A SUGGESTED SOLUTION FOR THE EJECTOR PROBLEM

by

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INTRODUCTION

The progress of an established industry, as well as the founding of a new one, calls for the accumulation of large funds of design data, experience and "know how". This can be realized through close co-operation between the designer, the manufacturer and the user.

This report is put forward with a view to enlist the help of manufacturers and previous investigators in compiling data about their flow apparatus, for the ejection and injection of fluids, during service. This will give the designer a better insight into the functions of the various parts of the apparatus and a more accurate means of estimating its performance.

The chief difficulty in the proper design of an ejector is the lack of a theory by means of which the proportions of the different parts, which correspond to the optimum operation, can be calculated for any given set of working conditions. Rigid mathematical analyses fail to represent the actual performance and give the best design proportions. This is due to the high speed mixing and diffusion processes, as well as the expansion and the recompression processes, which usually take place in the ejector's operation.

It thus follows that the development and construction of efficient ejectors, and their adaption to the many applications for which they can be used, require specialized experience. The design details and relative proportions of the internal parts vary according to the individual operating conditions. Nozzles, diffusers and other sections should be arranged to meet the most exacting requirements of the operating pressures, vacuum and the compression ratio.

The first part of the report will serve to illustrate (1) the principle of the ejector, advantages in its use and its fields of application, as well as (2) the influence of the geometrical proportions of its different sections on the performance, and the role each section plays.

In the second part of the report, a method for the representation of the performance factors of the ejector in the form of a design chart is suggested. This method extends the previous work carried out by the author on the ejector's problem (15). It produces, when successful, all the necessary information needed to construct an apparatus which performs efficiently for any given set of operating conditions.

SYMBOLS USED

a — cross-sectional area of the suction pipe.

A — cross-sectional area of the mixing chamber at the surface of enlargement.

d_d — diffuser exit diameter.

d_k — diameter of the parallel section.

d_t — forcing nozzle throat diameter.

H_a — change of enthalpy for the induced fluid (expanding from P_o to $P_{o'}$).

$H_{e'}$ — isentropic change of enthalpy of the forcing jet (expanding from P_e to $P_{o'}$).

H_i — isentropic change of enthalpy of the induced fluid (compressed from P_o to P_d).

H_k — actual change of enthalpy of the forcing jet from P_e to P_d.

$h_{to'}$ & $h_{ta'}$ — stagnation enthalpies of the forcing and induced streams, respectively, at the combining cone inlet section.

K — total enthalpy ratio $(M_a h_{ta'}/M_o h_{to'})$.

L — parallel section length.

M_a — mass rate of flow of the induced medium.

M_o — mass rate of flow of the forcing medium.

P_d — delivery pressure of the mixture.

P_e — initial pressure of the forcing fluid.

P_n — mixture pressure at the parallel section end section.

P_o — pressure inside the mixing chamber.

$P_{o'}$ — average pressure of the mixture at the combining cone inlet section.

u_a — induced medium velocity at impingement.

$u_{a'}$ — induced medium velocity at the combining cone inlet section (section o'-o', Fig. 1).

u_d — discharge velocity of the mixture.

u_n	velocity of the mixture at the parallel section end section (section n-n, Fig. 1).
u_o	velocity of the forcing stream at the forcing nozzle exit section.
$u_{o'}$	velocity of the forcing jet at the combining cone inlet section.
$V_{a'}$&$V_{o'}$	specific volumes of the induced and the forcing streams, respectively, at the combining cone inlet section.
X	axial distance from the nozzle exit section to the combining cone end section (section o"-o", Fig. 1).
X'	axial distance from the nozzle exit section to the combining cone inlet section (section o'-o', Fig. 1).
α	velocity ratio ($u_{a'}/u_{o'}$).
μ	mass ratio (M_o/M_a).
ρ_o&ρ_a	densities of the two streams before they meet.
$\rho_{o'}$&$\rho_{a'}$	densities of the two streams at the combining cone inlet section.
σ	velocity ratio ($u_n/u_{o'}$).
θ	diffuser total divergent angle.

REVIEW OF THE LITERATURE

In 1928 Mallenby (11) investigated the ejector performance experimentally. A design chart was constructed from his experimental results. For given initial and vacuum pressures, the ratios X/d_k, d_k/d_t and M_a/A_t (see Fig. 1) can be obtained from the chart. The characteristic design curves of Mallenby (11) were plotted from his own experimental results only; he made no attempt to check his results with those obtained from other sources using similar design data. In his conclusions, Mallenby (11) stated that the forcing nozzle position (or the distance (X), Fig. 1) is of minor importance. Besides, unstable flow appeared in the first part of the diffuser. This is apparantly expected since no parallel section was used.

In 1933 Watson (17) designed a steam-air ejector with the object of investigating the effect of certain quantities on the performance; the chief of which was the distance (X). He believed in the dominating effect of the forcing nozzle position on the overall operation. Until that time, investigators had different ideas about the importance of the distance (X). Mr. Robinson, in his discussion of Watson's (17) work, confirmed his remark, while Mr. Gresham disagreed with him and agreed with Mallenby (11) that this dimension is unimportant. Watson (17) was the first to consider the use of a parallel section between the combining cone end section and the diffuser inlet section; a 1½-inch section adimitted greater stability than a 1/16-inch parallel section. Although his experimental results show that appreciable pressure rise takes place in the combining cone, his design calculations were based on the assumption that the pressure of the mixture is unchanged along that section. Besides, he assumed in the calculations that the entrained air velocity, u_a, was 100 ft/sec, and the velocity of the mixture leaving the diffuser, u_d, was 220 ft/sec. From the present author's experience, it is believed that u_a=100 ft/sec is too low, while u_d=220 ft/sec is too high.

In 1939 Flugel (4) presented a theoretical analysis of the performance of ejectors in terms of non-dimensional parameters. A detailed account of his parameters, their significance and the theoretical approach used to tie them in the form of design formulas, will be given later in this report. Flugel (4) suggested that a parallel section with a length-to-diameter ratio, L/d_k=10, should be provided. The numerical values of the different coefficients given by Flugel (4) are not reliable for general use, since these coefficients are expected to have different values when the set of operating conditions change.

In 1941 Royds and Johnson (13) carried out a set of experiments on a steam-air ejector with the purpose of expressing the experimental results in a form which reveals the fundamental principles for application to the design of ejectors. A brief account of their findings is that: (a) the fluid in the diffuser obeys the same laws of discharge as in a convergent nozzle although with a different coefficient of discharge, (b) when $(\mu)=M_o/M_a$ is greater than 10 the combining cone length is not important, and for $(\mu) < 10$ the exit of the forcing nozzle should lie within the entrance of the cone. This finding contradicts commercial practice, where (μ) is much smaller than 10 and the nozzle can be placed outside the combining cone. In fact the optimum position of the nozzle, which gives an efficient operation, depends not only on the parameter (μ)

but also on the other working conditions applied. The authors also stated that the diffuser length is not important. The present author disagrees with them and believes that the diffuser proportions have an appreciable effect on the performance, as will be explained later. Finally, Royds and Johnson (13) pointed out that their results showed superior air entraining properties arising from an under-expanded jet. It is difficult to reconcile their remark with those of other investigators who prefer over-expansion. Ejectors appear to work satisfactorily under either condition.

Ln 1942 Goff and Hooger (5) investigated the velocity profile in the mixing zone produced by a homogeneous air stream issuing into still air. They concluded that the velocity profile, when the two mediums have different densities, is similar to that obtained in Tollmien's analysis, when both mediums possess the same pressure and temperature. The authors, accordingly, suggested that Tollmien's analysis may be extended to any density ratio. To support their statement, Goff and Hooger (5) constructed an ejector creating a density ratio $(\rho_o/\rho_a)=1.015$. They should have extended their experiments to cases when the density ratio between the twe streams is higher or lower than the ratio 1.015 investigated. The authors' problem is different from the one discussed in this report. However, the effect of the density ratio between the two streams on the velocity profile is of considerable importance in the present work.

Ln 1950 Kastner and Spooner (8) carried out a set of experiments on an air-air ejector and concluded that the best setting for the nozzle should be at $X=1\frac{1}{4}$ inches (Fig. 1) and the parallel section length-to-diameter, L/d_K, should be 7. These recommended values may be of interest to designers using similar operating conditions to those of the authors. Their results, based on the use of driving air at low initial pressures (from 7 psig to 20 psig), have limited fields of application. Kastner and Spooner (8) avoided carrying out any theoretical analyses. Their reason is that the problem is difficult to generalize from individual analyses.

In 1950 Johannesen (7) used an air-air ejector to investigate experimentally the influence of the velocity ratio (α) on the performance. However, no definite conclusions have been reached. This was reasoned by the author as follows: To improve the mixing process, the velocity difference (u_o-u_a) should decrease. This was afforded by either increasing u_a or decreasing u_o. In both cases all the other variables were changed accordingly, and hence the obtained values of (α) were not definite. Johannesen (7) emphasized the importance of the role which the parallel section plays; a length-to-diameter ratio $L/d_k=10$ should be provided. As previously mentioned, the optimum ratio could be much smaller than this recommended value, since the operating conditions are influential. In general, Johannesen (7) was unable to present with confidence a general law for the determination of the different proportions of the ejector.

In 1952 Smith (16) derived theoretical formulas investigating the pressure rise in an ejector. One of his formulas corresponds to a gas-gas operation. It was derived by applying the continuity and the momentum principles. The minimum amount of gas required for any duty, M_o, when $\rho_o/\rho_a=1$ can be calculated therefrom. Checking his formulas experimentally, Smith (16) found that they can be used with certain reservations. A ratio X/d_k from 0.5 to 1.0 was found to give an optimum operation. Concerning the parallel section, a ratio L/d_k from 6 to 7 was recommended; 6 was preferred. The present author believes that more sound design formulas could have been obtained if Smith (16) had included the velocity ratio (α) in his investigations.

In 1956 the present author (15) carried out an experimental investigation of the performance and design factors of an air-air ejector. Since the work presented in this report extends the previous investigation, a detailed review will be illustrated later. In brief, a design chart which included most of the variables involved has been constructed and recommended for use to obtain ready data for the design and to predict expected performance. The effect of the density ratio on the performance was ignored. Besides, in order to simplify the calculations, it was assumed that the driving jet issuing from the nozzle retains its shape until it reaches the combining cone entrance. This assumption was found later by the author to be doubtful, since when the nozzle moves far away from the cone, the jet is liable to spread out due to the divergence of the exit portion of the nozzle. Any pressure difference between the jet at the nozzle exit and its surroundings influences the shape of the jet.

In 1958 Fabri and Paulon (3) presented an interpretation of what they called "cylindrical mixers". For the sake of simplicity, the authors: (a) placed the primary flow exit from the nozzle at the inlet section of the parallel section, and (b) assumed an isentropic process of the forcing jet along the parallel section. By applying the continuity and the momentum principles for both streams along the parallel section, a formula was developed to give the optimum mass ratio (μ). Their formula was also based on the assumption that: (c) the Mach Number of the secondary stream at the parallel section end section is unity. The present author believes that assumptions (b) and (c) of Fabri and Paulon (3) are hardly warranted. Realizing the viscosity effect, the authors introduced the so-called "load loss" due to friction resistance along the parallel section. A formula, including a gas mean turbulent coefficient of friction, was suggested for this purpose. Similar formulas, using similar analysis, were obtained for the divergent section. Only one length of the parallel section was used. Since the forcing nozzle exit was placed at the parallel section inlet section, the present author believes that efficient results can only be obtained if a relatively long section is used. This is because the constant area mixing process is a slow process (4).

Further survey of the literature is not presented, since no further investigations, which can serve the purpose of this report, have been published.

PART I

USEFUL INFORMATION

The Fundamental Principle

In its simplest form the ejector consists of a nozzle, a mixing chamber and a convergent-divergent section (or just divergent, according to whether the mixture velocity at its inlet section is

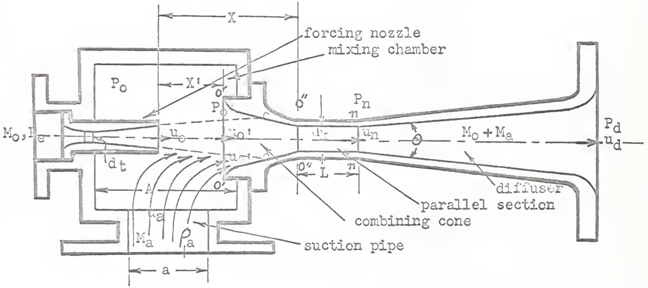

Fig. 1 A Section through an Ejector

supersonic or subsonic, respectively), Fig. 1. A driving fluid,usually steam, is supplied to the driving nozzle, where it expands and leaves the nozzle exit in the form of a high velocity jet. This rapidly moving jet draws molecules of any gas or vapor with which it comes in contact and imparts some of its velocity to them. The two streams intermix, partly in the mixing chamber where such intermixing almost takes place at constant pressure. The actual compression process occurs along the combining cone and the diffuser. A parallel section introduced between them is a common practice; its function will be illustrated later in the report. These sections combined convert the kinetic energy of the mixture into flow work. The mixture finally discharges at the back pressure.

Advantages

Ejectors are outstandingly simple and compact in design, with no moving parts, no packing glands and practically noiseless in operation. Continuous operation with almost no attendance is thus assured. These features, combined with their light weight, low initial and installation costs (as no foundation is required) as well as their long life expectancy, account for the rapidity with which wide new avenues of their application have been opened in practically all industry.

Fields of Application

In many cases ejectors can be utilized in applications where mechanical pumps cannot be used.

Steam-operated ejectors are particularly suited for handling gases and are, therefore, useful where air or vapor removal against high vacuum is required, such as in modern turbine plants. They are also suitable for a wide range of liquid transfer operations and for heating, circulating, agitating and mixing of fluids in various processing operations.

Air or gases can be used effectively in the place of steam for air or vapor removal (but are not suitable for pumping liquids). More recently, ejectors have been used to entrain cooling air, by jets of exhaust gases from aircraft powered by reciprocating engines, to augment the flow of the coolant. Air-operated ejectors can also be used to transfer powdered materials from one location to another.

Liquid-operated ejectors are effectively used on liquid-transfer operations and can also pump fluids with solids in suspension.

In general, when provided with an operating medium under pressure and working under certain conditions, ejectors will perform a wide variety of pumping operations. Their applications are practically unlimited and their performance, in some applications, is unique. While already widely used throughout industry, many new uses are constantly being developed.

Functions and Geometrical Proportions of the Different Sections

The roles which the different sections of the ejector play can efficiently be achieved when their general proportions are selected properly. Such roles and recommended proportions are presented in this report, according to published data, in order to give a better insight into the ejector's operation and the design problems connected with it:

The Forcing Nozzle. By virtue of converting the pressure head of the operating medium into a high velocity stream, the ejector is able to induce fluids into the suction chamber and discharge the mixture against a higher pressure. The conversion is carried out by the forcing nozzle, which supplies the energy input. When its total divergent angle is greater than 12 degrees, the forcing jet may lose contact with the diverging wall. A compromise was found in the common use of a total included angle varying from 6 to 12 degrees. A straight conical flare of the inside walls is a common practice. The outside shape of the nozzle is of importance only when the nozzle lies inside the combining cone. In this case, a conical outside surface with an angle less than the cone angle is preferable; it gives a nozzle effect in accelerating the induced fluid before its entrainment with the fluid from the forcing nozzle.

The axial position of the forcing nozzle with respect to the combining cone (the distance (X) or (X'), Fig. 1), influences the overall performance: when (X) or (X') increases, the free surface of the forcing jet (outside the nozzle) increases. Since this surface is responsible for the dragging effect, the amount of the fluid induced increases. This effect extends until an optimum position (a distance (X_{opti}), Fig. 1) is reached, beyond which detached shockwaves arise inside the combining cone and the flow will be carried back into the suction chamber, breaking down the ejector process.

The Combining Cone. When the jet of the two streams inside the mixing chamber possesses a sufficiently high velocity (with a Mach Number greater than unity), provision of a combining cone is necessary. Its function is generally the reverse of the accelerative expansion process which occurs in the forcing nozzle. It reverses the direction of the process, discharging the mixture at a lower velocity and higher pressure. The nozzle position and the combining cone angle are responsible, simultaneously, for obtaining a weak or a strong shock wave at section c-c, Fig. 1, where the jet strikes the combining cone walls. When these proportions are selected properly, the condition of flow at section c-c will be similar to a flow arround a wedge with an attached shock wave (7).

The combining cone angle depends on the condition of the incoming flow. While Smith (16) recommended 25 degrees, the Honolulu

Electric Product Company uses a priming ejector with a 6-degree cone angle. Koerting[1] recommended 9 degrees and Johannesen (7)suggested a range from 15 to 24 degrees. Kastner (8) recommended a range from 23 to 36 degrees, while Eldred, in his discussion to Kastner's (8) work, suggested 30 degrees for steam-operated ejectors. A 15-degree angle has been used by Jackson (6), Whittington[2], Weir[3] and Dalas[4]. This shows the difficulty of selecting a suitable cone angle. The selection depends mainly on the given specifications.

Concerning the combining cone length, it also varies over a wide range. Manufacturers have used cones with lengths ranging from 1.5 to 5 times the throat diameter, d_k.

The best shape of the combining cone should be a smooth sweep with a well rounded entry. This was stated by Mr. Fitt in his discussion to the paper written by Watson (17). However, a straight tapered cone with a short rounded entry is also efficient. The rounding helps in producing a smooth flow of the entrained fluid.

The Parallel Section. A parallel section introduced between the combining cone exit section and the diffuser inlet section dampens the waves caused by the interaction between the two streams (11) and thus secures stability of the flow operation[5]. Besides, a more

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1 "Strahalapparate" catalog No. 21 IV, 1950.

2 A primary ejector, size P-15, Serial No. 16A 1209-7, 1-10-1958, designed by Whittington Pump and Engineering Corporation.

3 A single-stage ejector installed in Fulham Power Plants, England.

4 A two-stage ejector installed at North Cairo Power Station, Egypt.

5 A stable operation is the operation of the ejector without violent fluctuations of the suction pressure (see "Standards for steam jet ejectors", Third Edition, Heat Exchange Institute, 1956, p. 11).

intense exchange of momentum between the two streams takes place inside the parallel section with the result that a better mixing process, accompanied by a higher pressure rise, is established. This has been evidenced by Flugel (4), Keenan (10) and Johannesen (7).

The optimum length-to-diameter ratio, L/d_k, of the parallel section varies with the operating conditions. A comparatively high ratio increases the wall friction resistance, which opposes the gained pressure rise. In a similar manner, when the ratio L/d_k is less than a certain optimum value, shock waves may appear in the early part of the diffuser, due to the incomplete mixing operation inside the parallel section.

The proper selection of the cross-sectional area of the parallel section is also very important. Small variations in d_k affect the amount of the induced fluid, M_a, considerably. Choking may occur when d_k is small, while the fluids may flow back if the diameter is larger than its correct value (11).

<u>The Diffuser</u>. When a correct length of the parallel section is used, the mixture which enters the diffuser is expected to be homogeneous and stable. In this case, the diffuser's function is to bring the homogeneous mixture to the delivery pressure by converting the kinetic energy of the mixture into flow work.

In a manner similar to that discussed for the parallel section length, a diffuser that is too long augments the wall friction losses, while a short one places a limit to the deccelerative compression process. When d_k and d_d, Fig. 1, are properly estimated, the diffuser length can be obtained, once the divergent angle, Θ, is selected. An angle of 6 to 8 degrees is a standard practice. von Eck (2) suggested that the choice of the diffuser angle depends on the value of the Reynold's Number of the fluid at the diffuser inlet section. He presented the following table:

R_e	50,000	100,000	150,000	200,000	$2x10^6$
θ	10	8.42	7.6	6.7	5.5

The above recommended values of the total divergent angle correspond to a homogeneous stream flowing inside the diffuser. Thus if the parallel section used provides a homogeneous mixture at its end section, the values of θ given by von Eck (2) can be used by ejector designers.

Although a 6-degree diffuser is a favorite among ejector manufacturers, Smith (16) suggested 10 to 15 degrees.

PART II

A METHOD FOR THE REPRESENTATION OF THE PERFORMANCE FACTORS OF EJECTORS

To obtain optimum values suitable for use in design work and in anticipating the performance of similar ejectors, it is suggested that the performance of such apparatus be represented in terms of non-dimensional parameters. To classify these parameters according to some mathematical reasoning, the analysis given by Flugel (4) was used as a basis. It incorporated the following parameters:

The degree of approach between the velocities of the induced and the forcing streams, $(\alpha) = u_a/u_o$, which govern the momentum exchange between them.

The degree of economy in the consumption of the forcing stream $(\mu) = M_o/M_a$.

The pressure ratio or the pumping head (P_o/P_d).

The extent of momentum exchange between the two streams at the end section of the parallel section $(\sigma) = u_n/u_o$.

The efficiency of the pumping action $(\eta) = M_a(P_d-P_o)/M_o(P_e-P_o)$ for liquids, or M_aH_i/M_oH_k for compressible fluids (see Fig. 2).

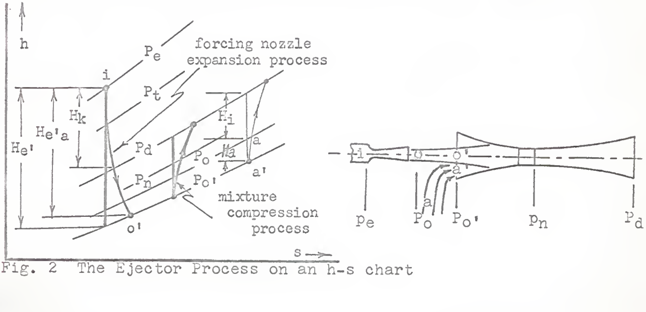

Fig. 2 The Ejector Process on an h-s chart

From the momentum and energy equations, Flugel (4) obtained mathematical equations relating the above parameters. These relations brought out the importance of the parameter (α) as a governing factor in the ejector performance. Applying these parameters to an experimental apparatus, the present author (15) established the practical form of the curves of Flugel (4) for equal density, at impingement, of the induced and the forcing mediums, viz. $(\rho_o/\rho_a) = 1$.

Since the proportions of the parallel section, as previously

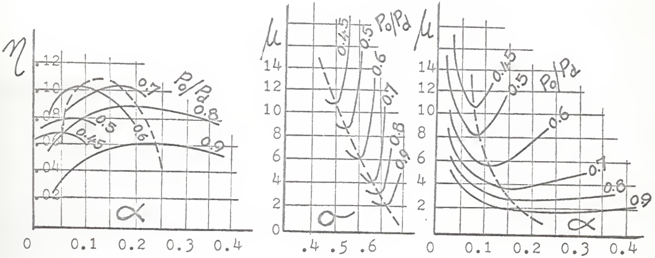

Fig. 3 A specimen of the curves obtained experimentally for $L/d_k = 5$ and $a/A = 0.5$. The dashed-line curves represent the optimum values of the different parameters, when 200 lb_m/hr air at P_e = 35 psig was supplied to the apparatus (15).

mentioned, influence the performance, provisions were made to vary the length-to-diameter ratio of this section, L/d_k, in order to obtain the suitable ratios which correspond to the different operating conditions.

The minimum pressure loss in the sudden enlargement of the suction pipe at its connection with the mixing chamber was found, by the momentum and energy equations and checked by experiments (15), to occur when the area ratio $a/A = 0.5$.

The optimum values of the different parameters were, therefore, chosen for the case of this area ratio (see Fig. 3). These optimum values were then plotted in the form of a chart having (α) as a common basis. Figure 4, presented on page 21, is plotted to scale from the author's previous experimental results (15). It is recommended for use to obtain the optimum design parameters and the expected overall efficiency when the operating conditions applied are similar to those investigated by the author.

For a more universal use of the chart in Fig. 4, it should be extended to cases where the densities of the two mediums at impingement are not the same. Figure 5, page 22, would then be typical for cases in which the forcing and the induced mediums are in the same phase viz., both being gases or liquids.

If the two mediums impinge on each other while at different temperatures, the subsequent equalizing of their temperatures during the mixing process affects their respective densities. The temperature difference may, therefore, be regarded as a factor influencing the density ratio of the two mediums and may be applied as a correction factor thereto. Since the final mixture temperature is not only a function of the respective temperatures of the two mediums, but also of their respective masses and specific heats, an additional non-dimensional parameter is proposed in the form: $(K) = M_a h t_a' / M_o h t_o'$. This parameter is applied to the density ratio as a correction factor. The optimum values of (K) at different values of (α) were obtained experimentally by the

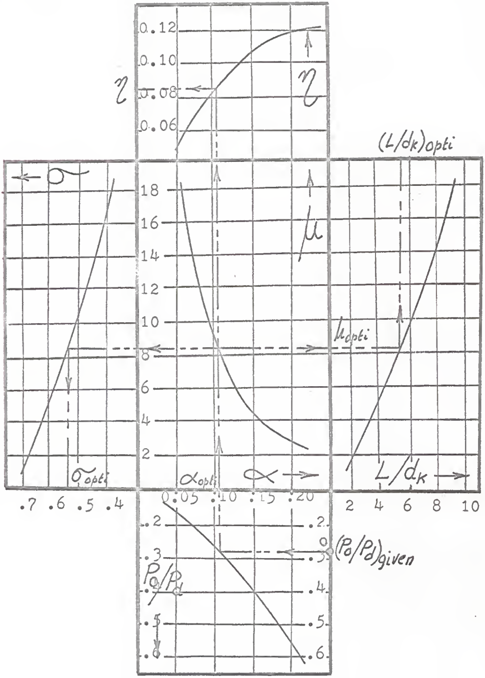

Fig. 4 An experimental chart recommended for design ($\rho_o/\rho_a = 1$). For a given value of P_o/P_e, enter the diagram at point O to obtain the optimum suitable parameters as well as the expected efficiency.

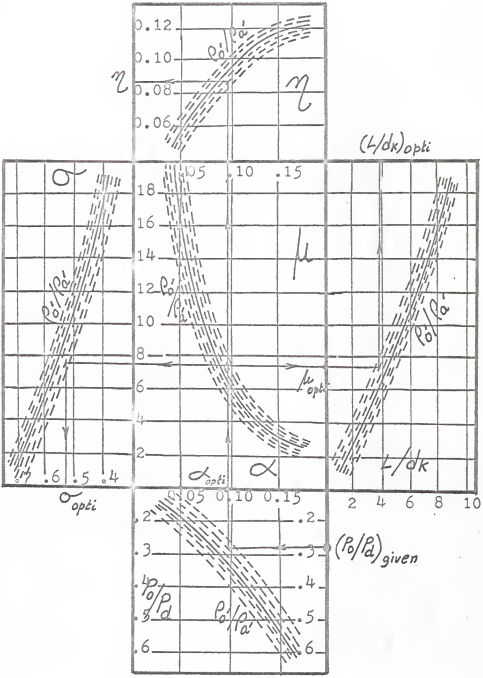

Fig. 5 Extension of the design chart of Fig. 4 to the cases where the density ratio $\rho_{o'}/\rho_{a'} \neq 1$.

author (15) for $(\rho_o/\rho_a) = 1$ and air as a forcing medium. For other density ratios, further experimental work or measurements from actual practice is required. The dashed-line curves shown in Fig.6-a are the expected shapes which may then be obtained. In Fig. 6-b, the expected variation of (K) with (μ) at different density ratios is

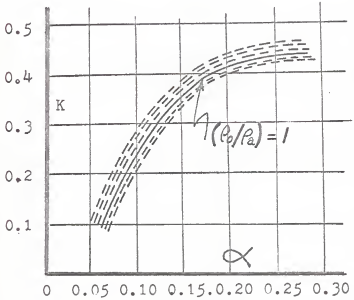

Fig. 6-a Variation of $(K)_{opti}$ with (α) for $\rho_o/\rho_a = 1$. The dashed-line curves show the expected variations at other density ratios.

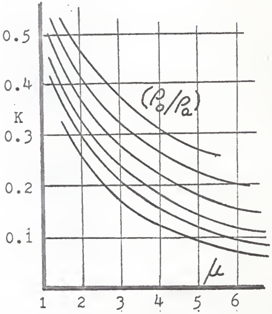

Fig. 6-b Variation of $(K)_{opti}$ with (μ) at different density rat-tios, according to theoretical analyses.

shown. The curves are plotted from mathematical equations derived by the author, which relate (ρ_o/ρ_a) to (K) and (μ).

The average pressure of the two streams at the combining cone inlet section, P_o', may be equal to, higher or lower than the vaccum pressure, P_o, inside the mixing chamber. Its value depends on the extent of the partial mixing process between the two streams along the distance (X'), Fig. 1. It is suggested that the change of pressure from P_o to $P_{o'}$, inside the mixing chamber, and from $P_{o'}$ to P_d, along the combining cone, parallel section and the diffuser, be expressed in terms of an additional non-dimensional parameter, $H_1/H_{e'}$, illustrated on the h-s chart of Fig. 2. This suggested enthalpy change ratio corresponds to isentropic processes, Once the

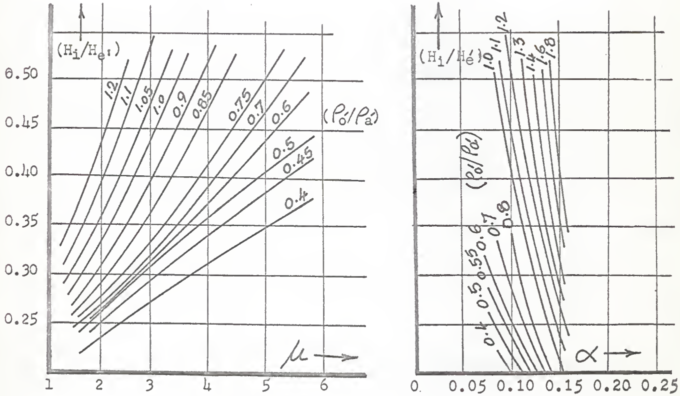

Fig. 7 Expected variation of $(H_1/H_{e'})$ with (μ) and (α) at different density ratios.

initial pressure of the driving fluid, P_e, its initial temperature, T_e, the vacuum pressure inside the mixing chamber, P_o, and the delivery pressure, P_d, at the exit section of the diffuser are given, a properly selected value of the ratio $(H_1/H_{e'})$ will lead, directly, to the estimation of the average mixture pressure of the two streams, $P_{o'}$, at the combining cone inlet section. Knowing $P_{o'}$, the designer can estimate the velocities and densities of the two streams, and, accordingly, the areas required for their flow, at section o'-o'. The cross-sectional area of the combining cone inlet section is, roughly, the sum of the two estimated areas. In brief, the selection of a proper value of the parameter ($H_1/H_{e'}$) for the given operating conditions will give the designer a better insight a better insight into the actual flow operation at the combining cone inlet section. In Fig. 7, the expected variation of the parameter ($H_1/H_{e'}$), with (μ) and with (α) at different density ratios is shown.

The Suggested Design Chart

The final form of the chart suggested for the design of ejectors may then be as shown in Fig. 8. It includes the parameters (P_o/P_d), ($\rho_{o'}/\rho_{a'}$), (μ), (α), (K), (σ), ($H_1/H_{e'}$) and (L/d_k).

Since the performance curves are based on non-dimensional parameters, the chart should be of universal application. Widening the scope covered by this chart should render it useful for the design of any ejector. This can be afforded through the recording of performance data of different types of ejectors in practice.

The Recommended Design Procedure

The procedure suggested to obtain the optimum recommended values of the different parameters from the design chart can be demonstrated as follows:

1- For the given (P_o/P_d) ratio, calculate (ρ_o/ρ_a) and assume $(\rho_{o'}/\rho_{a'})_1$, the density ratio between the two streams at section o'-o',Fig. 1. Enter the design chart at points (a) and (b). Follow the arrows a-c and c-d. Point (c) lies on the $(\rho_{o'}/\rho_{a'})_1$-curve and point (d) gives the recommended mass ratio, $(\mu)_1$. A horizontal line

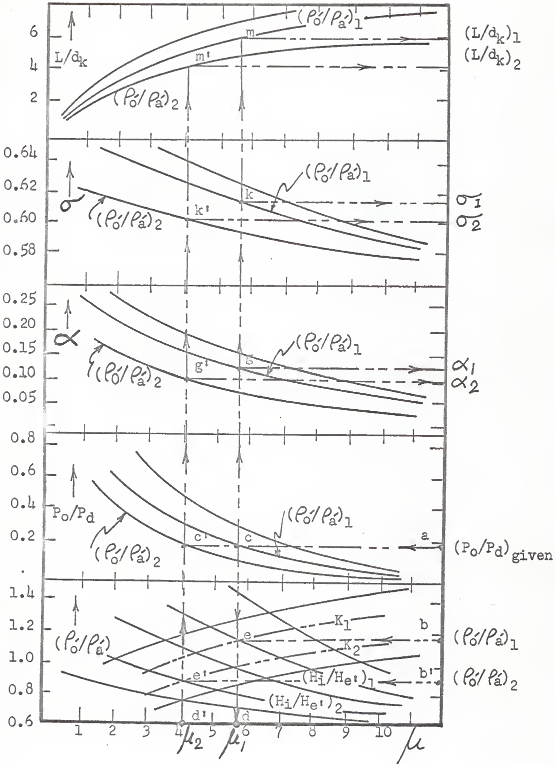

Fig. 8 The expected form of the design chart which can be used to obtain the optimum values of the different parameters when M_o, P_e, P_o, P_d and T_a (or h_a) are specified.

from point (b) intersects the line c-d at point (e). Point (e) lies on the $(H_1/H_{e'})_1$ and the $(K)_1$ curves. An upward vertical from point (c) intersects the $(\rho_{o'}/\rho_{a'})$- curves of the upper diagrams in points (g), (k) and (m). These points give the recommended values of the parameters $(\alpha)_1$, $(\sigma)_1$ and $(L/d_k)_1$, respectively.

2- Knowing h_a (or T_a), P_o and P_d, the isentropic enthalpy change, H_1, of the induced stream when separately compressed from P_o to P_d can be obtained (Fig. 9). Consequently, $H_{e'1}$ ($= H_1/(H_1/H_{e'})_1$)

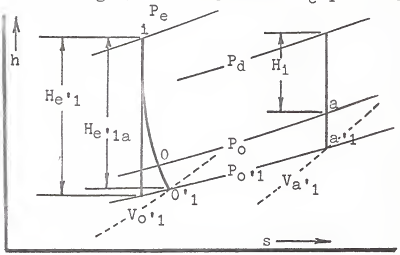

Fig. 9 Estimation of $V_{o'1}$ and $V_{a'1}$ for the given $(H_1/H_{e'})_1$ ratio.

will be estimated. Since the initial condition of the forcing medium is given, which is represented by point (i) on the h-s chart, Fig. 9, the average pressure at the combining cone inlet, $P_{o'}$, can now be estimated.

3- A proper selection of the forcing nozzle efficiency (inside the nozzle and along the distance (X'), Fig. 1) will locate point o'_1 which gives the the properties of the forcing stream at the combining cone inlet section. Thus $H_{e'1a}$ and $\rho_{o'1}$ ($= 1/V_{o'1}$) are obtained. The specific volume of the induced stream at section o'-o', $V_{a'_1}$, can now be calculated ($V_{a'_1} = V_{o'_1} \times (\rho_{o'}/\rho_{a'})_1$). The intersection of the $V_{a'_1}$-curve with the $P_{o'_1}$-curve on the h-s chart (Fig. 9) locates point (a'_1), which specifies the properties of the induced stream at the combining cone inlet section.

4- Since

$$K_1 = \frac{1}{\mu_2} \frac{h_{ta'}}{h_{to'}} = \frac{1}{\mu_2} \frac{h_{a'} + \alpha_1^2 H_{e'1a}}{h_1 + H_{e'1a}} \qquad \text{by definition,}$$

then, by substituting the values of h_1, $H_{e'1a}$, $h_{a'1}$ (measured from the h-s chart), and $(\alpha)_1$ and $(K)_1$ (obtained from the design chart of Fig. 8) in the above equation, the mass ratio $(\mu)_2$ can be calculated. If $(\mu)_2 \neq (\mu)_1$, which is recommended by the design chart, then the assumed value of $(\rho_{o'}/\rho_{a'})_1$ is not correct. In this case, a second approximation, $(\rho_{o'}/\rho_{a'})_2$ should be made.

5- Locate point (d') on the design chart, which corresponds to $(\mu) = (\mu)_2$. An upward vertical line from point (d') intersects the horizontal line a-c (or its extension) at point (c'). Point (c') lies on the $(\rho_{o'}/\rho_{a'})_2$-curve. This new value of the density ratio will be considered as a second approximation for the actual density ratio at the cone inlet section. A horizontal line from point (b'), which corresponds to $(\rho_{o'}/\rho_{a'})_2$, intersects the line c'-d' at point (e'). Point (e') gives the new recommended values: $(K)_2$ and $(H_i/H_{e'})_2$. An upward vertical line from point (c') intersects the curves of the upper diagrams at points (g'), (k') and (m'), which lie on the $(\rho_{o'}/\rho_{a'})_2$-curves of these diagrams. These points give the new recommended values, $(\alpha)_2$, $(\sigma)_2$ and $(L/d_k)_2$.

6- Proceed as in steps 2, 3 and 4 to calculate $(\mu)_3$. If $(\mu)_3 \neq (\mu)_2$, a third approximation should be made in a similar manner. In other words, better values of (μ), approaching the optimum value, may be obtained by successive approximations.

Use of the Recommended Parameters in Constructing a Complete Apparatus

When the optimum values of the different parameters are finally obtained, the designer can use them as a guide to construct a complete apparatus:

a- By knowing M_a $(=M_o/\mu)$, $P_{o'}$, $\rho_{o'}$, $\rho_{a'}$, $u_{o'}$ and $u_{a'}$ the inlet cross-sectional area of the combining cone can be estimated.

b- The length-to-diameter ratio, L/d_k, of the parallel section recommended by the chart for the given specifications, corresponds to a section with a sufficient length to provide a complete mixing operation between the two streams. Accordingly, a homogeneous mixture should appear at its end section. The velocity u_n, which is calculated from $(\sigma)_{opti} = u_n/u_{o'}$, will then be the velocity of a homogeneous mixture at the parallel section end section (section n-n, Fig. 1). This justifies the use of the continuity equation at that section to calculate the throat diameter (d_k) and hence the parallel section length can be determined $(=d_k(L/d_k))$. The major difficulty which will face the designer is to obtain the correct average value of the density, or the specific volume, of the mixture at section n-n, especially when the two fluids are dissimilar.

c- The extent of the possible pressure drop from P_o to $P_{o'}$ along the distance (X'), Fig. 1, will guide the designer in locating the best axial position of the forcing nozzle inside the mixing chamber. In some cases, when the initial velocity of the induced medium is relatively high, a rapid mixing operation between the two streams takes place along the distance (X'), Fig. 1 or Fig. 10, As a result, P_o will start to decrease along (X') and then start to increase. Accordingly, $P_{o'}$ at the cone inlet section will be higher than P_o. Estimation of the nozzle position in this case is

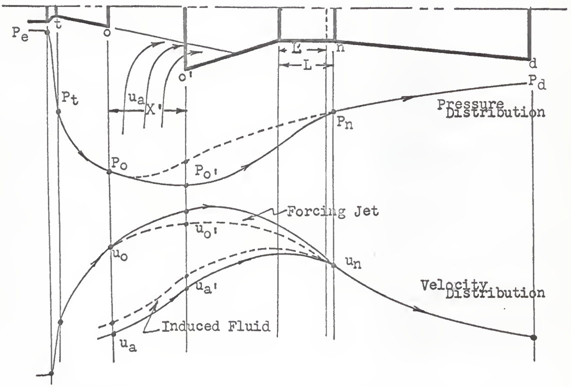

Fig. 10 Pressure and velocity distributions along the different sections of the ejector. When u_a is relatively high, the two mediums intermix rapidly and a relatively short length of the parallel section, L', will be required.

rather difficult. Figure 10 illustrates the possible two cases which are liable to take place. When the design calculations give $P_{o'} = P_o$, the nozzle position will depend in this case on the induced stream properties at the cone inlet section. A relatively small velocity and density of the induced stream, u_a and ρ_a, indicate that the forcing nozzle may lie inside the combining cone. More details on this point are beyond the depth of this report.

It should be noticed, that with further investigations, it may be found necessary to use different design charts, similar to the suggested one, for cases when the two streams are dissimilar. For example, characteristic curves for steam-operated ejectors have been investigated by the author and found to deviate from those obtained when air, under the same operating conditions, is used for air or vapor removal. Cases in which the forcing medium is a gas and the induced one is a liquid are unknown to the author. For cases in which the forcing medium is a liquid and the induced one is a gas, the density is far removed from unity, and a separate chart, similar to the one described above but covering other ranges of values of the parameters, will have to be constructed.

Finally, the author likes to add that the impact operations are of an uncertain and elusive nature. An ejector can be designed to give a good performance by using this set of suggested characteristic curves since they are obtained from experimental investigations and from the design data of successfuly manufactured ejectors . The correct choice of the nozzle, the combining cone and the diffuser angles is also very important. More improvements in the performance can be attained when the apparatus is carefully manufactured. A close tolerance, to insure accurate alignment of the forcing nozzle with the other sections, as well as finely finished surfaces and clear smooth passages for the induced load should be made.

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A SUGGESTED SOLUTION

FOR THE EJECTOR PROBLEM

by

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M. S., Ain Shams University, Egypt, 1956

AN ABSTRACT FOR A MASTER'S REPORT

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Major Professor

This report presents a method for obtaining the suitable values of the different variables involved in the ejector problem, for any given specifications.

The approach extends a previous experimental investigation carried out by the author. The optimum design values of the different variables were presented in terms of non-dimensional parameters related to each other and plotted in the form of characteristic curves. The purpose was to furnish ready data for the design and expected performance of ejectors within the range of operating conditions investigated.

For a more universal use the curves are extended to cases where the densities and stagnation enthalpies of the two streams at impingement are not the same. Two additional non-dimensional parameters are suggested for this purpose. Besides, a third parameter is proposed to account for the possible pressure change inside the mixing chamber after the two streams intermix. This will give a better insight into the extent of the partial mixing operation which takes place between the two streams before they enter the combining cone.

Since the flow operation inside the ejector cannot be subjected to strict mathematical analyses, it is suggested that the construction of a design chart be based on the performance data of many types of ejectors in practice and by actual measurements from experiments. Compiling optimum information about the design and performance of successful ejectors, and presenting it in the form of the suggested non-dimensional parameters, will widen the scope covered by the chart and render it useful in designing a complete apparatus.

A survey of the published literature and a brief discussion on the ejector's principle, are also presented.