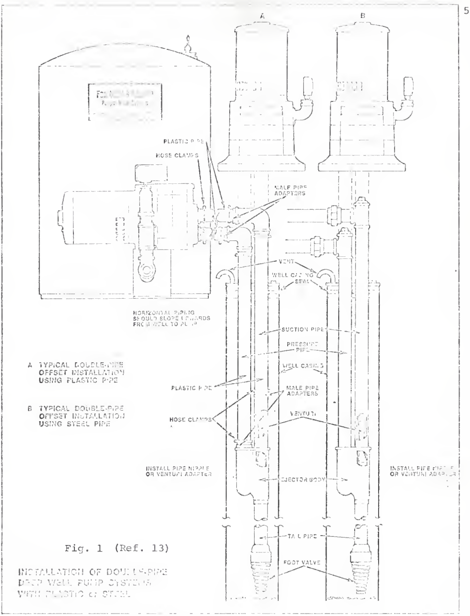
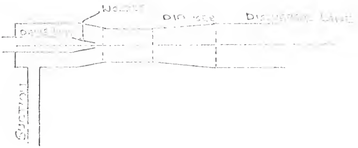


Fig. 1 (Ref. 13)

INSTALLATION OF DOUBLE-PIPE DEEP WELL PUMP SYSTEMS WITH PLASTIC or STEEL

Fig. 1a (Ref. 7)

Components of a Jet Pump



GENERAL THEORY

The basic principle of the jet pump is the transfer of momentum from one stream of fluid to another. The water jet pumps possess a special advantage of not involving either compressibility of fluid or any heat transfer as in the case of steam jet air pumps. The theory presented here is specifically applicable only to an incompressible fluid of small viscocity such as water. One treatment involves an energy balance with the computation of friction losses in the pipe lines, the impact loss and the pressure change in the mixing chamber. A second treatment involves the conservation of momentum theory and force balance. Both methods of analysis lead to basic equations of relationship of head and capacity and the overall efficiency.

The energy balance treatment equates the power supplied to the sum of the work done per second and the friction and the shock losses to give:

$$Q_1 \cdot \gamma \cdot (h_1 - h_d) = Q_2 \cdot \gamma \cdot (h_d - h_2) + K_j \cdot Q_1 \cdot \gamma \cdot V_j^2/2g + K_s \cdot Q_2 \cdot \gamma \cdot V_2^2/2g$$

$$+ (K_d + K_t)(Q_1 + Q_2) \cdot \gamma \cdot V_t^2/2g + Q_1 \cdot \gamma \cdot (V_j - V_t)^2/2g + Q_2 \cdot \gamma \cdot (V_t - V_s)^2/2g \quad \text{..............(1)}$$

where

$Q_1 \cdot \gamma \cdot (h_1 - h_d)$ = Power supplied

$Q_2 \cdot \gamma \cdot (h_d - h_2)$ = Useful work

$K_j \cdot Q_1 \cdot \gamma \cdot V_j^2/2g$ = Nozzle jet loss

$K_s \cdot Q_2 \cdot \gamma \cdot V_2^2/2g$ = Suction inlet loss

$(K_d + K_t)\ (Q_1 + Q_2) \cdot \gamma \cdot V_t^2/2g$ = Loss in discharge line

$Q_1 \cdot \gamma \cdot (V_j - V_t)^2/2g + Q_2 \cdot \gamma \cdot (V_t - V_s)^2/2g$ = Impact loss in mixing chamber

The energy loss per unit of time resulting from the mixing of two streams of different velocities in a chamber having parallel sides is shown by Lorenze (1910) (Ref. 4) to be

$$L = Q_1 \cdot \gamma \cdot (V_j - V_t)^2 / 2g + Q_2 \cdot \gamma \cdot (V_t - V_s)^2 / 2g \quad \text{.............(2)}$$

The loss of energy caused by frictional resistance at the boundaries of the mixing chamber is approximately $\gamma . K_t . (Q_1 + Q_2) . V_t^2/2g$, where V_t is the average velocity and K_t is a resistance factor computed in the same manner as for flow in the pipes. The frictional loss in the mixing chamber is an appreciable part of the total loss and the good agreement between the theoretical equation and the experimental results shows that this assumption is not very much in error.

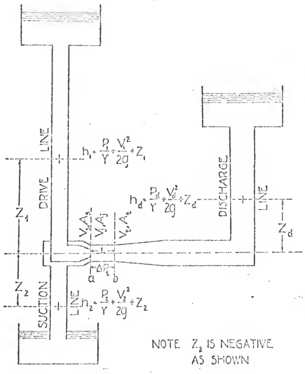
Referring to the Fig. 2 (Ref. 3), the flow equations for the three connecting pipelines are:

Drive Line $\quad h_1 = P_a/\gamma + 0 + (1 + K_j)\ V_j^2/2g$ ....(3)

Suction Line $\quad h_2 = P_a/\gamma + 0 + (1 + K_s) . V_s^2/2g$ ....(4)

Fig. 2 (Ref. 4)

Sketch of a Jet Pump



Discharge Line $h_3 = P_b/\gamma + V_t^2/2g + 0 = h_d + K_d.V_t^2/2g$ ..(5)

The continuity relations are the following:

$$\left.\begin{array}{lll} A_s+A_j=A_t & V_t=(Q_1+Q_2)/(A_s+A_j) & (Q_1+Q_2)/A_t \\ Q_1=A_j.V_j & Q_2=A_s.V_s & Q_1+Q_2=A_t.V_t \\ Q_2/Q_1=M & A_j/A_t=R & A_s/A_j=(1-R)/R \end{array}\right) \ldots(6)$$

In this set of equations, it is assumed that the nozzle extends to the intake of the mixing chamber and that the walls of the nozzle are very thin. Since the velocity heads are small in comparison with the pressure heads at the points at which the total heads are measured, the velocity head is based on the average velocity distribution. The computed friction loss is always based on the average velocity computed directly from the quantity and the area.

Subtracting (4) from (3) and inserting the relations given in (6) gives

$$h_1-h_2 = V_j^2/2g \;.\; \left[1+K_j-(1+K_s)M^2\left(\frac{R}{1-R}\right)^2\right] \qquad \ldots\ldots\ldots\ldots.(7)$$

Inserting this equation in (6) and substituting from (4) gives a result which can be simplified to

$$N = (1-T)/(T+M) = (h_d-h_2)/(h_1-h_d) \qquad \ldots\ldots\ldots\ldots.(8)$$

where

$$T = \frac{1+K_j+(1+K_s)M^3\left(\frac{R}{1-R}\right)^2+(1+K_d+K_t)R^2(1+M)^3-2R(1+M) -2\frac{M^2R^2}{1-R}(1+M)}{1+K_j-(1+K_s)M^2\left(\frac{R}{1-R}\right)^2}$$

Equation (8) represents the head capacity characteristics of all water jet pumps having straight cylindrical mixing chambers, and should approximate the characteristics of pumps having diverging or converging mixing chambers. All quantities involved are dimensionless. The relation between M and H depends on the five parameters R, K_s, K_j, K_t and K_d which vary with the design of the pump itself and with the lengths of the connecting pipe lines.

Efficiency

Two possible methods of computing the efficiency give different results. On the basis of the total energy into and out of the pump,

$$\eta = (Q_1+Q_2)\cdot h_d/(Q_1h_1+Q_2h_2) \qquad \text{.............(9)}$$

The value of η obtained from (9) depends on the datum for elevation used in computing h_d, h_1 and h_2 and can be made large by locating this datum far below the pump, since h_1, h_2 and h_d

become almost equal as z approaches infinity and hence η approaches unity.

The second method, which considers the total energy used and the work done, has a single value irrespective of the datum plane. The expression is

$$\eta = Q_2.(h_d-h_2)/Q_1.(h_1-h_d) = M.N. \qquad \text{............(10)}$$

EJECTOR CHARACTERISTICS

Several experiments were conducted by different investigators to develop the performance characteristics of the ejector. Mr. Gosline's and O'Brien's (Ref. 4) efforts were noteworthy.

The general shape of the characteristic curves was as shown in Fig. 5 and 6 which represented the data obtained from the pump showing the highest efficiency. The "cut off" point, or maximum quantity that can be pumped was illustrated.

Equation (8) represents the head-capacity curve for all liquid self-entrainment jet pumps. Knowing the loss coefficients and the geometrical proportions of the jet pump, the head-capacity and efficiency performance characteristics of a jet pump may be predicted from calculations. A typical curve was as shown in Fig. 7. The M-N and ηM curves (Fig. 7) for a specific jet pump may be determined by calculations or tests.

Effect of Nozzle Position

In Fig. 6, the effect of the distance from the tip of the nozzle to the inlet of the mixing chamber is shown for the area ratio giving the highest efficiency. The efficiency was increased slightly by pulling the nozzle back 1/4 inch, but a further increase of 1/8 inch caused a slight decrease in efficiency. For the condition shown, the maximum discharge was increased about 8 percent and the efficiency about 2.5 percent. Expressed as ratios,

Fig. 5 (Ref. 4)

Characteristics of a Cylindrical Pump,
Showing Effect of Cavitation

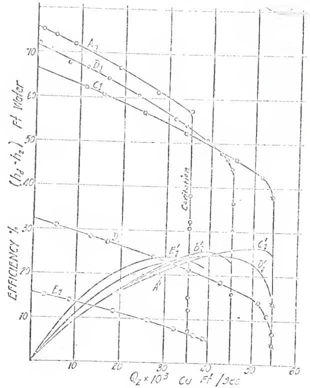


Fig. 6 (Ref. 4)

Effect of Nozzle Position on Performance.
Curves A, B, and C refer to head,
and A', B' and C' to efficiency.

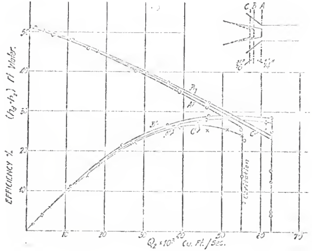
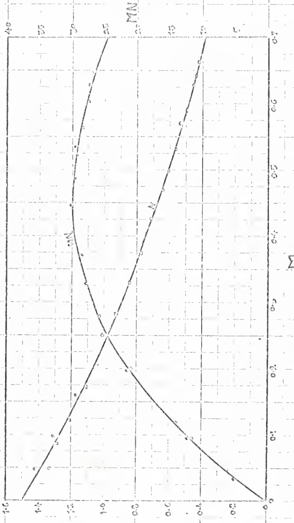


Fig. 7 (Ref. 7)

Water-Jet Pump Dimensionless Performance Characteristics Determined by Laboratory Test



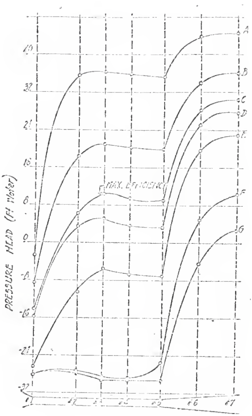
the best operating conditions were obtained when the tip of the nozzle was back from the entrance to the throat by a distance equal to the nozzle diameter.

Pressure Distribution Along the Mixing Chamber

Figure 10 shows the measured distribution of pressure along the mixing chamber for the combination of nozzle and mixing chamber showing the highest efficiency (R=25). Curve 'C' corresponds to the point of maximum efficiency of this combination. The pressure in the mixing chamber rises to a maximum value and then decreases, showing that in the pump which was used, it was longer than was necessary for complete mixing. More precisely, beyond the point of maximum pressure, the pressure rise caused by a combination of mixing process is more than counterbalanced by the friction loss. The curve clearly shows the great importance of the diffuser in the operation of the jet pump. The angle of diffuse was 11°, whereas the angle for least energy loss is 5.5°. The distance along the throat to the point of maximum pressure for the run showing the best efficiency was twelve times the diameter of the nozzle or six times the diameter of the throat.

Fig. 10 (Ref. 4)

Pressure distribution along mixing chamber and diffusor of a cylindrical pump.



DESIGN PARAMETERS OF AN EJECTOR

The design of efficient jet pumps and especially the ejector components of jet pumps has been of interest for a considerable time. Perhaps the reason for this interest is the simplicity of the ejector concept for imparting energy to a fluid stream. Although an ejector is a simple type of pump in that it has no moving parts, the prediction of optimum performance for an ejector is more complex than might at first be recognized. First of all there are a number of design parameters which may be varied. For example, a sketch of a simple ejector is shown in Fig. 11 (Ref. 9) and typical design variables are noted. They are

θ = Nozzle external angle

d = Nozzle diameter

s = Nozzle to mixing tube spacing

ℓ = Mixing tube length

t = Mixing tube diameter

α = Inlet angle of mixing tube

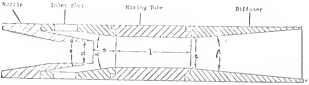
β = Diffuser angle

From the above parameters, the most important ones were

1. R = Ratio of nozzle area to mixing tube area.
2. $\frac{\ell}{t}$ = Ratio of mixing tube length to tube diameter.
3. $\frac{s}{d}$ = Ratio of spacing s between the nozzle exit and mixing tube entrance to the nozzle diameter.

Fig. 11 (Ref. 9)

Ejector Parameters



θ = Nozzle external angle

d = Nozzle diameter

s = Nozzle to mixing tube spacing

l = Mixing tube length

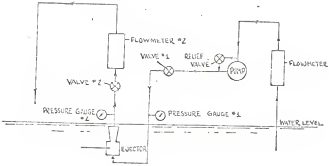
t = Mixing tube diameter

α = Inlet angle of mixing tube

β = Diffuser angle

Fig. 10a (Ref. 9)

Sketch of the Test Apparatus



OPTIMUM DESIGN PARAMETERS

In spite of complexities and possible ranges of design variables, several investigators have sought to optimize parameters and attain best performance. The performance is expressed in terms of efficiency defined by

$$\eta = Q_2(H_d-H_2)Q_1(H_1-H_d) = MN$$

where $M = Q_2/Q_1$ and $N = (H_d-H_2)/(H_1-H_d)$

A maximum value of η near 40 percent was anticipated in early work (Ref. 4), but was not achieved. F. Schulz (Ref. 15) predicted a maximum ejector efficiency of 38 percent for a ratio of nozzle area to mixing tube area of approximately 0.25 and a ratio of mixing tube length to the tube diameter in the range of 5 to 8. Another key parameter specified was the ratio of spacing s between exit and mixing tube entrance to the nozzle diameter. A value near unity was suggested. Ueda (Ref. 16) also predicted a maximum efficiency near 38 percent. For a flow ratio of secondary flow to primary flow near unity, an area ratio R of 0.25 was suggested. R. G. Cunningham's (Ref. 10) studies showed the significant dependence of maximum efficiency on the ratio R. Maximum efficiency was found for a value of R equal to 0.30. N. H. G. Mueller (Ref. 11) showed that the optimum area ratios fall near 0.3, optimum ratio of mixing tube length to diameter of about 7, and the spacing between nozzle and inlet to mixing tube is near unity. According to Mueller's (Ref. 11) recommendations, efficiencies in the vicinity of 37 percent should be realizable.

EXPERIMENTAL RESULTS (OPTIMUM PARAMETERS)

An extensive experimental program was carried out by the Deming Division of Crane Company (Ref. 9) by which a set of optimum design parameters for water ejectors was determined. Results were in a close agreement to the other research on the topic. Curves were prepared to facilitate the design process based on the experimental curves.

Optimum Parameters for Maximum Efficiency

Figure 11a shows a typical nozzle, mixing tube, and diffuser employed in the study. Three area ratios R = 0.295, 0.39 and 0.50 were employed in the study.

As has been noted in other studies, the efficiency appeared to vary inversely with R. The highest efficiency obtained along with the best combination of elements for various values of R was as given in Table 1. The various ranges tested gave values for which changes in the efficiency were relatively small.

Table 1

$R=\sqrt{\frac{d}{t}}$	$\frac{s}{d}$	Range of $\frac{s}{d}$	$\frac{\ell}{t}$	Range of $\frac{\ell}{t}$	α	Range of α	β	Range of β	ηmax
.295	0.8	0.8-1.2	4.1	3.5-4.5	40°	30°-50°	6.5°	5.5°-8°	42%
.39	1.3	1.0-2.0	4.1	3.5-4.5	40°	30°-50°	7.0°	6.6°-8°	35%
.50	1.4	1.2-1.6	4.1	3.5-4.5	40°	30°-50°	5.5°	5.5°-6.6°	33%

Within the experimental limitations, the greatest diffuser length proved most satisfactory. Maximum values were in the range from 29t for $\beta=5°$ to 16t for $\beta=9°$.

Cavitation--A Function of Secondary Flow Rate

The plots of M versus N for an R=0.295 ejector for various primary flow rates and various spacing ratios were as shown in Fig. 12. It was noted here that for the parameters chosen, the data points could be approximated with a straight line over a large portion of the total range. For high values of M the curves dropped off due to cavitation. The effect of cavitation on performance was obvious in Fig. 13 when M-N curves for an R=0.5 ejector were plotted. With the increase in the primary flow rate, it was seen that the cavitation point on the plots moved

Fig. 11a (Ref. 9)

Typical Components Using the Test Program

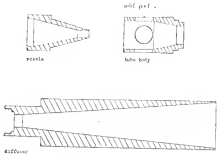


Fig. 13 (Ref. 9)

M-N Curves for R=0.5 Ejector Showing Effect of Cavitation

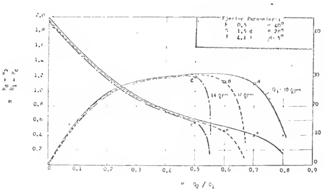
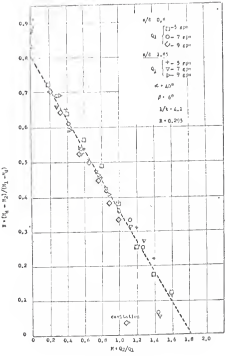


Fig. 12 (Ref. 9)

M-N Curves



toward lower values of M. Q_2 was computed for the various values of M chosen for the points circled in Fig. 13 and it was noted that

for point A, $Q_2 = M \times Q_1 = 0.7 \times 10 = 7$ gpm

for point B, $Q_2 = 0.6 \times 12 = 7.2$ gpm

for point C, $Q_2 = 0.5 \times 14 = 7.0$ gpm

Thus, it could be seen that the cavitation was primarily a function of the inlet secondary flow Q_2.

Application--Determining Optimum Configurations

Figure 14 was a single characteristic curve for each R and was obtained from the averages of a number of N-M curves for various R values. From a design standpoint one might expect that an ejector with an R ratio corresponding to one of the values in Fig. 14 and with the design parameter comparable to those listed in Table 1 would have efficiencies that should match closely those in Fig. 14.

Figure 15 was an envelope of M-N curves shown in Fig. 14 as well as curves depicting the variation of efficiency, shut-off N(M=0), and N values along the envelope curve as a fuction of R. The envelope curve was particularly useful in that for a given value of N, a point on the envelope curve gave the maximum value of M and hence the maximum efficiency for the ejectors under investigation. Given a point on the envelope curve the corresponding value of R can be found from N-R curve in Fig. 15 (Ref.9).

Fig. 14 (Ref. 9)

M-N and Efficiency Curves for Various R Values

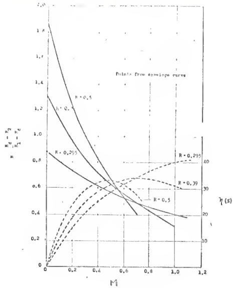
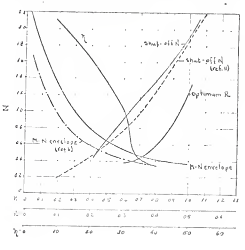


Fig. 15 (Ref. 9)

Envelope of M-N Curves and Associated Curves

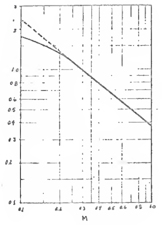


These curves were much useful in designing optimum ejector configurations. The analytical expression for the envelope curve was approximated by the equation $N \times 0.38\ M^{-.81}$ ($M \geq 0.25$) in the straight region for values of M greater than 0.25. The above expression was obtained by plotting the envelope M-N curve on logarithmic coordinates as shown in Fig. 16.

The result of the several studies on ejector performance showed that the area ratio R was perhaps the single most important parameter in determining ejector efficiencies.

Fig. 16 (Ref. 9)

Plot of envelope M-N curve on logarithmic coordinates



JET PUMP SYSTEM CHARACTERISTICS--

THEIR USE IN DESIGN

An experimental investigation was made by the Deming Division of Crane Company of the system characteristics. The reason for the above investigation was to present design charts for determining ejector characteristics to give optimum performance as a part of the jet pump system. The typical curves of ejector characteristics were as shown in Fig. 17 for three different nozzle to mixing tube area ratios R=0.295, 0.39 and 0.5. Then, an envelope of the performance curves was made which is shown in Fig. 18. The significance of the envelope curve was such that for a given value of N, a point on the envelope curve yields a value of M which corresponded to maximum efficiency for a class of ejector operations.

In order that the system operates to give maximum secondary flow rate Q_2, corresponding to a particular point on the envelope curve, it is necessary to set the inlet head to the pump and to select the operating point of the pump driving the primary flow. The optimum inlet setting for a pump driving an ejector was done as follows:

Referring to Fig. 19, the nondimensional head for an ejector was given by

$$N = \frac{H_d - H_2}{H_1 - H_d} \qquad \text{..............(1)}$$

Fig. 19 (Ref. 9)

Jet Pump System

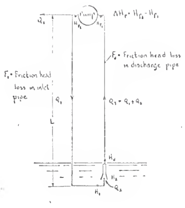


Fig. 17 (Ref. 9)

Ejector Characteristic Curves for Various R Values

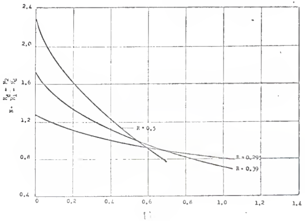


Fig. 18 (Ref. 9)

Envelope of Ejector Characteristics

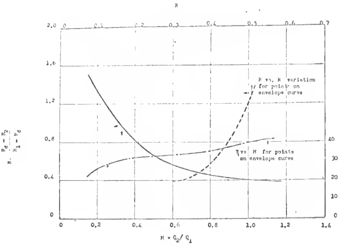
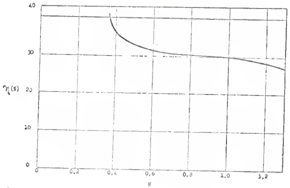


Fig. 18a (Ref. 9)

Efficiency versus N for the Points on the Envelope Curve



$$N = \frac{H_{p_1} + L + F_d - H_2}{H_{p_2} + L - F_i - (H_{p_1} + L + F_d)} \qquad \ldots\ldots\ldots\ldots\ldots(2)$$

$$= \frac{H_{p_1} + L + F_d - H_2}{\Delta H_p - (F_d + F_i)}$$

where

$F_d = F_d(Q_T)$ = frictional head loss in discharge pipe

$F_i = F_i(Q_1)$ = frictional head loss in inlet pipe

Therefore, it was concluded that for a given drive pump to operate at depth L (ΔH_p, F_d, F_i and H_2 known), the parameter N depends upon the inlet head H_{p_1} to the drive pump. A plot of η=MN versus N based on the envelope curve (Fig. 18) was as shown in Fig. 18a. It was noted that low values of N gave higher efficiencies. Therefore, one must attempt to set low values of H_{p_1} insofar as this is possible. For centrifugal pumps mounted at the top of a well and employed as the drive pump, a reasonable value for H_{p_1} is -25 ft which will probably keep the pump out of cavitation.

System Head-Capacity Characteristics

Referring to the jet pump system shown in Fig. 19, the inlet head to the ejector H_1 was found from

$$H_1 = H_{p_2} + L - F_i$$

$$H_1 = \Delta H_p - H_{p_1} + L - F_i \qquad \ldots\ldots\ldots\ldots\ldots(3)$$

The friction head loss was found from

$$F_i = \frac{f_i L}{D_1} \frac{V_1^2}{2g} = \frac{8 f_i L}{D_1^5 \, g\pi^2} Q_1^2 \qquad \ldots\ldots\ldots\ldots\ldots(4)$$

where f_i was the inlet pipe friction factor. The ejector discharge head H_d was found from

$$H_d = H_{p_1} + L + F_d \qquad \ldots\ldots\ldots\ldots\ldots(5)$$

where

$$F_d = \frac{8 f_d L}{D_2^5 \, g\pi^2} Q_T^2 \qquad \ldots\ldots\ldots\ldots\ldots(6)$$

From equation (4)

$$F_i = \frac{8 f_i L}{D_1^5 \, g\pi^2} \left(\frac{Q_T}{1+M}\right)^2 \qquad \ldots\ldots\ldots\ldots\ldots(7)$$

Equation (2) was rewritten as follows:

$$N = \frac{H_{p_1} + L + F_d\ (Q_T) - H_2}{\Delta H_p - [F_d(Q_T) + F_i(Q_T, M)]} \qquad \ldots\ldots\ldots\ldots\ldots(8)$$

After the drive pump (ΔH_p , Q_T) was selected and the optimum inlet head H_{p_1} set, it was noted that for M=0, the equation (8) was

$$N_o = \frac{H_{p_1} + L + F_d\ (Q_T) - H_2}{\Delta H_p - [F_d(Q_T) + F_i(Q_T, 0)]} \quad \text{.............. (9)}$$

For $M = \infty$, $Q_1 = \frac{Q_T}{1+M} = 0$ and $F_i = 0$, the value of N was

$$N_\infty = \frac{H_{p_1} + L + F_d\ (Q_T) - H_2}{\Delta H_p - F_d\ (Q_T)} \quad \text{.............. (10)}$$

Thus, $N_o \geq N \geq N_\infty$ for all values of M.

Application

The following system was considered

$L = 100$ ft

$\Delta H_p = 140$ ft

$Q_T = 16$ gpm

$D_1 = D_2 = 1^1/4$ inches (galvanized pipe)

$H_{p_1} = -25$ ft

$H_2 = 0$ (negligible submergence)

The head loss for various flow rates through the galvanized pipe were found from empirical loss curve as shown in Fig. 20.

$$N_o = \frac{-25 + 100 + 7}{140 - 2(7)} = \frac{82}{126} = 0.65$$

$$N_\infty = \frac{82}{140 - 7} = \frac{82}{133} = 0.615$$

Figure 21 showed the actual variation of N between the limits. This curve was called the system characteristic curve. It was

Fig. 20 (Ref. 9)

Experiment Head Losses Through
100 feet of galvanized pipe

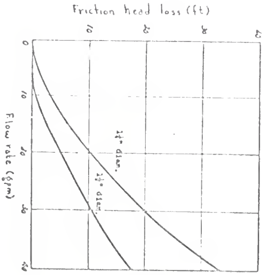
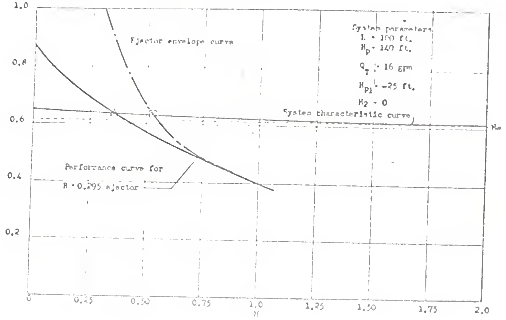


Fig. 21 (Ref. 9)

System Characteristic Curve
for a Particular System



concluded that the system characteristic curve is usually almost horizontal.

Then, an M-N ejector curve for an R = 0.295 was plotted along with a system characteristic curve as shown in Fig. 21. The point of intersection of the two curves gave the operating point (Δ) for the complete jet pump system. Q_2 and Q_1 were determined for Q_T = 16 gpm from

$$Q_2 = \frac{M}{1+M} Q_T = \frac{0.35}{1.35} \times 16 = 4.15 \text{ gpm}$$

$$Q_1 = \frac{1}{1+M} Q_T = \frac{1}{1.35} \times 16 = 11.85 \text{ gpm}$$

From the flat nature of the system characteristic curve, it was preferred to have an ejector characteristic intersect this curve as far to the right as possible, since it would result in larger values of Q_2 for the same operating point and the better efficiencies. It was found that the ejector envelope curve was the right-hand limit to the ejector characteristics. For example, the ejector envelope curve was shown in Fig. 21 which intersected the system characteristic curve at M = 0.525. Thus, if an ejector could be found which passed through this point (□) the flow rate Q_2 would be $\frac{.525}{1.525} \times 16 = 5.5$ gpm which was a 34 percent increase over the value found for R = 0.295 ejector.

After the operating point, Q_T and ΔH_p were specified, the design values for ejectors operating on the envelope curve were found as follows: First, Q_1 was found from $Q_1 = \frac{Q_T}{1+M}$. Then, H_1 was found from

$$H_1 = \Delta H_p + L - F_i - 25.$$

The nozzle velocity was approximated on the assumption that $H_2 \simeq 0$ to be

$$V_m = \sqrt{2g\ H_1}$$

$$\text{Also, } V_m = \frac{Q_1}{\pi d^2/4}$$

Converting d in inches and Q_1 in gpm and H_1 in feet d was found from

$$d = \frac{0.23\ Q_1^{1/2}}{H_1^{1/4}} \qquad \ldots\ldots\ldots\ldots\ldots(11)$$

From Fig. 18, for a point (M,N) on the envelope curve, a value of R was determined from envelope R-N curve (Fig. 18). And mixing tube diameter was found from

$$t = \frac{d}{\sqrt{R}}$$

The mixing tube length, spacing, and various other design variables were found from Table 1. For example,

$$H_1 = 140 - 25 + 100 - 7 = 208 \text{ ft}$$

$$Q_1 = \frac{16}{1+M} = \frac{16}{1+0.528} = 10.5 \text{ gpm}$$

$$d = \frac{0.23\ (10.5)^{1/2}}{(206)^{1/4}} = 0.20 \text{ in}$$

From Fig. 18 at N = 0.63, R = 0.42. Therfore, $t = \frac{d}{\sqrt{R}} = 0.30$ in.

The spacing between nozzle and mixing tube entrance was taken to be $s = 1.3 \times d = 0.26$ in.

Table 2

R	$(s/d)_{opt.}$ (Mueller)	$(s/d)_{opt.}$ $(1-R)/2R$ (Cunningham)
0.19	1.2	2.13
0.27	1.2	1.35
0.34	1.2	0.98
0.42	1.1	0.69

The mixing tube length was $\ell = 4.1\ t = 1.23$ in. The diffuser angle was chosen to be 7.0°.

EXPERIMENTAL VERIFICATION

The main purpose of the experiment was to verify the nature of the characteristic curves that have been reported. Figure 17 shows the sketch of the test apparatus. Data were obtained for various combinations of mixing tubes and nozzles. The satisfactory results were obtained with the ratios R = 0.333 and R = 0.277 which very closely compared with the reported R = .295. The diameters of the nozzles were 0.2514 inches and 0.3125 inches and those of mixing tubes were 0.435 inches and 0.5937 inches respectively.

The data were tabulated as given in Tables 3 and 4. The plots M versus N and M versus MN (for R = .277) were shown in Fig. 22. The plot of M versus N was almost a straight line, which was in agreement with the fact that over a large portion of M-N curves, before cavitation occurred, the data points approximated a straight line such as that obtained in Fig. 12. In Fig. 13 and 7 also, M-N plot was shown approximately straight.

The plots of Q_2 versus $H_d - H_2$ and η (for R = .277) were obtained and shown in Fig. 22a, which compared favorably with those of Fig. 5 and 6. The small difference in slope of the curves was due to the different scales of plots.

The plots of M versus N and MN were obtained for the nozzle area to the mixing tube area ratios R = .333, R = .277 and R = .179 for comparison. Over a large range of values, the efficiency appeared to vary directly with R, as shown in Fig. 22b. The plots compared favorably with those of Fig. 17, except that the data points here were approximated to the corresponding straight lines. These plots may be used to select the correct ejectors to give the required performance. For example, for shallow well performances, where quantity is the criterion, one may select the ejector with less steep characteristics in the required range. Similarly, for deep well performances where the discharge head is the criterion one may select the ejector with steep characteristics in the required range. Since the ejector discharge sets the inlet to the pump driving the primary flow, the driving pump has to be chosen accordingly.

Table 3

R = .277

Run	Q_1 % of 350 ppm	Q_T % of 350 ppm	Q_2 % of 350 ppm	H_{p_2} psi	H_{p_1} psi	H_1 psi	H_d psi	M	N	η (MN) %
1	44.3	47.8	3.5	44	16	45.51	17.51	.08	.625	.05
2	43.1	52.7	9.6	42	14	43.51	15.51	.22	.56	.1232
3	42	55.5	13.5	38.5	12	40.01	13.51	.322	.51	.164
4	41	59.2	18.2	36	10	37.51	11.51	.44	.44	.1935
5	40	62.8	22.8	34	8	35.51	9.51	.57	.37	.211
6	39	65.7	26.7	31	6	32.51	7.51	.685	.30	.2055
7	38	69	31	29	4	30.51	5.51	.816	.24	.1958
8	37	72.5	35.5	26	2	27.51	3.51	.96	.15	.144
9	36.5	73.5	37.0	25	0	26.51	1.51	1.01	.06	.06

Table 4

R = .333

Run	Q_1 % of 350 ppm	Q_T % of 350 ppm	Q_2 % of 350 ppm	H_{p_2} psi	H_{p_1} psi	H_1 psi	H_d psi	M	N	η (MN) %
1	32.2	32.5	0.4	56	27	57.51	28.51	.01242	.968	.0118
2	30.4	36.7	6.3	49	20	50.51	21.51	.207	.729	.152
3	29.9	38	8.1	47	18.5	48.51	20.01	.298	.702	.21
4	29.1	40.6	11.5	43	15	44.51	16.51	.396	.59	.23
5	28.9	43.2	14.3	40	12.5	41.51	14.01	.495	.51	.252
6	28.0	47.0	19.0	36	8.5	37.51	10.01	.68	.364	.248
7	27.5	51.5	24.0	30	4.0	31.51	5.51	.87	.212	.185
8	26.8	51.5	24.7	28	0	29.51	1.51	.922	.051	.047

Table 5

$R = .179$

Run	Q_1 % of 350 ppm	Q_T % of 350 ppm	Q_2 % of 350 ppm	H_{p_2} psi	H_{p_1} psi	H_1 psi	H_d psi	M	N	η (MN) %
1	27	27.5	0.5	40	11	41.51	12.51	.0185	.52	.0096
2	26.9	30.9	4.0	39	10	40.51	11.51	.1485	.396	.059
3	26	38.1	12.1	36	8	37.51	9.51	.466	.34	.16
4	26	41.5	15.5	35	7	36.51	8.51	.596	.304	.182
5	25.2	48.1	22.9	32	5	33.51	6.51	.91	.241	.219
6	25	50.5	25.5	31	4	32.51	5.51	1.02	.204	.208
7	24.6	55	30.4	29	2	30.51	3.51	1.24	.13	.1612
8	24	59	35	26	0	27.51	1.51	1.46	.058	.085

Fig. 22

MN Curve

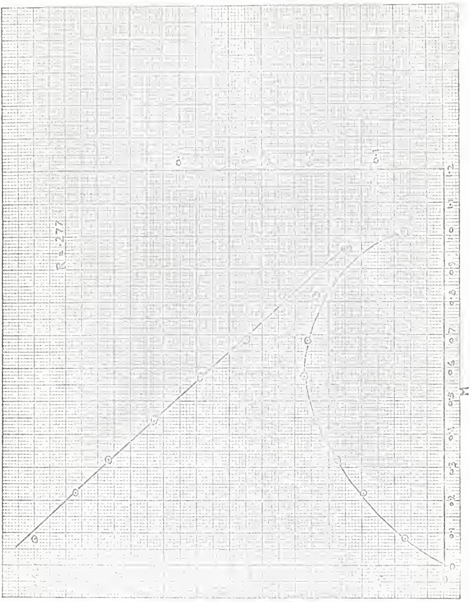


Fig. 22a

H_d - H_2 -- Q_2 Curve

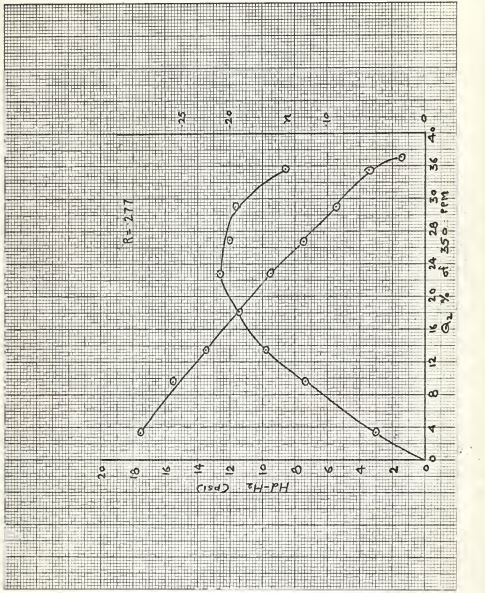
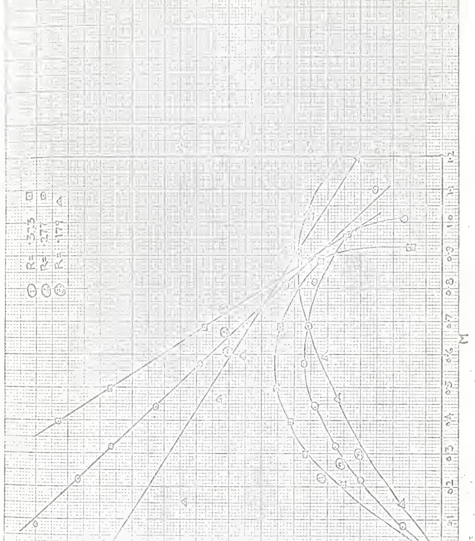


Fig. 22b

MN Curves



MATHEMATICAL ANALYSIS FOR OPTIMUM AREA RATIO

Efficiency of the jet pump has been defined as $\eta = M.N.$ For a particular M, η would be maximum when N is maximum. i.e. when $N = (1-T)/(T+M)$ is maximum (please refer to equation (8) in "General Theory") where T was given as follows:

$$T = \frac{1+K_j+(1+K_s)M^3\left(\frac{R}{1-R}\right)^2+(1+K_d+K_t)R^2(1+M)^3-2R(1+M) - 2\frac{M^2R^2}{1-R}(1+M)}{1+K_j-(1+K_s)M^2\left(\frac{R}{1-R}\right)^2} \quad \ldots\ldots\ldots\ldots\ldots(1)$$

T must be minimum for maximum N, where M is constant.

Differentiating equation (1) with respect to R and equating to zero, and assuming following values of other constants (practical values [Ref. 4]).

$M = 1.42$

$K_d = 0.70$

$K_j = 0.25$

$K_s = 0.02$

$K_t = .28$

gave an 8th order equation in R which was

$$-232.0\ R^8 + 1179.95\ R^7 - 2408.23\ R^6 + 2423.74\ R^5$$
$$-1293.68\ R^4 + 511.39\ R^3 - 167.19\ R^2 + 42.16\ R - 6.04 = 0$$

The above equation was solved to give R equal to 0.367, which was very close to the value of R equal to 0.333 obtained in experiments. The small difference might be due to the different values selected for the various constants. For the values of R less than R = 0.367 the efficiency was expected to vary directly with R and for the value of R greater than R = 0.367 the efficiency was expected to vary inversely with R. Since the values of R chosen for the experiments were less than R = .367, the efficiency appeared to vary directly with R for a constant M.

APPLICATION - AS AN OPERATING ENGINEER - ILLUSTRATION

An ejector in combination with a driving pump, associated piping and storage tank constitutes a complete hydraulic system. The exact application depends upon

1. pumping level, the vertical distance from the pump to the water level when the system is operating.
2. water required - the discharge capacity required for. a specific application.
3. the average discharge pressure required for a specific application.
4. location of the pump - corresponding piping layout for a specific application.

Since the performance of both the pump and the ejector are linked together, the system performance can be optimized by the correct selection of the following components:

1. Centrifugal pump
2. Ejector assembly
3. Size of the piping
4. Size of the storage tank

1. Selection of Centrifugal Pump

As already mentioned in "Jet Pump System Characteristics," if the operating point of the pump driving the primary flow is selected for a system and the optimum inlet head to the pump set, there is a point on the envelope curve at which the jet pump system operates and for which the secondary flow rate, Q_2, is probably maximum. Thus, selection of centrifugal pump determines ΔH_p, Q_T and the inlet head H_p, to the pump.

2. Ejector Assembly

After the system characteristic curve is determined, which is usually a horizontal line, it is preferred to have an ejector characteristic intersect the curve (Fig. 21) as far to the right as possible since this gives better quantity ratio M at a better efficiency. The right hand limit for all ejector characteristics is the envelope curve as shown in Fig. 21. From a number of ejectors, one is picked up which is operating near the intersection of envelope curve with the system characteristics.

3. Size of Piping

The continuity relations with due allowances for friction, etc., determines the size of the pipe.

4. Size of the Storage Tank

Basically, the jet pumps are used for low capacity requirements. Estimated water requirements for general service around the farm and home is approximately 119 gallons a day. The fact that the above requirement is intermittent and is not continuous permits us to use the jet pump along with a storage tank of correct size. The pump operates continuously at a low capacity and stores the water in the tank which may be used at the required hours. Knowing the peak hours of maximum quantity and pressure required, the size of the tank could be determined as follows: The tank should be sized in such a way that it can store the required amount of water for the peak hours. Also, its height should be determined such that it gives the required service pressure at the peak hours.

As an illustration, the selection of an ejector and the pump for the discharge rate of 6 gpm and the discharge head of approximately 100 feet was considered.

The performance curves furnished to Professor A. H. Duncan by Fairbanks Morse Pump Division of Colt Industries were used. Figure 22 is the AEC3 shallow well performance with ejector 4360. Figure 23 is the corresponding AEC3 centrifugal performance. First of all, the operating point (i.e. capacity 6 gpm and discharge head of approximately 100 feet) was located on Fig. 22. Five feet static lift was assumed and the corresponding centrifugal performance in Fig. 23 required that one must have the centrifugal pump having 24 gpm capacity and maintaining the pressure difference of approximately 57 feet. Different curves are shown for different static lifts.

For the head requirement of approximately 115 feet and the discharge of approximately 7 gpm, the nozzle with a diameter of 3/16 inch could be satisfactorily selected. For the above selection the catalogs furnished by Fairbanks Morse were referred to.

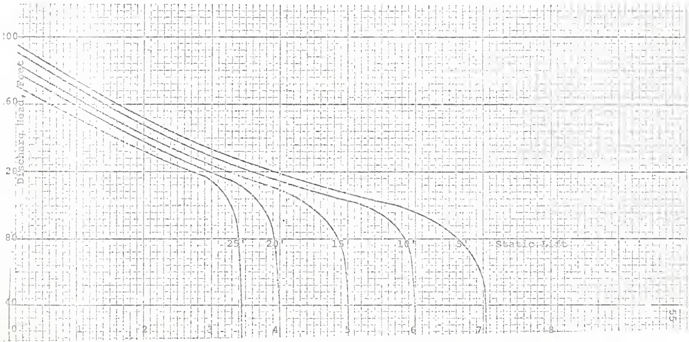
The different sections of the catalog also gave the friction losses for steel, copper, and plastic pipe in the pipe lines and fittings for the more precise selections of the pump.

Fig. 23 (Ref. 13)

AEC3 Centrifugal Performance
At 5' Static Lift

Fig. 24 (Ref. 13)

AEC3 Shallow Well Performance With Ejector 4360



Capacity GPM

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A STUDY OF WATER
JET PUMPS

by

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B. Engg., M.S. University of Baroda, India, 1963

AN ABSTRACT OF A MASTER'S REPORT

submitted in partial fulfillment of the

requirements for the degree

MASTER OF SCIENCE

Department of Mechanical Engineering

KANSAS STATE UNIVERSITY
Manhattan, Kansas

1968

The jet pump is a device for pumping fluids by means of a high velocity jet of the same or another kind of fluid. The principle phenomenon involved is the transfer of some momentum from the high velocity fluid to the low velocity fluid during the mixing process, with the result that all fluid leaving the mixing tube has about the same velocity. The general type of pump operating on this principle is classified as a momentum pump. Special names have been developed for jet pumps applied to specific services, the names being ejector, injector, hydraulic compressor, etc.,.

The jet pump is applied to many pumping problems due to its low first cost, simplicity in operation and ability to mix thoroughly two fluids. On the other hand, the usual installation is limited to small power inputs since the efficiency of the pump is inherently low. Development of theory has been slow because the fluid flow equations are complicated particularly in case of compressible fluids. High velocity jets of steam are used to pump air. Or sometimes, the jets of high velocity air are used to pump air.

As was pointed out earlier, the general class includes injectors and steam jet air pumps, as well as water jet pumps, but the latter possesses the decided advantage of not involving either compressibility or heat transfer.

The water jet pump was first used by James Thomson about 1852, and the theory of pumping by jet action was developed in 1870 by Rankine. The low efficiency of this pump has limited the field of application to conditions in which the absence of valves or working parts and the small size of the unit are sufficient to offset the greater power requirement of this than of other possible types. This unfavorable circumstance has caused the water jet pump to receive less attention than it deserves. A number of writers including Jesse (1904), Lorenz (1910), Gibson (1924), LeConte (1926), and Bergeron (1928) (Ref. 7) have developed theoretical equations, but almost no experimental data for checking these equations were available.

This report has been designed to summarize particularly the operation, theory, design characteristics and the application of "water jet pumps" in which the water is used as a primary as well as secondary fluid. Optimum design parameters, the jet pump system characteristics and their use in design have been reported and discussed. Various advantages of jet pumps have been mentioned. Experimental verification was made and the results reported.