N70-17357

NASA CR 86319

HIGH SPEED - HIGH FLOW PUMPING SYSTEMS

John A. Wilson

DEPARTMENT OF MECHANICAL ENGINEERING
UNIVERSITY OF NEW HAMPSHIRE
DURHAM, NEW HAMPSHIRE

Contract NAS 12-2141

FINAL REPORT

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Program Director and Technical Monitor

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Purpose of Study

There is a need to develop means of increasing the rotating speeds of pumping system components to be used in future generation aircraft. The requirement of increased speeds is motivated by changes and improvements in aircraft power generating and processing systems. Successful applications of these new electrical power processing systems will depend largely upon the development of compatible power consuming and accessory devices. The characteristic high frequency of these systems is most readily adapted to electric motors that run at high speed. The use of high speed motors for direct drive of fuel pumps is dependent upon the availability of pumping techniques with reliable high speed operating characteristics.

The earlier stages of this investigation consisted of a review of the recent literature dealing with high performance pumping methods and their related problems, in particular cavitation, and a review of the current state of the art as found in typical applications today. In this way a reasonable point of departure was established and what would appear to be the most promising areas for further study and possible eventual development could be identified.

Because no single specific design situation was considered, a broader study was undertaken which would allow evaluation of more basic concepts, and more generalized results were thus obtained. The area of application did suggest some natural constraints however, and rough guidelines regarding system performance requirements and fuel characteristics were thus available.

The development of high performance pumping systems has been largely an evolutionary process with both theory and empiricism assuming significant roles. The state of the art, specifically with respect to aircraft fuel systems, is just that - an art, because many of the phenomena involved are, even now, poorly understood.

A review of the recent literature reveals very little about aircraft fuel systems, their sometimes unique operational requirements or problem areas. Considerable competition between industrial organizations concerned with this rather specialized area is largely responsible for this absence of available information. Nevertheless, the literature does contain numerous studies wherein a predominantly classical analysis of the behavior of turbomachinery has been extended to include pumps, or inducers with high suction specific speeds that operate at low net positive suction head, NPSH. Because of the degree of sophistication to which this kind of analysis on both inducers and conventional impellers has progressed, it is highly unlikely that further significant advances in the operating characteristics of either device will be obtained without some major departure from their traditional application. It is with this as a guiding assumption that this study has dealt with and attempted evaluation of other less familiar and sometimes novel approaches to the solution of pumping at high speed and/or from low NPSH. It is also obvious that continued investigation of new ideas and imaginative application of basic concepts holds the most promise if further major improvements are to be obtained.

Brief Description of Typical Aircraft Fuel System

In general jet aircraft fuel systems contain at least two types of fuel pumps. These are engine mounted high pressure pumps and tank mounted booster pumps.

Each engine has one engine mounted pump that delivers fuel to the main combustion chamber. It is a positive displacement gear type pump and runs at a design speed of about 6000 rpm. Typical maximum flow rate capability is under 100 gpm although the larger designs under development, such as the SST, have flow rates up to 250 gpm. The delivery pressure can go as high as 1100 psi above the inlet pressure. Because of the wide range of power output required by a jet engine there is an accompanying wide range of required fuel flow. The range of required fuel flow is usually expressed in terms of a turn down ratio which is defined as the required maximum flow rate divided by the operating flow rate. This ratio can easily exceed values of 15:1. Because a gear pump is a positive displacement pump with its design flow rate equal to the maximum flow required, unneeded fuel is returned to the pump inlet by the fuel control which is mounted downstream of the pump.

Each engine mounted pump has a centrifugal or axial flow boost stage, usually engine mounted and running at 6000 rpm that receives fuel and pressurizes it to supply sufficient NPSH to the high pressure main gear pump. These boost pumps have typical pressure rises of 80 to 100 psi.

The engine mounted boost pumps receive fuel from tank mounted boost pumps which supply them with adequate NPSH to operate satisfactorily at all altitudes. The individual flow rates of tank boost pumps appear to have no relation to engine pump flow rates because there are more of them, they are often manifolded together to create redundancy and they have the ability to deliver fuel from all tanks. Representative pressure rises are around 18 psi. Cavitation is a problem for tank mounted boost pumps because the height of fuel above them can become as low as 5 to 10 inches and the fuel in the tank can be boiling, especially at high altitudes and high rates of climb.

The advantage of using a fixed displacement pump for the main high pressure supply is its ability to develop high heads at low speeds. This is particularly necessary at cranking speeds which are about 10% of design speed. The fixed displacement pump can deliver adequate pressure at this speed whereas a centrifugal pump can develop only 1% of its design pressure rise at 10% of design speed unless independently driven. Early gas turbine fuel systems employed fixed displacement pumps and engine manufacturers have not been persuaded to change this proven, dependable, long life design. This type of pump was used in early afterburner designs also because of the development background that it had. However, for the higher afterburner flows encountered in current engines the weight advantage of the centrifugal pump was recognized and it is now used extensively for afterburner applications. Typical speeds of centrifugal afterburner pumps range up to 30,000 rpm. The low torque associated with high speed centrifugal pumps also permits a weight reduction in their drive mechanism. Another important advantage of the centrifugal pump is that dependence upon fuel lubricated sliding surfaces is eliminated thus offering a high degree of reliability from this standpoint.

The difficulties encountered in adopting high speed centrifugal pumps to main engine fuel pump operation are most closely associated with two requirements. These are the development of a suitable means of handling turn down ratios without excess fuel heating and the provision of adequate NPSH to prevent cavitation damage. Because no single specific system design specification was assumed in this study, methods of overcoming either one as well as possibly both of the above requirements were deemed suitable for investigation. In this way chances were maximized that some concept that would prove adaptable to at least a portion of future fuel system designs would evolve. Concepts that were considered include vapor core pumps, injection of high energy swirling flow in pump inlets, jet pumps, cavitation separation devices and shear force impellers. Because this study is concerned with the adaptation of pumps to high speed electric motor drives, it is assumed that the problem of adequate pressure at engine cranking speeds is absent. That is, the pumps are assumed running before start up or that independent starting fuel systems are a part of the prime mover. The above would have to be true with main power plants lacking a drive shaft and accessory gear pad as would be the case, for example, with a ram jet.

Vapor Core Pumps

For current and future engine applications requiring high fuel flow rates the centrifugal pump offers the greatest advantage. This is evident from figure 1 where a weight comparison of the centrifugal pump with the gear pump operating at similar speeds is shown. Flow rates above 80 gpm are not given simply because this is the extent of the normal range. Only recent designs such as the SST go above 80 gpm and reliable data were not available. While such a comparison can at best only be representative, the promise of further speed increases with the centrifugal pump would result in an even greater reduction in pump weight per unit of flow.

The problem that is encountered with a constant speed centrifugal pump is that the temperature at extreme turn down conditions, such as high altitude and low fuel flow, can become prohibitive. One device which overcomes this problem associated with variable flow rates is the vapor core pump. This principle, while not new, has had little attention in this country and has been developed largely in England. It is now just beginning to attract seriously the active attention of some manufacturers in this country. In this pump, flow into the impeller eye is regulated by a throttle valve whose effect is to produce a core of fuel vapor in the eye, the diameter of the core varying as a function of speed, flow and delivery pressure. Because of this feature it is known as the vapor core pump. It combines the turn down range of a variable speed pump with the light weight and simplicity of a direct drive centrifugal pump.

Normally operating centrifugal pumps running at constant speed have quite high frictional power loss especially in the lower specific speed range which would be required for typical aircraft flow rates and pressures. This power loss manifests itself as a temperature rise in the pumped fuel. While this rise may not be significant at design flow it can be unacceptably high at minimum flow conditions. Maintaining design flow by recirculating some of the fuel back to the inlet simply results in repeated accumlative heating of a portion of the fuel before it enters the engine, with the same unsatisfactory result. In the vapor core pump, with a throttle at the impeller eye, a core of vapor is formed, the diameter of which varies with the demands of the system. With low flows pressure requirements are generally also low and the impeller chamber can be nearly empty. In this way the disk friction loss is reduced thus reducing the associated fuel temperature rise. As the system pressure and flow demand approach maximum value the vapor core decreases or disappears altogether and the pump functions in the conventional manner.

An immediate question that arises when considering such an arrangement is that of cavitation. It should be remembered that cavitation damage occurs when vapor bubbles formed in low pressure inlet regions collapse in the higher pressure sections of the flow through the pump. This problem is normally overcome by providing sufficient head above vapor pressure in the inlet region to prevent bubble formation. By contrast cavitation damage does not occur in the vapor core pump because the two phase flow is not that of bubbles surrounded by liquid but rather that of liquid droplets surrounded by vapor. The inlet throttle valve is located in the eye in such a way that a vapor/liquid interface or free surface is formed within the impeller and fuel crosses the vapor core and enters this surface as a stream of droplets. Beyond this vapor liquid interface the fuel is centrifuged in essentially the normal fashion to the pump delivery. Tests conducted by Dowty Fuel Systems Limited, and described in reference (1)* would suggest that acceptable operation is

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* Numbers in parentheses refer to references listed at the end of this report.

achieved even with gas/liquid mixtures in the inlet line. In other words, the pump seems capable of functioning with a V/L or vapor liquid mixture in its inlet.

Because the impeller does not run full at part load conditions a significant reduction in shaft power is realized in contrast to a conventional centrifugal pump. Additionally, the reduced temperature rise of the fuel passing through the pump provides for additional heat sink capacity of the fuel when maximum engine injector fuel temperature limits are imposed. A plot of the power characteristics of a vapor core pump based on actual tests by Dowty Fuel Systems Limited is shown in figure 2. The design point for the full running pump was at 1000 psi, 21, 000 rpm and about 160 gpm. Major power savings even at only slightly less than maximum pressure are obtained at all reduced flow rates. The fuel temperature rises encountered during these tests are shown in figure 3.

Some tests have been run in this country by Chandler Evans, Inc. of West Hartford, Connecticut. They report (2) that with minor changes in mechanical design of impeller fluid passages a design free of any cavitation erosion was obtained.

The inlet pressurization requirement of the vapor core pump is about 30 psi above fuel vapor pressure when operating in the vapor core mode. When operating in the full running condition it should have a suction specific speed limit near that of any other centrifugal pump and a reasonable figure is 8000 using gpm units for flow rate. The NPSH requirement is then dependent upon rotating speed and flow rate. These requirements for conventional design pumps with an assigned suction specific speed of 8000 are shown in figure 4.

The vapor core pump lends itself to possible adaptation to systems having thrust augmentation or afterburners. The incorporation of the fuel requirement of this subsystem into the main engine fuel pump specification merely extends the range of turn down ratios and the addition of a separate pump is unnecessary. A simple numerical example will help to develop this point. Let Q_m and Q_R represent the maximum main engine and afterburner flows respectively and R the main engine turn down ratio under maximum engine speed or afterburner operating condiitons. Then for a single pump delivering both flows the overall turn down ratio is

$$\frac{\text{max. combined flow}}{\text{min. engine flow}} = \frac{Q_m + Q_R}{Q_m/R} = R(1 + \frac{Q_R}{Q_m})$$

A typical value of R is 5 and of Q_R/Q_m is 4.6. The resulting combined ratio is

$$R \quad (1 + \frac{Q_R}{Q_m}) = 5(5.6) = 28$$

Within this value the main engine ratio of 5 could be accompanied by an afterburner ratio anywhere in its range which can get as high as 100 to 1. Normally afterburner pumps require special valves at inlet as they run continuously and are pumped dry with closed inlets when the afterburner is shut off. Deriving afterburner flow from the main pump, whose normal pressure rise at maximum flow is suitable for this function, provides a tangible reduction in pump system complexity.

Inlet Flow Conditioning Using Swirling Jet Flows

One of the basic needs of any high speed centrifugal pump whether it be a conventionally operating centrifugal type or one which is full running only part of the time with vapor core operation during the remainder is that of adequate net positive suction head at the inlet. This NPSH is provided by boost stage pumps or inducers which have the capability of operating at very low and even sometimes vanishing NPSH and developing sufficient head to meet the NPSH requirements of downstream high pressure stages. If the developed head of boost stages can be increased then the operating speed of the following downstream stage can be increased also. One means of increasing the head developed by boost stages is by increasing their rotating speed. In order to do this however an increase in suction specific speed will be necessary. Assume that suction specific speed values can be increased for a given boost stage and represent the higher value by primed quantities. Then, if the boost inlet NPSH remains constant and the specific speed, N_s, is maintained constant by suitable modification in blade shape the following can be written where B refers to boost stage.

$$(N_s')_B = (N_s)_B$$

or equivalently $$\left(\frac{N'\sqrt{Q'}}{(H')^{3/4}}\right)_B = \left(\frac{N\sqrt{Q}}{H^{3/4}}\right)_B$$

where N is speed in rpm, Q is flow rate in gpm and H is head.

Rearranging $$\left(\frac{H'}{H}\right)_B = \left(\frac{N'\sqrt{Q'}}{N\sqrt{Q}}\right)_B^{4/3}$$

But the high pressure stages have essentially constant suction specific speeds so that (with P referring to main pump stage)

$$S_P' = S_P$$

or equivalently

$$\left(\frac{N'\sqrt{Q'}}{(H_{sv}')^{3/4}}\right)_P = \left(\frac{N\sqrt{Q}}{H_{sv}^{3/4}}\right)_P$$

where H_{sv} is the net positive suction head. Assuming that $(H'_{sv})_p = H_B'$ and that $(H_{sv})_p = H_B$

that is, that the boost stage H_{sv} is small compared to its head rise, then

$$\left(\frac{N'}{N}\right)_P = \left(\frac{\sqrt{Q}}{\sqrt{Q'}}\right)_P \left(\frac{H_{sv}'}{H_{sv}}\right)_P^{3/4} = \left(\frac{H_{sv}'}{H_{sv}}\right)_P^{3/4}$$

$$= \left(\frac{H'}{H}\right)_B^{3/4} = \left(\frac{N'\sqrt{Q'}}{N\sqrt{Q}}\right)_B = \left(\frac{S' H_{sv}'^{3/4}}{S \cdot H_{sv}^{3/4}}\right)_B$$

or with no change in the boost stage H_{sv}

$$\left(\frac{N'}{N}\right)_P = \left(\frac{S'}{S}\right)_B \qquad \ldots\ldots 1$$

In other words the limiting ratio of final to original speed for the main pump equals the ratio of improved to original suction specific speed of the boost stage.

With this result as at least an approximate indication of the relationship of main pump speed improvement to boost suction specific speed increase it seems quite natural to consider methods of improving boost stage suction specific speed. One method that has attracted only modest attention in the past (3) and yet seems worthy of further consideration is that of injecting, in an essentially tangential manner, some of the higher pressure boost delivery flow into the inlet of the boost stage. In this way a deliberate swirling motion is set up in the inlet flow. With proper location and shape of the injection pattern suitable control over the distribution and strength of the swirling flow should be obtained. Swirling flow in the inlet will result in two beneficial changes in inlet flow. The relative velocity between the impeller or inducer blade leading edges will be reduced with swirl in the direction of blade rotation and a radial pressure gradient will be set up causing an increase in pressure toward the blade tips where the cavitation speed limiting problem predominates. As an initial supposition in the analysis of this situation the swirling flow is assumed to begin at some fraction, x, of the outer radius of the inducer or at $x = r/r_o$, and the distribution of swirl from this point on outward to r_o is taken as linear. Consideration of other studies involving swirling flow has shown this to be a reasonable approximation.

Admittedly the influence of the wall boundary layer is omitted and there will be a tendency for such a velocity profile to redistribute the point of maximum velocity inward away from the wall as it progresses downstream in the flow. It is expected that the region under consideration here occurs within one diameter of the swirling jet input and that some control over the mixing profile will be obtainable from the jet geometry in this range.

As a first step the radial pressure distribution resulting from this kind of swirl pattern should be developed. The tangential or swirling velocity at any point between xr_o and r_o is given by

$$v_\theta = V_\theta \frac{r - xr_o}{r_o - xr_o} = \frac{V_\theta}{r_o}\left(\frac{r - xr_o}{1 - x}\right) \quad \ldots\ldots \quad 2$$

where V_θ is the maximum swirling velocity, at r_o.

Assuming inviscid, incompressible flow, the radial equilibrium equation is

$$\frac{dp}{dr} = \frac{\gamma}{g}\frac{v_\theta^2}{r}$$

where γ is fluid specific weight.

Substituting the equation for v_θ gives

$$\frac{dp}{dr} = \frac{\gamma}{g}\frac{V_\theta^2}{r_o^2}\frac{1}{r}\left[\frac{r^2 - 2xrr_o + x^2r_o^2}{(1-x)^2}\right]$$

Integrating between xr_o and r_o and dividing by γ the result gives the increase in head between the non swirling region and the outer edge of the swirl region at the blade tips, that is, the swirling head at the blade tips, H_{sw}.

$$H_{sw} = \frac{V_\theta^2}{g}\frac{1}{(1-x)^2}\left[\frac{1}{2} + \frac{3x^2}{2} - 2x - x^2 \ln x\right] \quad \ldots\ldots \quad 3$$

Note that when x equals zero the swirl extends to the center of the inlet line and the usual result of $V_\theta^2/2g$ associated with solid body rotation is obtained.

The result shown in equation 3 will be non dimensionalized by dividing both sides by $V_\theta^2/2g$ and calling the result B.

$$B = \frac{H_{sw}(2g)}{V_\theta^2} = \frac{2}{(1-x)^2}\left[\frac{1}{2} + \frac{3x^2}{2} - 2x - x^2 \ln x\right] \quad \ldots\ldots \quad 4$$

B is the fraction of solid body head at the blade tips which results from a linear swirl distribution starting at xr_o. The result is shown plotted in figure 5. The dimensionless parameter B will be useful in subsequent analyses.

Attention is now directed toward obtaining the optimum diameter for minimum net positive suction head requirement or equivalently maximum speed of rotation or maximum suction specific speed.

The ideas discussed here are based on the following simple equation for cavitation in turbopumps

$$H_i = K\frac{W^2}{2g} + H_v \qquad \ldots\ldots \quad 5$$

where H_i is the static head before the entrance to the blading, K is a vane cavitation coefficient, W is the fluid relative velocity before entering the blade passages and H_v is fluid vapor pressure. The equation states that the static head just before entering the blading should be sufficiently above vapor pressure to allow for a pressure drop upon entering the blades and avoid cavitation or allow only limited cavitation as governed by the value of K. K for incipient cavitation is defined as

$$(1+K) = \left(\frac{W_B}{W}\right)^2$$

where W_B is the relative velocity of the fluid at the minimum pressure point within the blades.

With inlet swirl the static head, H_i, at the blade tip is equal to the net positive suction head upstream of the boost stage, minus the velocity head in the inlet line, plus the increase radially in pressure due to the swirling pattern in the flow, plus vapor pressure.

$$H_i = H_{sv} - \frac{V_a^2}{2g} + B\frac{V_\theta^2}{2g} + H_v \qquad \ldots\ldots \quad 6$$

where V_a is the inlet axial velocity, V_θ is the swirling velocity at the blade tips and B is taken from figure 5 and based upon the inward extent of the swirl pattern.

The inlet tip relative velocity W is obtained from the velocity vector diagram, figure 6, and the relationship which will be used in this development is

$$W^2 = V_a^2 + (U_t - V_\theta)^2 \qquad \ldots\ldots \quad 7$$

Making substitution for H_i and W^2 in equation 6 and rearranging gives

$$H_{sv} = \frac{V_a^2}{2g}(1+K) + K\frac{(U_t - V_\theta)^2}{2g} - B\frac{V_\theta^2}{2g} \qquad \ldots\ldots \quad 8$$

A dimensionless swirl number $\lambda = V_\theta/U_t$ will prove useful and introducing this into equation 8 results in

$$H_{sv} = \frac{V_a^2}{2g}(1+K) + \frac{K U_t^2}{2g}(1-\lambda)^2 - \frac{B U_t^2 \lambda^2}{2g} \qquad \ldots\ldots \quad 9$$

Because the purpose of this part of the analysis is to obtain optimum inlet diameters and resulting suction specific speeds the above equation must be written in terms of inlet diameter where appropriate. Accordingly

$$V_a = \frac{1}{450}\,\frac{4Q}{\pi d^2} = C_1\frac{1}{d^2}$$

where Q is volume flow rate in gpm and C_1 is a simplification to help decrease the number of symbols used. Similarly

$$U_t = \frac{\pi N}{60}\,d = C_2\,d$$

where N is inducer speed in rpm and C_2 is introduced for economy of terms. With these changes equation 9 becomes

$$H_{sv} = \frac{C_1^2}{2g}(1+K)\frac{1}{d^4} + C_2^2\frac{[K(1-\lambda)^2 - B\lambda^2]}{2g}d^2 \qquad \ldots\ldots \quad 10$$

Differentiating with respect to d to obtain the diameter allowing minimum H_{sv}

$$\frac{\partial H_{sv}}{\partial d} = -\frac{4C_1^2}{2g}(1+K)\frac{1}{d^5} + 2C_2^2\frac{[K(1-\lambda)^2 - B\lambda^2]}{2g}d$$

and setting this equal to zero and solving for d gives

$$d_{opt} = \left[\frac{2C_1^2(1+K)}{C_2^2[K(1-\lambda)^2 - B\lambda^2]}\right]^{\frac{1}{6}} = .091\left[\frac{Q^2(1+K)}{\left(\frac{N}{100}\right)^2[K(1-\lambda)^2 - B\lambda^2]}\right]^{\frac{1}{6}} \quad \ldots \quad 11$$

with Q in gpm.

Substitution of this back into equation 10 for H_{sv} gives

$$(H_{sv})_{opt} = A(1+K)^{\frac{1}{3}}[K(1-\lambda)^2 - B\lambda^2]^{2/3} N^{4/3} Q^{2/3} \quad \ldots \quad 12$$

where A is a numerical constant.

The corresponding optimum suction specific speed is

$$S_{opt} = \frac{5060}{(1+K)^{1/4}[K(1-\lambda)^2 - B\lambda^2]^{\frac{1}{2}}} \quad \ldots \quad 13$$

The equation for optimum S_s for the case of zero swirl reduces to

$$S = \frac{5060}{(1+K)^{1/4} K^{1/2}}$$

which is identical to results obtained in the literature, (4), (5) in studies without inlet swirl.

Before it is possible to assess the effect of preswirl on suction specific speed it is necessary to properly match the value of λ and the value of B or equivalently the strength and inward radial extent of swirling flow distribution. In other words for any given amount of swirl at the tip there is a definite value of r into which the swirl pattern should extend so that the worse cavitation situation does not occur at radii inside of the swirl pattern on the one hand and so that the swirl pattern does not extend in further than necessary on the other. It should be recognized that what follows simply matches the cavitation condition at the two end points of the swirl pattern and does not consider points in between. This is the only possibility if, as is the case here, a linear swirl distribution is assumed. Later analyses in this study develop equations for the swirl distribution that maintains constant cavitation conditions throughout the entire swirl region but the result is close to linear and its complexity does not warrant introduction into the analysis at this time.

In terms of parameters in the blade tip region the net positive suction head H_{sv} was given by equation 8 as

$$H_{sv} = \frac{V_a^2}{2g} + K_t \frac{\left[V_a^2 + (U_t - V_\theta)^2\right]}{2g} - B\frac{V_\theta^2}{2g}$$

At the radius xr_o where swirling flow starts the suction head is given as follows

$$H_{sv} = \frac{V_a^2}{2g} + K_x\left[\frac{V_a^2 + U_x^2}{2g}\right] \quad \ldots\ldots 14$$

where K_t and K_x refer to blade coefficients at the tip and the swirl starting point respectively and U_x is the blade tangential velocity at radius xr_o. Equating these two expressions for H_{sv} will require in essence that the swirling motion at the tip be just sufficient to match flow conditions at xr_o in order that the upstream H_{sv} requirement be the same in both cases. This gives the following result

$$K_t\left[V_a^2 + (U_t - V_\theta)^2\right] - B V_\theta^2 = K_x\left[V_a^2 + U_x^2\right] \quad \ldots 15$$

Assuming that K_t and K_x are nearly the same and simply calling them K gives

$$K(U_t - V_\theta)^2 - B V_\theta^2 = K U_x^2$$

Dividing both sides by KU_t^2 gives

$$(1-\lambda)^2 - \frac{B\lambda^2}{K} = \left(\frac{U_x}{U_t}\right)^2$$

but $\left(\frac{U_x}{U_t}\right)^2 = x^2$

thus an equation relating λ, B and x is obtained.

$$(1-\lambda)^2 - \frac{B\lambda^2}{K} = x^2 \qquad \ldots . \; 16$$

B and x are not independent however as they are related by equation 4. This means that equation 16 is really the required relationship between λ and x. The result is shown graphically in figure 7 for various values of K typical of high suction specific speeds. In other words for a given blade coefficient, matched values of swirl starting point x and swirl strength λ are shown. Using these values of λ and x and for a given x the corresponding B value for figure 5 the optimum suction specific speed, equation 13, can be obtained. This result is shown in figure 8 again for various values of K. Note that a proper swirl strength which starts at x = .5 essentially doubles suction specific speeds. Recalling that the ideal increase in main pump speed is related to the increase in boost stage suction specific speed it is clear that worthwhile main pump speed improvements should result from jet induced swirl in the inlet to boost stages.

At this point it is appropriate to consider the best swirl velocity distribution between xr_o and r_o. That is, instead of merely matching the end points as is the case with a linear distribution the optimum pattern giving V_θ as f(r) should be obtained.

If the swirl pattern is properly matched at all radii beyond xr_o then the difference between the static head in the inlet line ($H_{sv} - \frac{V_a{}^2}{2g}$) and the pressure inside the blades, or the extent of cavitation when present, will be the same at any radius greater than xr_o. That is

$$\left[K\frac{W^2}{2g} - h_{sw}\right]_{xr_o<r<r_o} = \left.K\frac{W^2}{2g}\right]_{xr_o} \qquad \ldots . \; 17$$

where h_{sw} is the head due to the swirl at any radius r. Call this quantity representing either a head difference or an indication of the extent of cavitation, when it is present, ΔH_B. Then, using equation 7 in a form for any r,

$$\Delta H_B = K\frac{W^2}{2g} - h_{sw} = K\frac{\left[V_a^2 + (u_x - \mathcal{V}_\theta)^2\right]}{2g} - h_{sw}$$

but $\quad u_x = \frac{r}{r_o} u_t$

Thus $$\Delta H_B = K\frac{V_a^2}{2g} + K\frac{\left(\frac{r}{r_o}u_t - \mathcal{V}_\theta\right)^2}{2g} - h_{sw} \qquad \ldots 18$$

Dividing both sides by u_t^2 will result in an equation in terms of λ_r where $\lambda_r = \frac{\mathcal{V}_\theta}{u_t}$ is a local value insofar as $\mathcal{V}_\theta$ is a local tangential velocity.

$$\frac{\Delta H_B}{u_t^2} = K\frac{V_a^2}{u_t^2}\frac{1}{2g} + \frac{K\left(\frac{r}{r_o} - \lambda_r\right)^2}{2g} - \frac{h_{sw}}{u_t^2}$$

If ΔH_B is to be held constant between xr_o and r_o then its derivative with respect to r will be zero in this region. Differentiating then gives

$$\frac{1}{u_t^2}\frac{d\Delta H_B}{dr} = \frac{K\left(\frac{r}{r_o} - \lambda_r\right)\left(\frac{1}{r_o} - \frac{d\lambda_r}{dr}\right)}{g} - \frac{1}{u_t^2}\frac{dh_{sw}}{dr} = 0$$

but $$\frac{dh_{sw}}{dr} = \frac{\mathcal{V}_\theta^2}{gr}$$

and after substitution there is obtained

$$K\left(\frac{r}{r_o} - \lambda_r\right)\left(\frac{1}{r_o} - \frac{d\lambda_r}{dr}\right) - \frac{\lambda_r^2}{r} = 0 \qquad \ldots 19$$

or $$K\frac{r}{r_o^2} - K\frac{r}{r_o}\frac{d\lambda_r}{dr} - K\frac{\lambda_r}{r_o} + K\lambda_r\frac{d\lambda_r}{dr} - \frac{\lambda_r^2}{r} = 0$$

The second and third terms can be combined and the fourth term contracted to give

$$K\frac{d}{dr}\left[\frac{\lambda_r^2}{2} - \frac{r}{r_o}\lambda_r\right] = \frac{\lambda_r^2}{r} - K\frac{r}{r_o^2} \qquad \ldots 20$$

The solution of this equation is a transcendental relationship between r and λ. The results for three values of K are shown in figure 9 where the swirl at the blade tip is plotted against swirl starting point x. A comparison of figure 9 with 7 shows that the optimum matched pattern does not posses the peaking characteristic for the tip swirl strength obtained in the earlier development. In addition, the maximum values are somewhat less than those obtained for the linear case. In figure 10, for K = .05, the swirling velocity component distribution for several starting points is shown. Figure 11 compares the optimum matched swirl pattern with the assumed linear distribution used earlier. At $r = xr_0$, λ_r is zero and equation 19 then gives

$$\left(\frac{1}{r_0} - \frac{d\lambda_r}{dr}\right) = 0$$

Notice that this requires a steeper swirl velocity gradient at the swirl starting point than with a linear distribution. Accordingly swirl velocities will be higher at the inner end of the swirl region and a greater radial pressure rise results. This in turn results in the somewhat lower tip swirling requirements.

One possible method of obtaining swirling inlet flows may be to inject an anulus of swirling flow through a ring nozzle arrangement as shown in figure 10. The relative size and position of the ring ahead of the inducer should provide some flexibility with respect to obtaining an approximation of the best swirl pattern at the inducer blade edges.

The jet momentum requirements will now be considered. Using the symbols shown in figure 12 the angular momentum flux entering the flow from the ring nozzle is given by

$$\dot{H} = 2\pi \rho t r_R^2 V_{zj} V_{\theta j} \qquad \ldots 22$$

where V_{zj} is the jet axial velocity, $V_{\theta j}$ is jet tangential velocity coming from a relatively thin annulus of width t and mean radius r_j.

At the inducer inlet plane a linear swirl velocity will again be assumed increasing from zero at xr_o to V_θ at r_o.

The angular momentum flux for a differential element, dr, in the swirl pattern is given by

$$d\dot{H} = 2\pi\rho r^2 V_{zm} v_\theta \, dr \qquad \ldots\ldots \quad 23$$

substituting v_θ from equation 2 gives

$$d\dot{H} = 2\pi\rho V_{zm} r^2 \left(\frac{r - xr_o}{r_o - xr_o}\right) V_\theta \, dr \qquad \ldots\ldots \quad 24$$

where V_{zm} is the axial velocity of the mixed flow or V_a. For simplicity the axial component velocities of both jet and main flow will be assumed equal. In reality the jet axial should be somewhat higher to resist backflow.

Integrating between xr_o and r_o gives the total angular momentum of the swirling mixed flow.

$$\dot{H} = \pi\rho V_{zm} V_\theta r_o^3 \frac{(3 - x - x^2 - x^3)}{6} \qquad \ldots\ldots \quad 25$$

Equating this angular momentum with that of the annular jet before the mixing process an expression for the required jet velocity for a given V_θ (or conversely λ if U_t is known) is obtained

$$V_{\theta j} = \frac{V_{zm} V_\theta r_o^3 (3 - x - x^2 - x^3)}{12 t r_R^2 V_{zj}} \qquad \ldots\ldots \quad 26$$

However

$$Q_j = 2\pi r_R t V_{zj} \qquad \text{and} \qquad Q = \pi r_o^2 V_{zm}$$

where Q_j is the jet flow rate and Q is the total flow entering the inducer. With these substitutions

$$V_{\theta j} = \frac{V_\theta r_o (3 - x - x^2 - x^3)}{6 r_R} \left[\frac{Q + Q_j}{Q_j}\right] \qquad \ldots 27$$

Application and the presentation of a specific example utilizing the above described technique is difficult, without a specific application in mind. Qualitatively however, one possible application that seems apparent is in conjunction with a conventional centrifugal pump or the vapor core pump. With a regular pump or the vapor core pump running at design flow, that is in the non vapor cored mode, suction heads in excess of 200 ft are necessary in order to obtain speeds above 20,000 rpm and with flow rates from 200 to 400 gallons per minute. The increased heads developed while applying inlet swirl to boost stages may make the use of two stages of mechanical boost unnecessary. Additionally, the application of jet induced preswirl, as developed in reference (3), to situations involving variable flow rates makes its use in combination with centrifugal vapor core main pumps seem logical. There is a wide range in what current pump manufacturers seem to feel is a maximum obtainable speed by centrifugal pumps. The one common denominator seems to be that cavitation is responsible in some way as the limiting phenomenon. Certainly increased NPSH without the mechanical problems associated with multiple staging will contribute to the solution of this problem.

As an aid to sensing relative magnitudes of the quantities involved the following example is given. An inducer with a flow rate of 100 gpm and a speed of 11,000 rpm will be assumed to have a blade cavitation coefficient K of .025. It is desired to apply inlet swirl and to raise the suction specific speed, in theory, by a factor of two. This will be done by increasing the flow by 25% and using this excess flow as nozzle flow to the swirl generating region ahead of the inducer. Let primed quantities represent conditions with swirl and unprimed quantities the original conditions. With no change in NPSH and s' = 2(s) the following equation gives the new speed

$$N' = 2N\left(\frac{Q}{Q'}\right)^{\frac{1}{2}} = 2\,(11{,}000)\left(\frac{1}{1.25}\right)^{\frac{1}{2}} = 20{,}000$$

The optimum diameter, assuming swirl in to x = .5 and λ = .17 as found from figure 7 and which is necessary for s'/s = 2 gives an optimum diameter of .185 feet or slightly under 2.25 inches. The blade tip speed is 193 fps and the required swirling flow at the tips has a velocity of 33 fps. The diameter of the jet ring will naturally influence the required jet velocity but if the ring is assumed halfway between xr_o and r_o or at $.75r_o$ an initial jet velocity of about 75 fps is required. This is slightly less than jet velocities used in the swirling jet induced flow described in reference (3). Ideally a jet velocity of 75 fps can be developed by a pressure drop of about 90 feet thus the range of boost stage pressure being sought to provide main pump inlet pressurization for high speed operation should be about right for bleed off to obtain jet velocitiies of this approximate magnitude. Note that this modification doubles the suction specific speed of the boost stage and in the limit would double the speed of any following centrifugal pump. Admittedly, departures from such idealized theoretical predictions are to be expected but if further development substantiates the theoretical considerations presented here, application of jet-induced preswirl will prove useful in some form over a range of low inlet NPSH.

The fluid mechanics problem involved in the flow mixing which takes place between the swirling jet and the axial fuel flow is highly complex.

The flow is turbulent and an analytical solution describing the kinematics of the mixing process would require a solution of the Navier Stokes equations with turbulent or Reynolds stresses included. Because of the complexity of the problem it would seem that an experimental investigation in this area will be required in order to most effectively study the nature of the mixing process. Wong's paper (3) is one that describes the result of some effort in this area, and it has virtually no analytical treatment. It is essentially limited to the presentation of results of some experimental studies while acknowledging the difficulties involved in analytically attempting to describe the mixing flow.

Cavitation and Inlet Swirl

One problem that does not lend itself to a theoretical solution because of the inherent complexity and number of unknowns is that of necessary levels of NPSH upstream of the swirling jet injection point. Mixing must occur without creating unacceptable levels of cavitation in the inlet flow caused by the mixing turbulence. Because of the radial pressure field set up in the swirling mixture the tendency to cavitate should be less than in linear jet pump mixing. The study of preswirl conducted by Wong et. al. (3) did not give attention to this point. In fact, a study giving any ducted jet pump cavitation inception parameter could not be found in the literature. There are some studies (6), (7), (8), (9) that consider a head breakdown parameter for linear jet pumps but that situation is closely related to choking two phase flow in the secondary or pumped flow inlet to the mixing zone and a situation closely paralleling that does not exist in the arrangement of the swirling jet. Rouse, (6), did an inception parameter study for a jet issuing into an open filled tank. Comparison of his results with photographic records in reference (7) would suggest at least qualitatively that cavitation inception does not occur nearly as readily in ducted as in free jets. As in the past, with other jet forms a careful experimental program would be required to develop reliable prediction parameters for rotating flows.

Ordinary cavitation is a real problem for tank mounted boost pumps, and other design criteria are used in place of suction specific speed. These involve limiting values of blade tip speed based upon blade angles and fluid thermodynamic properties. Because of the uncertainties surrounding swirling jets and the resulting development or suppression of cavitation before actual pump inlets at very low and zero NPSH the question remains if this concept can be applied to tank boost pumps. Serious experimental investigation will be necessary to answer this question. Such experimentation may show possible application of the technique to a cavitation separation device where a clear liquid flow with a small static head could be delivered through ports or a volute collecting ring around the outside of the mixing region while vapor and any evolved gasses are vented out along the axis. The liquid flow would then serve as inlet to a single boost stage upstream of a main centrifugal or vapor core pump. A single boost stage in this context may require inlet swirl to increase its head generating capacity to meet main stage NPSH requirements and/or make it better suited to handling off design flow rates.

If conventional tank boost pump practice is adhered to, a second boost stage would be required before a main centrifugal pump. The application of inlet swirl to this second boost stage would allow use of higher speeds on this as well as following main stages and/or a means of decreasing recirculated excess fuel flow at the second stage by improving cavitation resistance at off design flow rates.

Additional Methods and Concepts

Linear Jet Pump

When considering various methods of using jets of high energy flow to achieve increased boost capability or high pump speed the linear jet pump cannot be overlooked. The linear jet pump has been used as a boost stage in some small aircraft where low flow rates and some positive NPSH in the fuel tanks will allow its use. The basic problem which develops as NPSH in the secondary or pumped flow becomes too small is that of choking in the secondary throat passage. The sonic velocities in two phase mixtures, like a cavitating flow, can become very low with only small amounts of vapor present. With decreasing secondary NPSH the flow will cavitate first in the mixing section of the pump, but as NPSH goes down the two phase region progresses back to the pump throat where choking limits the maximum secondary flow rate to values below the desired flow rate of the pump. This of course eliminates the application of the linear jet pump as a tank boost stage when NPSH has virtually disappeared under high altitude conditions. One other factor which limits the use of linear jet pumps is concerned with space requirements. In order to obtain mixing and diffusion with proper head recovery a length of from five to ten pipe diameters is often necessary. This is prohibitive in some applications.

Another characteristic of the linear jet pump is the high ratio of jet flow to pumped flow required for even modest efficiencies and flow rates from very low NPSH. This ratio is on the order of one to one for a typical application to a low pump flow rate (8 gpm) and pressure rise (18 psi). When considering flow rates in the hundreds of gallons per minute range the desirability of alternative solutions is clearly evident.

With a system where a low NPSH boost stage is used immediately ahead of a main high pressure pump the possibility of using a linear jet pump to raise the boost inlet head should be considered. This is essentially the same approach suggested before using jet induced swirling inlet flow. The principal difference here is that the inlet head to the boost stage is raised uniformly throughout the entire inlet region and there is no swirl component to reduce fluid relative velocities with respect to the impeller or inducer blades. Several trial calculations showed that similar amounts of jet flow from the same high pressure source were only capable of accomplishing approximately half of the improvement obtainable with swirling jets. This may be due to the fact that in the swirling flow case the jet energy is distributed spatially where it is needed and not wasted along the center of the inlet flow where it is of no value. The linear jet does not lend itself to such a selective distribution of its energy.

Pumping from Low NPSH Using Shear Force Impellers

Some development has been done (11), (12) on the concept of using viscous shear forces instead of blades in pump impellers. The appearance of such impellers resembles a stack of closely spaced metal disks or washers. Fluid enters axially into the center hole or impeller eye and flows outward between the disks to the diffusing section of the housing surrounding the outer radius of the impeller. As the fluid passes between the disks viscous shear forces give the fluid a tangential motion in the direction of impeller rotation and a head is developed much as in a radial flow pump. The difference lies in the fact that tangential velocities are obtained by shear rather than solid blades.

The inherent cavitation resistance of shear force pumps is primarily due to the absence of the low pressure zones present on the airfoil lifting surfaces of traditionally designed pump blading. This means that a shear force pump is capable of functioning at very low, though finite net positive suction head. The apparent necessity of some finite NPSH results from the fact that as zero NPSH is approached the shear force pumps that have been built and tested, as reported in the literature, tended to accumulate gas or vapor bubbles between the rotating disks in the inlet region and did not pass them through to the outlet. At first this would seem to rule out the use of this type of pump especially in aircraft applications where a definite evolution of dissolved gases must be recognized as a possible inlet flow condition. Because there was no more than passing mention of this failure mode in the available information on this pump, some further exploration of its potential value was felt worthwhile.

The bubble trapping effect is actually the result of an equilibrium set up at some point in the rotor disk space between the outward drag caused by liquid flowing past the bubble and the fluid pressure gradient in the rotor which because of buoyancy effects tends to force the bubble inward toward the axis. It would be helpful to have some idea of the pressure gradients which would be obtainable and still keep the bubble moving outward through the rotor. The problem of analytically determining the drag on a bubble within the confines of the rotor package is highly complex with fluid surface tension interaction with rotor surface properties being a significant unknown. No relevant studies with results applicable to this case could be found in the literature and the initiation of a detailed study was felt beyond the scope of this investigation. As a very rough order of magnitude estimate however, the pressure gradient against which a single bubble in an infinite fluid could be moved by drag forces of surrounding fluid was calculated as follows.

The net pressure force on the bubble is given by

$$-\int_S p\underline{n}\cdot\underline{dA}$$

and this is just equal and opposite to the fluid drag force D at equilibrium. The drag force was assumed similar to that on a sphere in a free stream and given by

$$D = \frac{C_D \rho U^2 A}{2}$$

where C_D is a drag coefficient, U is the free stream velocity, ρ is fluid density and A is the projected area of the sphere. Combining these two forces in an equation of equilibrium and using Gauss's theorem on the surface integral gives

$$-\int_V \nabla p \, dV + \frac{C_D \rho U^2 A}{2} = 0$$

The gradient of p is in the impeller radial direction and given by $\frac{dp}{dr}$. Assuming it to be constant in the vicinity of the bubble gives

$$\frac{dp}{dr}\int_V dV + \frac{C_D \rho U^2 A}{2} = 0$$

or, using r_B for bubble radius

$$\frac{dp}{dr} V_B = \frac{dp}{dr}\frac{4}{3}\pi r_B^3 = C_D \frac{\rho u^2}{2}\pi r_B^2$$

which reduced to, writing $\frac{dp}{dr}$ as a head gradient

$$\frac{dH}{dr} = \frac{C_D}{g}\frac{3u^2}{8 r_B}$$

From numerical results given in references (11) and (12), fluid velocities roughly five feet per second appeared appropriate. A disk spacing of .015 inches was assumed. With a hydrocarbon fuel this gave bubble Reynolds numbers around 1000 and a corresponding drag coefficient, reference (13) page 163, of 1.3. The resulting head gradient is about 350 feet per foot of impeller radius. This would appear to be too small to be useful in generating boost stage heads. The gradient may be large enough for application to some kind of very low head generating chamber or cavitation separation device that could be mounted within a fuel tank. This would represent a significant departure from conventional practice and experimental development may be required before such a device could be reasonably evaluated.

Conclusions

The development of more sophisticated pumping systems with increasingly demanding performance capabilities will be a slow and tedious process. The current state of the art has been achieved only through slow and continued efforts by many individuals and organizations. Improvements will doubtlessly be made but the state of development of currently used methods suggests that major improvements will come only by research into different techniques and substantial modification of traditional practices. A study such as this can only attempt to identify some promising variations or new ideas and to give them some evaluation.

The vapor core pump holds much promise as a high speed main pump capable of handling turn down ratios encountered in aircraft applications and suitable for direct drive by high speed electric motors. The decrease in power requirements over recirculation of excess flow is a substantial benefit. The size and weight requirements over the gear pump are considerable especially in the higher flow ranges. Of any method in the developmental process today that has been discovered during this investigation the vapor core main pump would seem most promising and deserving of continued development.

High speed centrifugal pumps whether conventional or full running vapor core will require high NPSH. The combination of jet induced preswirl used in conjunction with boost impellers or inducers is one way by which additional heads can be developed by boost stages in addition to providing a means of reducing recirculation flow at turn down conditions. Reduced or off design flow rate handling capability has been significantly improved by use of preswirling techniques, (3). The value of less recirculated flow at reduced flow rates will increase with higher pressure boost stages. Further development of swirling inlet flow both for reduced flow handling improvements and greater head development for main pump inlet pressurization should be a worthwhile undertaking.

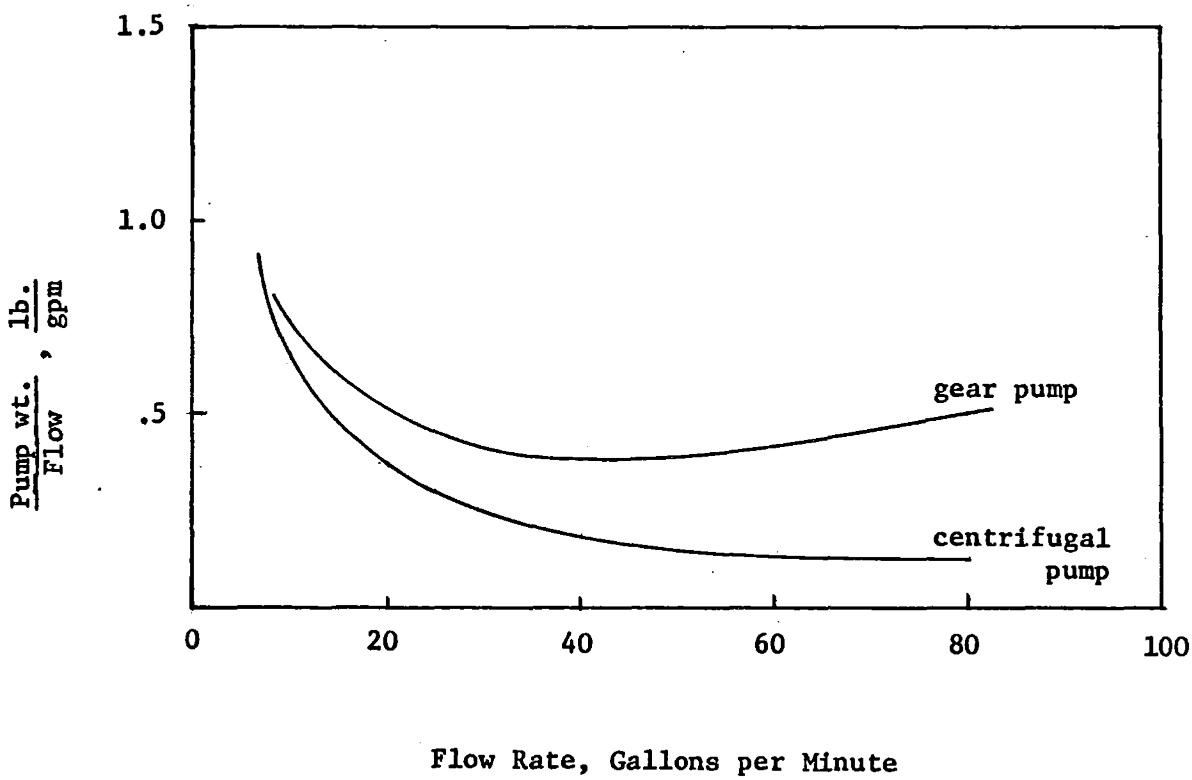


Figure 1. Pump Weight Comparison

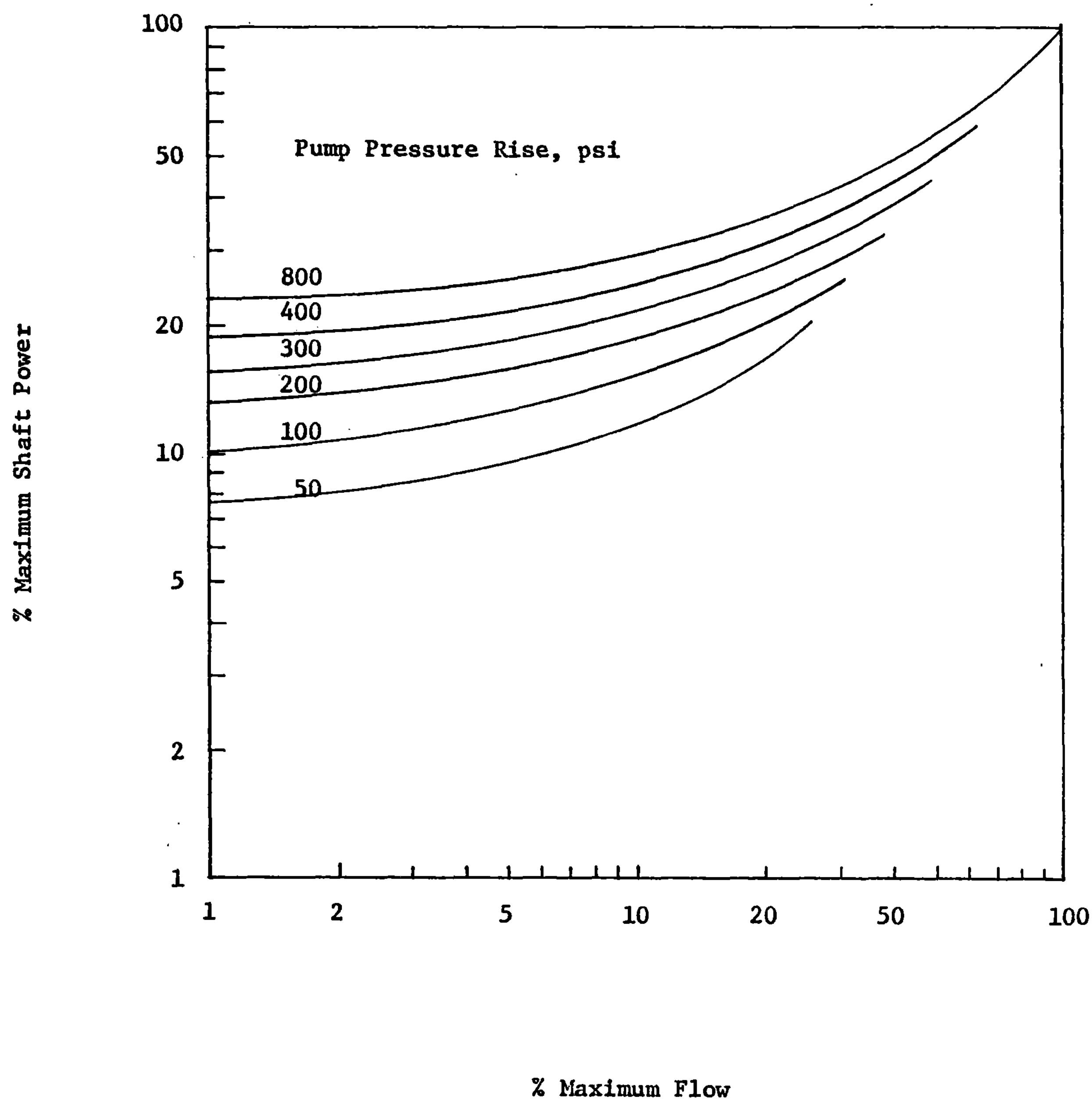


Figure 2. Pump Power vs. Flow Rate at Several Pressures

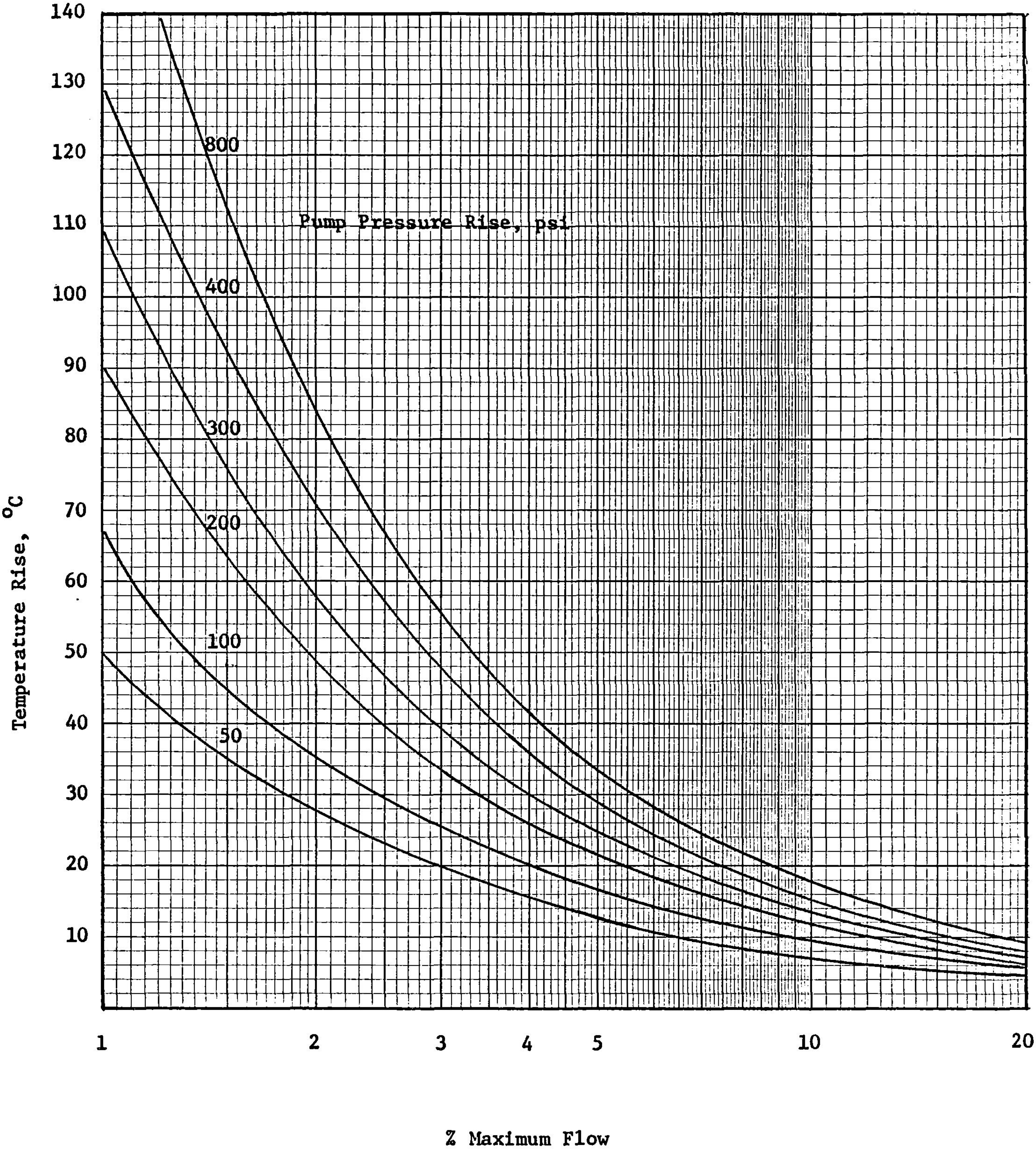


Figure 3. Vapor Core Pump Temperature Rise

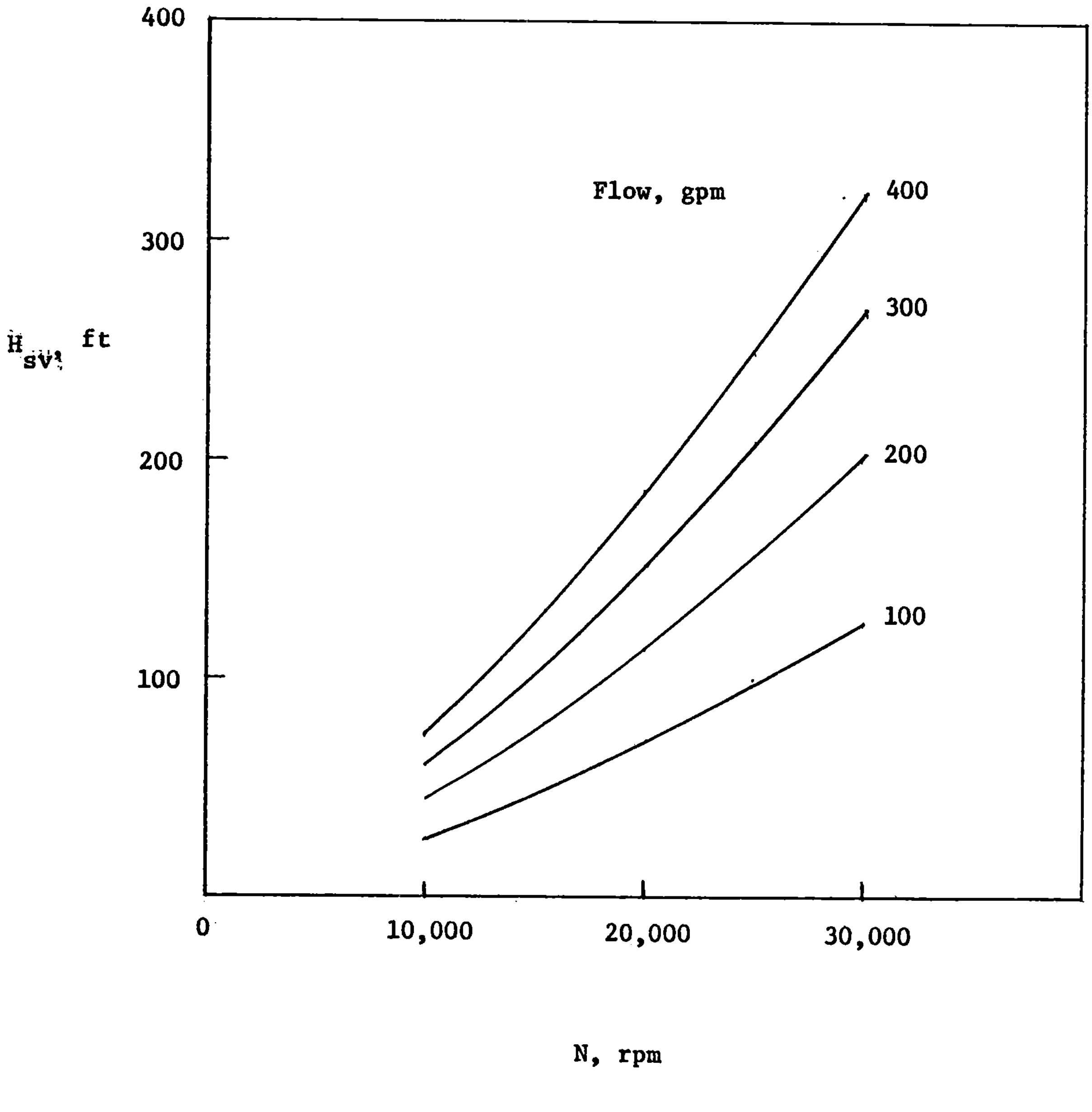


Figure 4. Pump Required NPSH for Various Speeds and Flows
Suction Specific Speed 8,000 (gpm units)

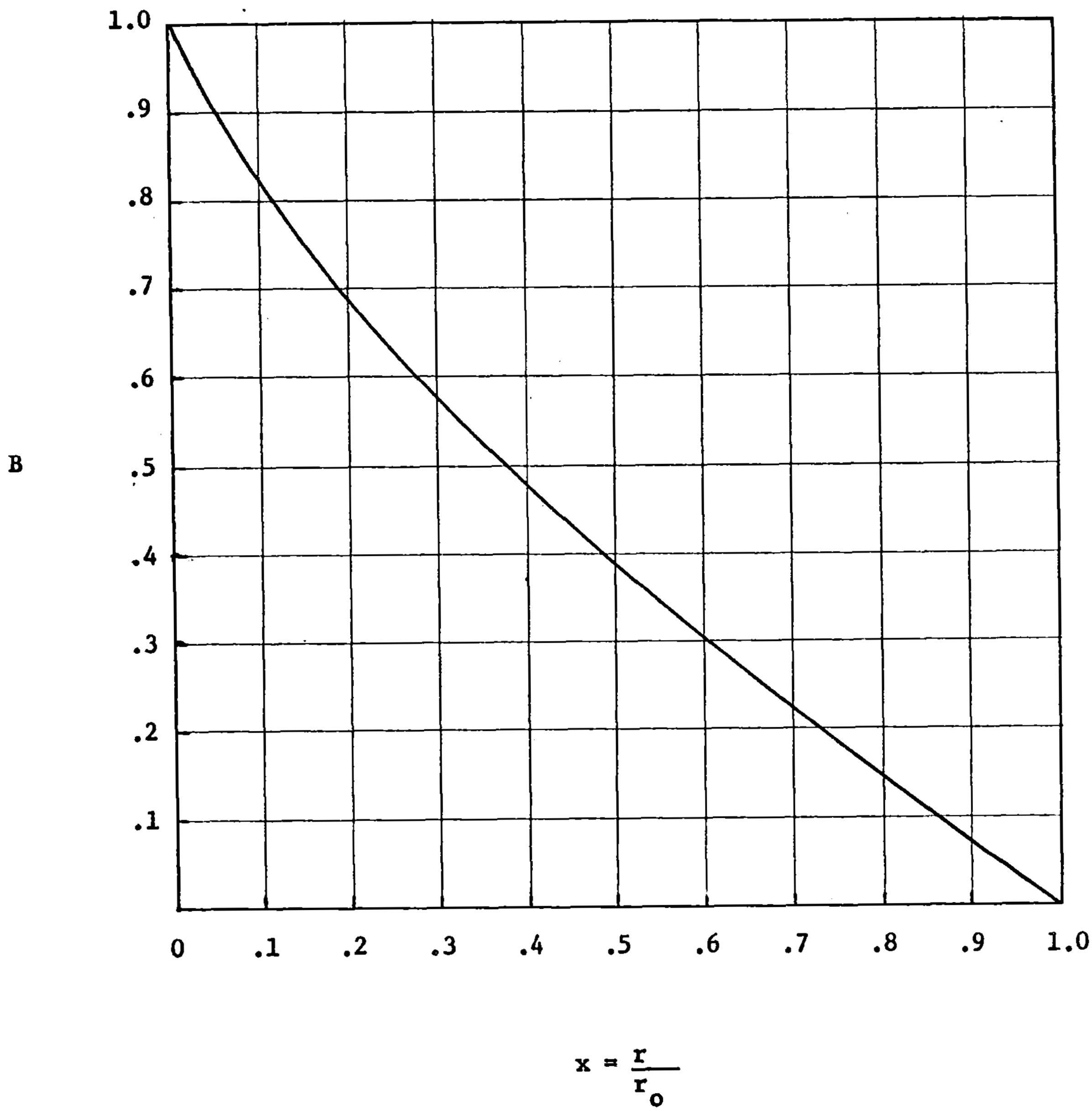
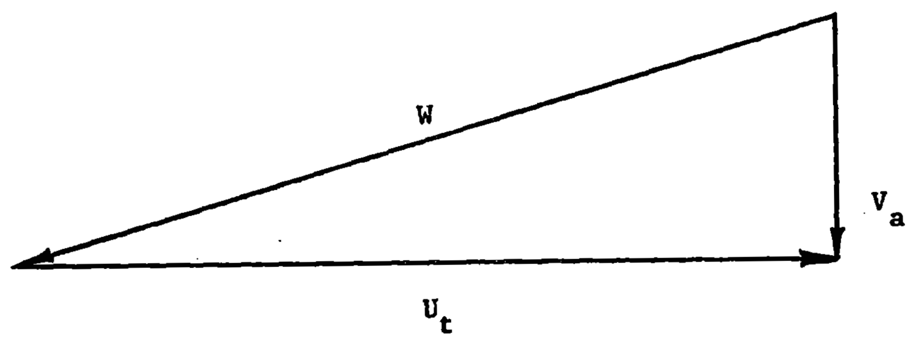
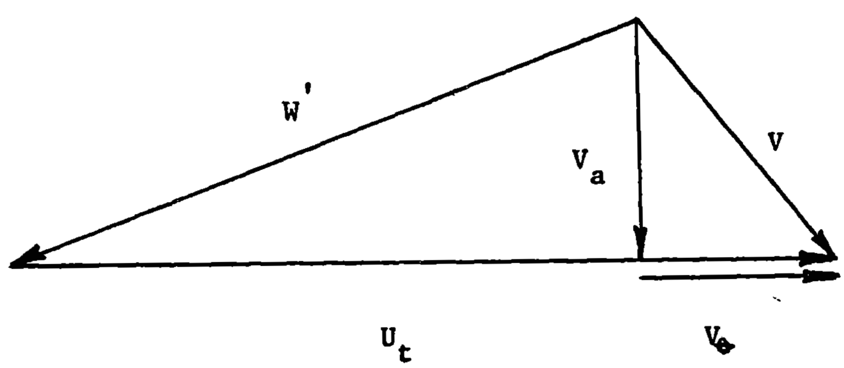


Figure 5. B vs. Swirl Starting Point, x.



$$W^2 = V_a^{\ 2} + U_t^{\ 2}$$

No Preswirl



$$(W')^2 = V_a^{\ 2} + (U_t - V_\phi)^2$$

With Preswirl

Figure 6. Fluid Velocity Diagrams

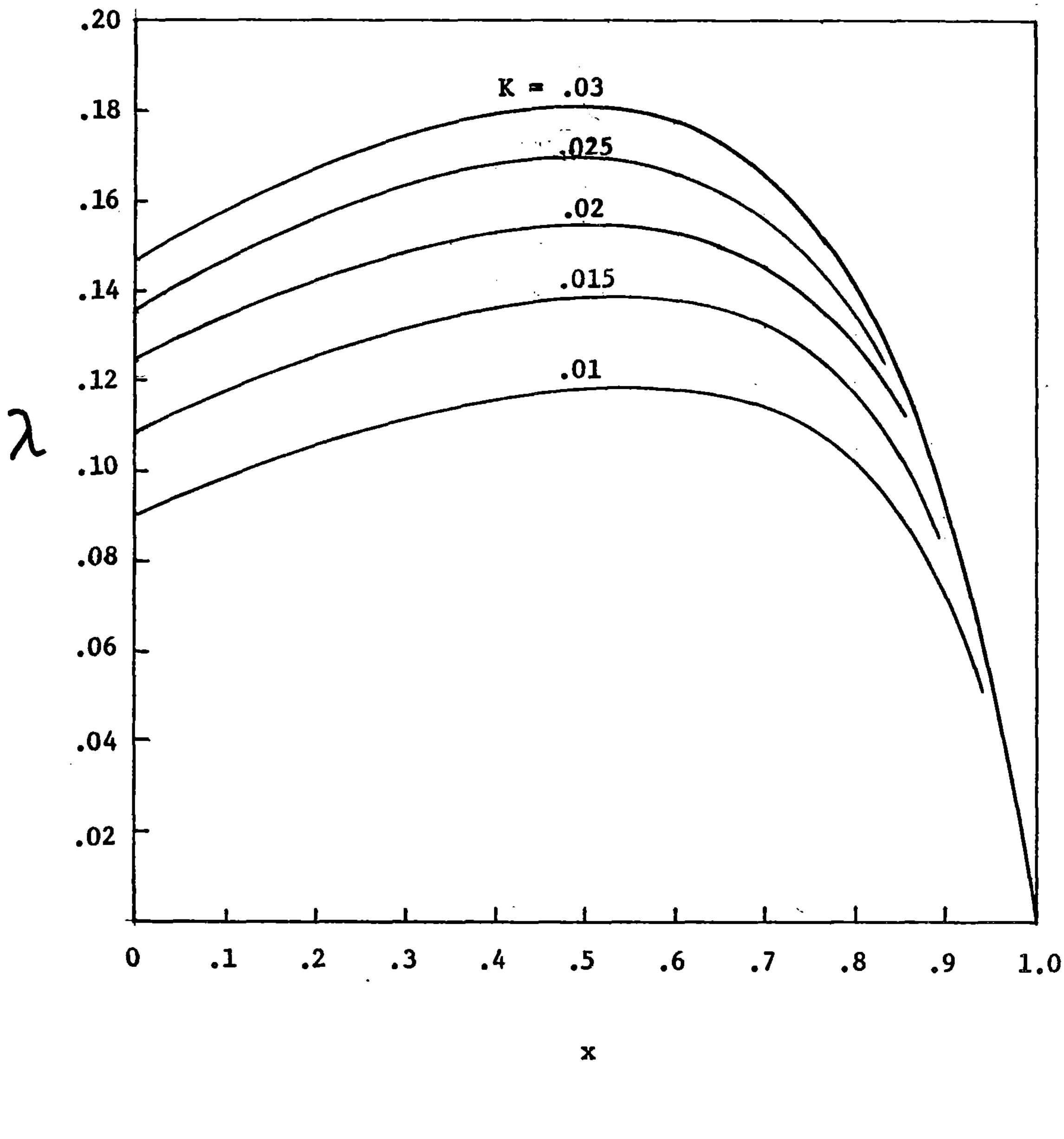


Figure 7. Swirl Strength, λ, vs. Swirl Starting Point, x.

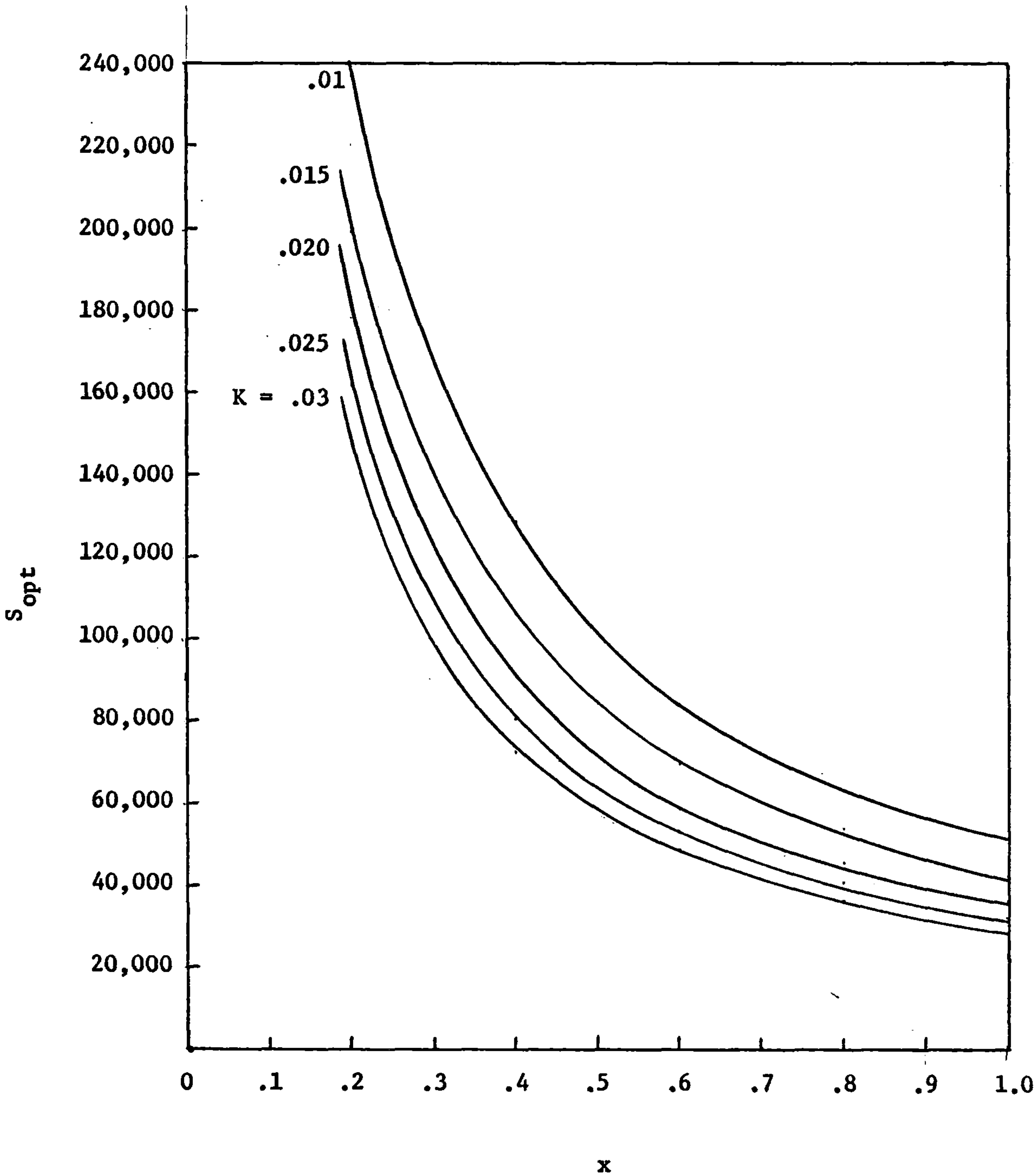
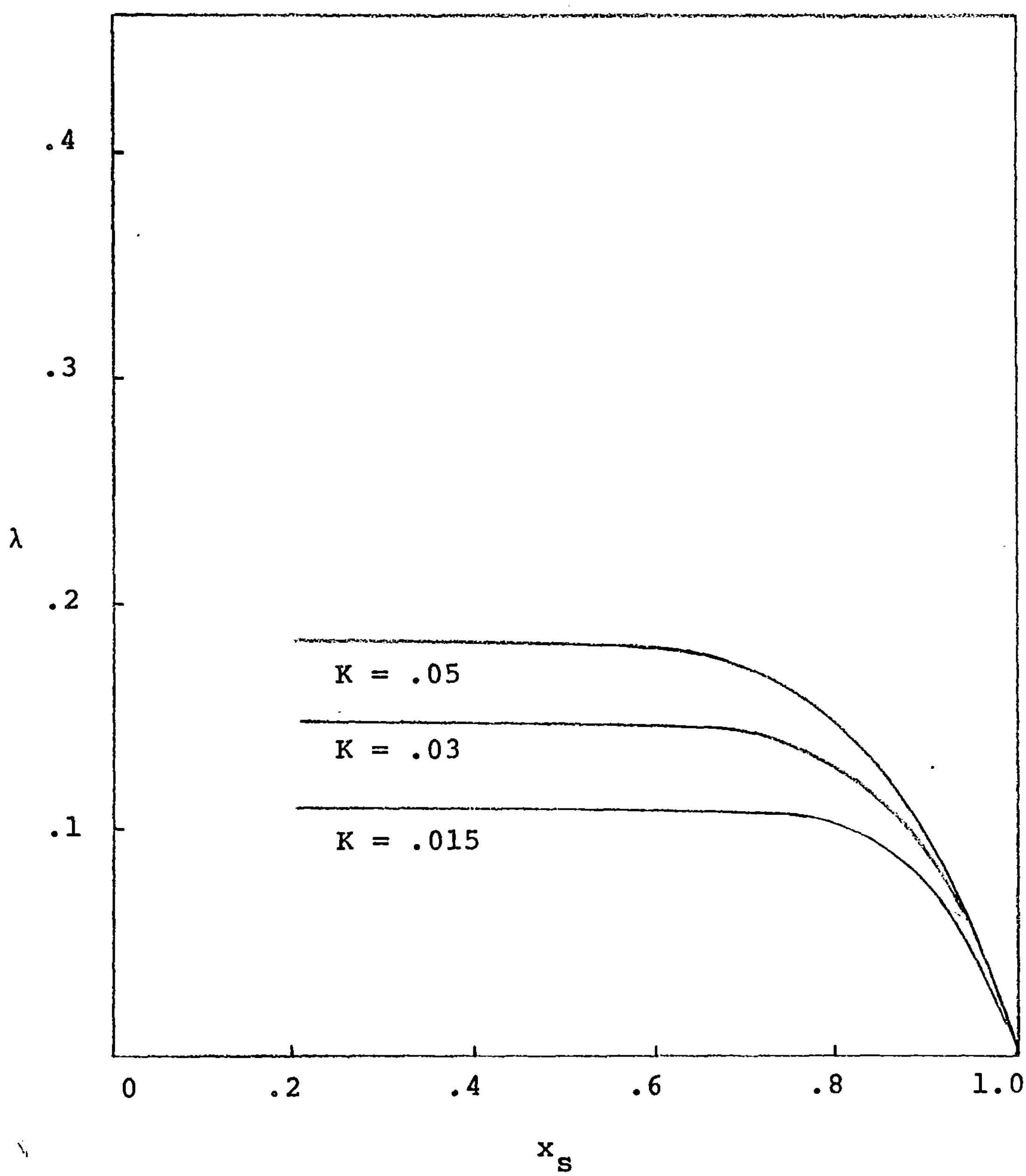
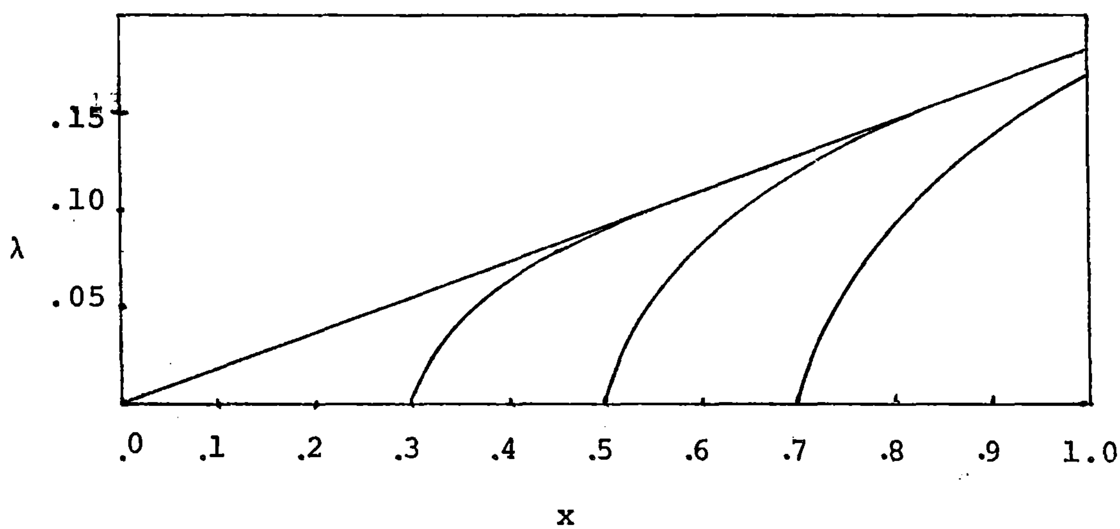


Figure 8. Suction Specific Speed vs. Swirl Starting Point



Swirl Tip Strength vs. Swirl Starting Point

Figure 9



Swirl Distribution for K=.05

Figure 10

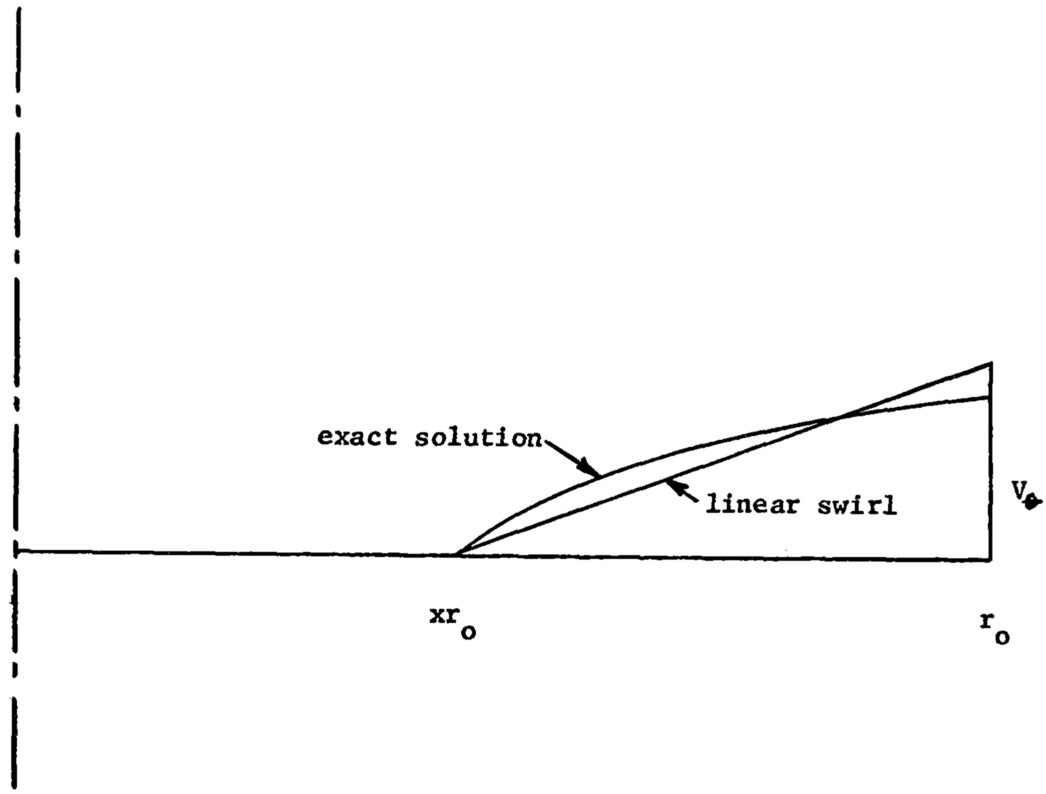


Figure 11. Linear Swirl Velocity Distribution Compared with Optimum Matching Swirl Pattern.

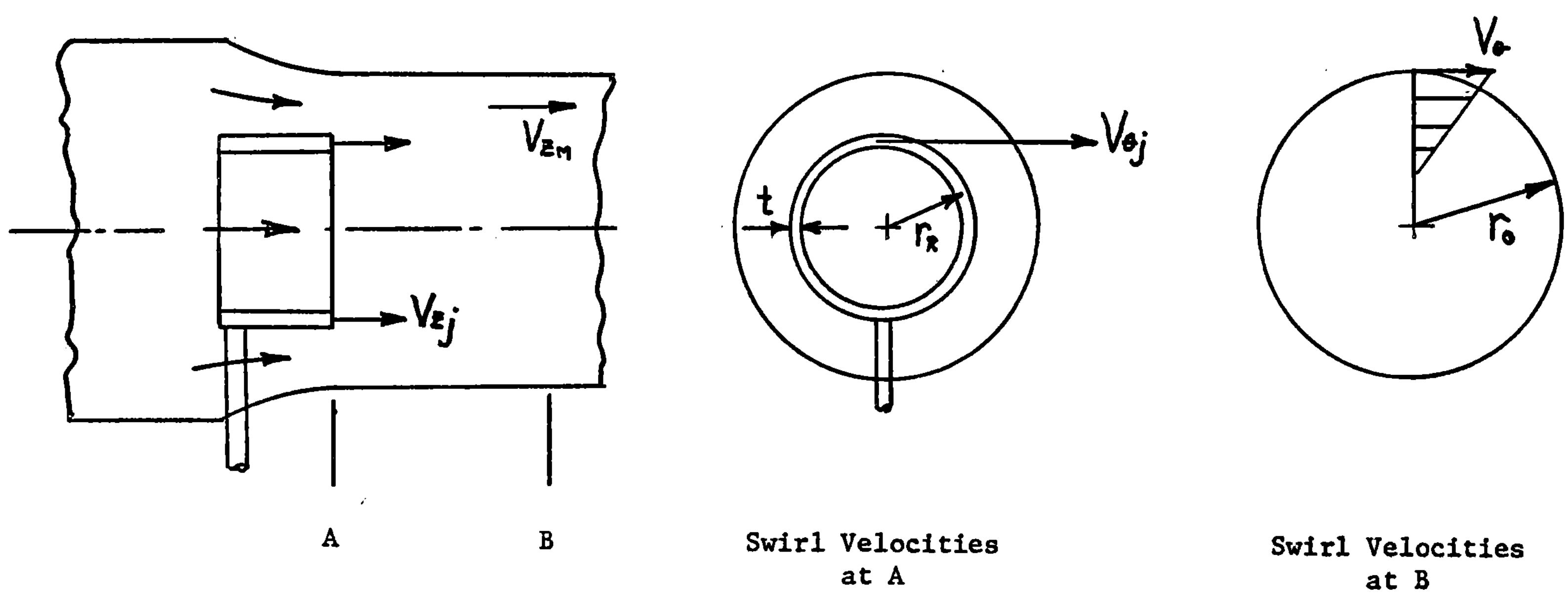


Figure 12. Conceptual Layout of Jet Arrangement

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