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A Survey of Instabilities Within Centrifugal Pumps and Concepts for Improving the Flow Range of Pumps in Rocket Engines

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A SURVEY OF INSTABILITIES WITHIN CENTRIFUGAL PUMPS AND CONCEPTS FOR IMPROVING THE FLOW RANGE OF PUMPS IN ROCKET ENGINES

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ABSTRACT

Design features and concepts that have primary influence on the stable operating flow range of propellant-feed centrifugal turbopumps in a rocket engine are discussed. One of the throttling limitations of a pump-fed rocket engine is the stable operating range of the pump. Several varieties of pump hydraulic instabilities are mentioned. Some pump design criteria are summarized and a qualitative correlation of key parameters to pump stall and surge are referenced. Some of the design criteria were taken from the literature on high pressure ratio centrifugal compressors. Therefore, these have yet to be validated for extending the stable operating flow range of high-head pumps. Casing treatment devices, dynamic fluid-damping plenums, backflow-stabilizing vanes and flow-reinjection techniques are summarized. A planned program has been undertaken at NASA Lewis Research Center to validate these concepts. Technologies developed by this program will be available for the design of turbopumps for advanced space rocket engines for use by NASA Lewis in future space missions where throttling is essential.

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INTRODUCTION

New rocket engines in the 20 000 to 50 000 lb thrust class will be utilized by NASA Lewis for future space-vehicle propulsion and launch-vehicle upper stages. The current candidate is a rocket engine fueled by liquid hydrogen and oxygen.[1] Pressurization of the propellants to high combustion chamber pressure imposes stringent requirements on turbopumps to provide a high performance wide operating range that is free from stall and cavitation. Anticipated space missions could require that engines be throttled to levels as low as 5 percent of design thrust.

Cryogenic pump design historically has focused on meeting the performance goals at a single point or within a narrow operating range. Designs were driven by the performance at the design point and, to some extent, by producibility. Throttling over a broad stable operating range is an additional requirement that may require somewhat of a tradeoff with the performance at the design point.

System analysis has shown that high degrees of engine throttling require the turbopumps to operate stably down to nearly 20 percent of their design flow coefficients. Figure 1(a) illustrates an engine throttle line superimposed onto a pump operating map of head versus flow. The pump is sized to operate at its design point (point 1) for design thrust of the engine. However, throttling the engine typically causes the pump to move away from its design flow coefficient and ultimately intersect with the surge line, point 2 in Figs. 1(a) and (b). The change in flow coefficient results from the relationship between the pump surge line and the engine operating line, or throttle line. These two lines always intersect at some throttled condition. This intersection can be a limiting factor of engine throttling range and appears to be independent of engine cycle. The amount of throttling range gained by system optimization of the engine cycle is limited by the stable flow range of the pump. The most effective way to improve engine throttling range is to provide a pump with a wider stable performance map. As point 3 illustrates, a pump with an improved surge margin can provide more engine throttling capability before the point of intersection of the system and surge lines. The required surge margin of the turbopump imposed by the engine throttling range influences the conceptual design of the pump. To achieve a wide stable operating range, the pump configuration must have less susceptibility to hydraulic instabilities.

Conceptual-design ideas to obtain improved pump operating range, particularly higher surge margin, are summarized. Several of the design concepts have been under development for use in gas turbine compressors and industrial pumps, but have not yet been applied to turbopumps in rocket engines. The goal of stable operation of compressors is the same as for pumps, with the exception of cavitation. In some cases, cavitation appears to be the result of local flow separation and may be a precursor to pump surge. Using compressor technology to prevent, or minimize local separations may delay cavitation and, as a consequence, delay pump surge. Design concepts to minimize or delay local separations include various casing treatment devices, recirculation and the limits on hydraulic loadings within the blading. Because of the many similarities between compressors and pumps, similar improvements in flow range can be expected if these technologies are developed for cryogenic turbopumps.

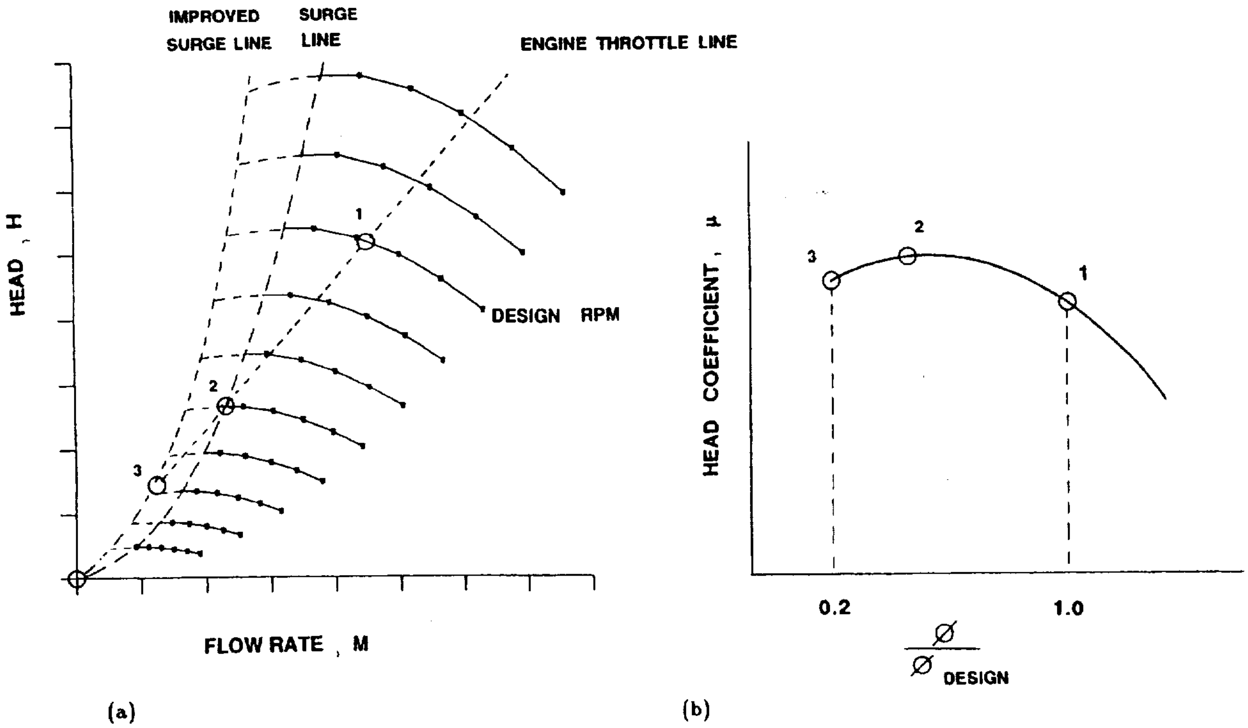


Figure 1. Turbopump operation at the engine design point (1) and typical throttled point (2) on the head - flow map (a) and on the normalized map (b). Improved pump surge margin allows engine deep-throttling to point (3).

PUMP INSTABILITIES

Several types of hydraulic instabilities can occur under the various operating conditions that pumps may experience in a rocket engine. Engine operation is typically chosen to avoid these pump-related instabilities, limiting the throttling range of the engine.

The various types of instabilities that a pump can encounter depend on the pump configuration, inlet conditions and the flow rate. Following is an attempt to categorize the types of pump instabilities. Figure 2 shows a normalized map for a typical pump. The relative location of the known types of pump instabilities are flagged by letters corresponding to the following sections in which they are discussed.

A. Cavitation

Cavitation is one of the primary sources of instability in turbopumps.[2,3] The fundamental cause of cavitation is that the local static pressure approaches the fluid vaporization pressure thereby allowing vapor bubbles, or cavities, to form within the flow. Cavitation is initiated at the pump inlet and can appear in various degrees, depending on the local static pressure and flow rate. The onset of cavitation typically occurs at flow rates that are above the design value, as shown by A in Fig. 2, but is also a strong function of the local static pressure near the inlet of the pump rotor. Flow rates higher than the design value result in increased velocity and decreased static pressure, particularly on the impeller blade suction surface near the throat region. The decreased static pressure causes the onset of cavitation at the throat and ultimately results in preformance deterioration.

The local static pressure at the rotor inlet is also determined in part by the total pressure, or Net Positive Suction Head (NPSH), of the fluid supply. Decreasing the NPSH enables the onset of cavitation to occur at lower flow coefficients (A′). At a fixed value of NPSH, increasing the flow beyond the onset of cavitation results in an increase in the size of the vapor bubbles at the throat, until complete breakdown of performance occurs from choking.[4] This condition is represented in Fig. 2 by the regions to the right of A and A′. Cavitation of this type is an instability that is not related to stall or separation and does not have any dynamic flow reversals.

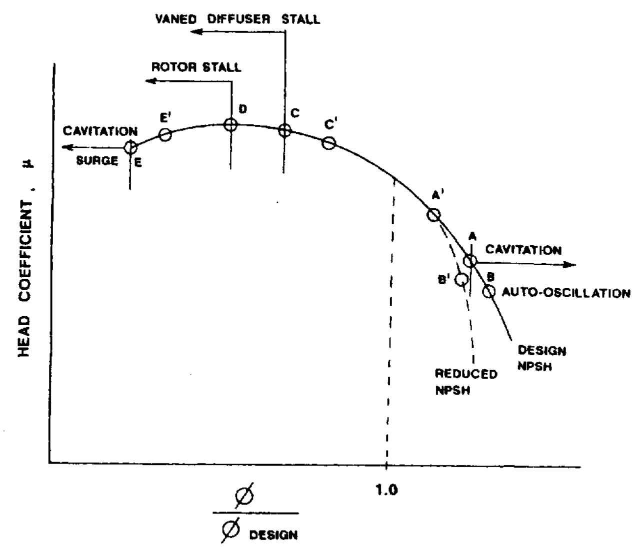
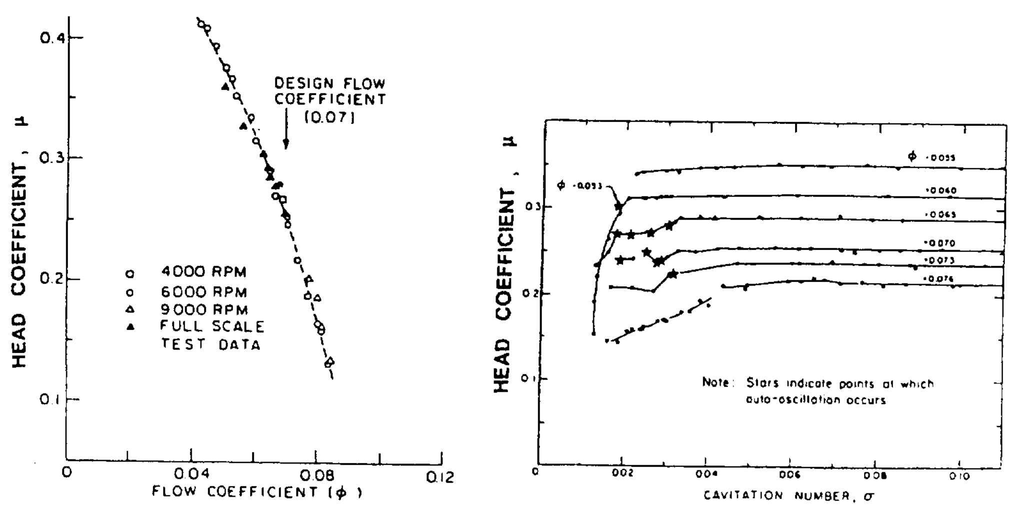


Figure 2. The location of turbopump instabilities along a typical normalized head-flow map. The prime (') designation indicates the onset of the instability at a reduced level of NPSH.

B. Auto-Oscillation Cavitation

As in the previous case, this instability can occur at or above the design flow coefficient. Unlike the previous case, this type of cavitation manifests itself as a dynamic instability with severe flow oscillations. Figures 3(a) and (b) from Ref. 5 show the performance of the pump with and without cavitation.



(a)

(b)

Figure 3. Non-cavitating (a) and cavitating (b) performance of a pump.[5] Stars indicate points at which auto-oscillation occurs.

It can be seen clearly that at lower values of cavitation number, this instability can occur at virtually any flow coefficient, even above the design condition. The relative location of these auto-oscillations is point B in Fig. 2. It appears to the right of the onset of cavitation (A) and is also a function of NPSH. Pump stall is not the primary cause of these auto-oscillations, since it can also occur at operating points above the design flow coefficient that are well within the negatively sloped part of the performance curve. The onset of this type of instability is in some way connected to the internal compliance of the volume of vapor formed by cavitation.[2] The oscillations may be driven by a number of phenomena that take place within a pump such as blade passing frequency, white noise caused by cavitation, and so on.

C. Diffuser Stall

This type of pump instability manifests itself as a local stall within the pump diffuser. In Ref. 6 the pump with a crossover diffuser experienced instabilities at reduced flow coefficients that appeared to result from a breakdown of pressure recovery in the diffuser. At the point where diffuser stall was encountered, the time averaged stage head dropped by about 8 percent, causing a discontinuity and a hysteresis zone in the head - flow map (Fig 4(a)). Suction performance tests at 87 percent of design flow coefficient show the abrupt stall in the diffuser at reduced NPSH (Fig. 4(b)). With increased NPSH, diffuser stall was moved to below 80 percent of design flow coefficient. However, at reduced flow coefficients, significant discharge pressure oscillations occurred that were over 15 percent peak-peak of the stage head rise. Damage to the pump became a concern at low flow coefficients. A qualitative comparison of the location of the onset of vaned-diffuser stall and its dependence on NPSH is illustrated in Fig. 2 by C and C′ (reduced NPSH).

At flow coefficients between 92 and 108 percent of design, the pump operated normally without pressure oscillations or inducer instabilities. At flow coefficients above 108 percent of design, pressures were also stable but a gradually rising dependence on NPSH was observed, as is typical of cavitation at high flows.

Although dynamic flow reversals could not be verified in this pump during diffuser stall, the magnitude of the pressure oscillations was sufficiently high to prevent prolonged operation there. In a rocket engine system, controls would prohibit pump operation to the left of the first severe instability that can be encountered, which in this case appears to be diffuser stall. In addition to potential damage to the pump, operation of the pump to the left of the discontinuity on the characteristic map could cause control difficulties because of the hysteresis.

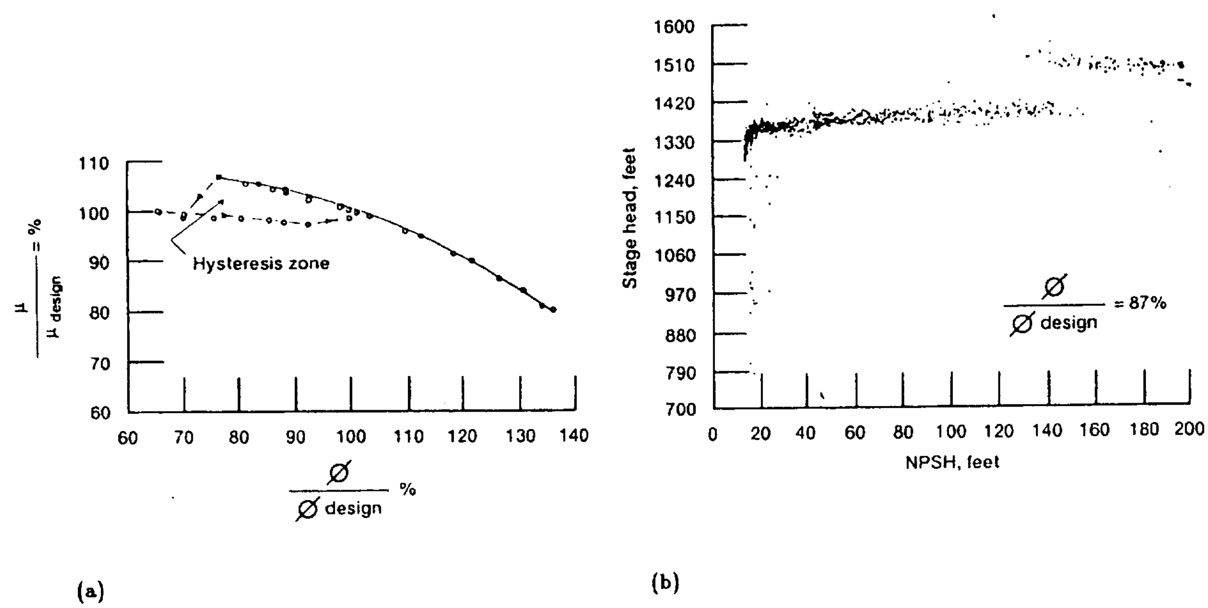


Figure 4. (a) Stall in the diffuser created a discontinuity in the head coefficient. The onset of diffuser stall in Ref. 6 depends on flow coefficient and NPSH. (b) Suction performance at 87 percent of design ϕ.

D. Rotor Stall

As the flow coefficient is reduced from the design value the rotor appears to stall. Flow reversal zones begin to form upstream and downstream of the rotor. The zone of flow reversal, or backflow, located ahead of the rotor typically forms near the outer wall.[4,7,8] Figure 5 from Ref. 8 illustrates the flow reversal ahead of the rotor near the outer streamlines. For comparison to the other forms of instabilities, point D in Fig. 2 illustrates the typical relative occurrence of the onset of rotor stall on a pump map. The relative location of C to D in Fig. 2 cannot be generalized with accuracy, since they are both dependent on the particular rotor-diffuser match that the pump designer selects. However, in compressors, the typical stage is matched so that rotor stall occurs to the left of vaned-diffuser stall (D to the left of C).

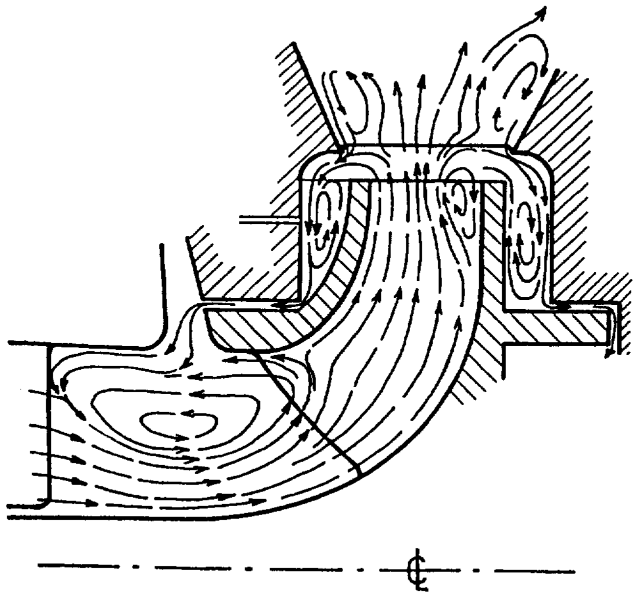


Figure 5. Secondary flow patterns in and around a pump impeller at reduced flow coefficient operation.[8] The occurrence of reverse flow ahead of the rotor is an indication of rotor stall.

A large component of tangential swirl velocity in the reverse flow creates a vortex that propagates upstream of the rotor. At intermediate flows below D, the static pressure at the core of the vortex drops below the vapor pressure, causing a small pocket of vapor to appear ahead of the rotor.[4] The pump can operate with a small degree of flow reversal ahead of the rotor in a seemingly steady two-phase manner with little impact on overall performance. In compressors it has been observed that, once reverse flow occurs, the fraction of reverse flow increases almost linearly with reduction in flow coefficient.[9] The magnitude of this flow "regurgitation" has been correlated with the flow coefficient. Surge-free performance was measured in a compressor[9] at flow coefficients near 50 percent of design where the reverse flow was 15 percent of the through flow.

E. Cavitation Surge

The appearance of flow reversal and a vapor pocket ahead of the rotor is a precursor to a much more severe condition called cavitating surge.[4,10] As the flow coefficient is further decreased, ultimately the pocket of vapor ahead of the rotor increases in size until it becomes partially injested and causes the complete breakdown of energy transfer, thus triggering surge. It may be assumed that the rotor can tolerate only a certain fraction of reverse flow before the onset of surge. Cavitation surge manifests itself as an unsteady phenomenon with severe flow reversals and pressure fluctuations at the pump inlet and discharge. High head pumps are more inclined to damage due to cavitation surge because of the elevated pressure levels. The location of pump surge on the normalized pump map of Fig. 2 is point E. This potentially damaging type of instability appears to be analogous to compressor-stage surge because of its dynamic nature.

In high-head rocket-engine pumps, the slight compressibility of liquid hydrogen is an additional factor that may support the dynamics of surge. The compressibility of liquid hydrogen alone may be sufficient to provide the compliance necessary for the dynamic behavior of pump surge. Depending on the degree of cavitation required to support surge in liquid hydrogen, the flow coefficient where surge is initiated may also be a function of the NPSH. Operating a liquid-hydrogen pump at a reduced value of NPSH may increase the degree of cavitation and, therefore, cause surge to occur at a higher flow coefficient (E′ of Fig. 2). From a rocket-engine standpoint, this is in the wrong direction and would decrease throttling capability.

The effect that NPSH has at low flow coefficients may appear contradictory to its effect on cavitation at high flow coefficients. But the flow physics at high flow appears to be quite different from that at low flow. At low flow, cavitation occurs ahead of the rotor and appears to be caused by the high component of absolute tangential velocity of the flow reversal. Increasing the flow at this condition will decrease the incidence, flow reversal and its tangential component and increase the local static pressure. At high flow, cavitation appears at the rotor throat and is caused by the excessive relative velocity there. Decreasing the flow at this condition will decrease the relative velocity at the throat and therefore increase the throat static pressure. However, for accurate quantitative descriptions of these phenomena, much work needs to be done in the area of two-phase code development.

Cavitation surge is a source of severe vibrations that can quickly result in damage and, as a consequence, a decrease in engine life. Because of rocket-engine safety and life requirements, system operation near surge is restricted by the controls, limiting the available engine throttling range in the process.

DESIGN CONCEPTS TO DELAY LOCAL STALL AND STAGE SURGE

The various types of instabilities discussed in the previous sections are dominated by cavitation-induced phenomena. If this were the only cause of instabilities, then one way to increase the range of stable flow would be to increase the supply NPSH, and "force" the pump not to cavitate at any flow. But this is not the preferred method from the standpoint of space-vehicle operational efficiency, as it would require low-head boost pumps or increased tank pressures which would require presumably heavy propellant tanks.

The more operationally efficient method to avoid pump instabilities is to address the pump design itself. The hydraulic design of the pump rotor and diffuser must, to some extent, control its resistance to the instabilities induced by flow separation and cavitation. If the causes of some of the instabilities can be eliminated in the design process, then cavitating surge may occur at lower flow coefficients. In addition, high-flow cavitation may be made to occur at higher flow coefficients.

In the following sections, pump design concepts that may improve on the various instability margins are discussed. Section I concentrates on design parameters of the pump stage and correlations to stall criteria. Sections II and III summarize other techniques and add-on devices some of which have been very successful at delaying the onset of pump and compressor stage surge.

I. PUMP HYDRAULIC DESIGN

Much work remains to be done in the area of modeling the pump stage at a level where it can predict stage surge. In addition to blade-vane interaction effects, instabilities related to the dynamics of the system further complicate the modeling effort. Flow models with these capabilities may require extreme sophistication that is not available today, such as the ability to analyze unsteady two-phase flows. Therefore, the designer must currently rely on qualitative analysis and simplified empirical correlations. The open literature contains more documentation of quantitative stall criteria for compressors than it does for pumps. The stall correlations given in the following sections include simple performance parameters and surge data from compressor technology, which may be analogous to pumps to a great extent, with the exception of predicting cavitation. Therefore, until they are verified, the extent to which compressor stall criteria are applicable to pump stall and surge should be regarded as only qualitative.

Simple design criteria that can influence stall margin are typically used in the conceptual-design phase. The ongoing attempt by designers to establish simple correlations between several compressor flow parameters and stall have resulted in reasonable success. This was particularly important before the advent of three-dimensional Navier-Stokes compressor flow codes. The level of sophistication of the simple design criteria includes steady meanline, two- and quasi-three-dimensional flow parameters. These "lumped parameter" design criteria in most cases treat the components of the stage independently. For example, correlations exist for predicting rotor stall independent of the correlations that exist for diffuser stall. In this way, interaction effects are lost but, in some cases, have been shown not to play an important role in stage surge anyway.

IMPELLER BLADE LOADING

A. One-Dimensional Loading Parameter

The most fundamental loading parameter is the head coefficient of the pump stage (μ). This is one of the first parameters that the designer specifies in the conceptual design phase for sizing the pump. The maximum values of head coefficient per stage are determined empirically, and are only vaguely correlated with the stall-free flow range that is attainable.[11] Values for μ at the design point may vary from 0.15 to 0.80, depending on specific speed (pump configuration). This head coefficient refers to the impeller only (it does not account for losses through the diffusion system). In a pump stage with an inducer-impeller combination, the contribution of the inducer to the overall head rise may be neglected at an initial level of analysis. Impeller work in terms of Euler's equation is:

$$H = \frac{R_2\, C_{u2} - R_1\, C_{u1}}{G_c} \tag{1}$$

Impeller head coefficient is:

$$\mu = \frac{H\,G_c}{U_2^2} \tag{3}$$

The range of stall free flow for the pump is inversely proportional to the head coefficient. In a high head multistage pump, this loading parameter is often a trade off between the maximum range of stall free flow, and the number of pump stages.

B. Two-Dimensional Loading Parameters

1. Relative Velocity Ratio

Correlation between the level of rotor-blade loading at the design flow condition and the range of potential stall-free flow can be quantified with two-dimensional analysis. The simplest two-dimensional loading parameter, commonly used during conceptual design, is the ratio of inlet-to-exit impeller relative velocities W_{1rms}/W_2. Both relative velocities are functions of the impeller inlet and exit annulus areas and the head level:

$$W_{1rms} = \sqrt{C_{1rms}^2 + U_{1rms}^2} \tag{3}$$

$$W_2 = \sqrt{(U_2 - C_{u2})^2 + C_{m2}^2} \tag{4}$$

Values of the impeller relative-velocity ratio at impeller stall vary from 1.52 to 1.7 but an average value of 1.6 is suggested.[11-13] A reduced value of this ratio at the design point will result in a higher range of stall-free impeller flow before the critical value of 1.6 at stall is reached.

To minimize the impeller relative-velocity ratio at the design point, either W_{1rms} must be decreased, or W_2 increased. W_{1rms} can be minimized by optimizing the inducer inlet-velocity triangles at the hub, rms, and shroud.[14] For a given minimum flowpath hub radius (R_{1h}), there is a flowpath shroud radius (R_{1s}) that results in a minimum relative-fluid velocity at the shroud (W_{1s}) and also at the rms radius (W_{1rms}).

$$W_{1s} = \sqrt{C_{1s}^2 + U_{1s}^2} \tag{5}$$

$$C_{1s} = \frac{M\,R_g\,T_1}{\pi\,A\,P_1} \tag{6}$$

$$A = \pi\sqrt{R_{1s}^2 - R_{1h}^2} \tag{7}$$

$$U_{1s} = \frac{\pi\,R_{1s}\,RPM}{360} \tag{8}$$

In addition to benefiting stall margin, a minimized inlet shroud relative velocity (W_{1s}) also maximizes the inlet static pressure at this critical area of the pump, where cavitation is usually initiated. The maximized inlet static pressure will suppress the onset of cavitation and improve the pump's Suction Specific Speed capability.

High leading-edge sweep in compressors has been effective at minimizing the shock losses encountered at tip relative Mach numbers above 1.5. Wide-chord axial compressors that produce pressure ratios in excess of 2.0 per stage have performed with an acceptable surge-free flow range, in part because of the high degree of leading-edge sweep.[15] If transonic relative Mach number in compressors can be considered analogous to the choking of inducers due to high flow cavitation, then pump inducers may benefit from similarly high degrees of leading-edge sweep and increased chord length. A highly swept leading edge may improve the suction performance of pump inducers.

2. Diffusion Factor

The diffusion factor (DF) quantifies the level of loading within pump blading. DF loading is another key parameter that is used to size pumps and determine the number of stages required to produce a given head. Diffusion factor can be determined from a two-dimensional level of analysis and described in terms of the local-velocity triangles at the blade leading and trailing edges. The parameters that effect the diffusion factor within pump blading are the inlet and exit tangential velocities, the ratio of inlet to exit relative velocities, and the blade solidity.

$$DF = 1 - \frac{W_2}{W_{1rms}} + \frac{R_2 C_{u2} - R_1 C_{u1}}{(R_1 + R_2) W_2 S} \tag{9}$$

$$S = \frac{\pi R_2}{Z\,L} \tag{10}$$

A revised diffusion-factor correlation by Rodgers[11,12] takes impeller average-shroud curvature into account.

$$DF = 1 - \frac{W_2}{W_{1rms}} + \frac{\pi \mu R_2 U_2}{Z L W_{1rms}} + 0.1\frac{b}{r_s}\left(1 + \frac{W_2}{W_{1rms}}\right) \tag{11}$$

$$b = \frac{R_{1s} - R_{1h} + B_2}{2} \qquad \textit{and} \qquad r_s = R_2 - R_{1s} \tag{12}$$

The number of impeller blades (Z) and the ratio of overall inlet to exit relative velocity (W_2/W_{1rms}) are significant parts of both DF loading parameters. For a given level of impeller work, a higher number of impeller blades results in decreased DF loading. Data analysis of centrifugal compressor tests in Ref. 11 gives values for the diffusion factor at stall between 0.67 and 0.82. A conservative design would use the lower value as the limit for stall.

If flow modeling with streamline analysis is used, the values of DF may vary in the spanwise direction within the same blade row. Controlling the spanwise-work distribution evenly over the blade will result in minimal spanwise variation of DF and prevent "overloading" some regions. The difficulty with this and other lower-level stall criteria is that a reliable data-base must be established to encompass a wide range of configurations from axial inducers to centrifugal compressors. In addition, the application of compressor correlations to pumps must be verified.

C. Blade-to-Blade Loading Parameter

Loading parameters in the blade-to-blade plane also have been correlated with impeller stall. A simplified overall expression of blade-to-blade loading (BBL) was presented in Ref. 11. This loading parameter also appears as a separate term in the revised DF loading equation (11).

$$BBL(overall) = \frac{\pi \mu R_2 U_2}{Z L W_{1rms}} \tag{13}$$

For a compressor, typical maximum value for this loading is 0.25 to 0.35 at the stall point.

Another blade-to-blade loading parameter that evaluates the local value of loading through the entire meridional length of the blade was shown in Ref. 16. To quantify this loading, a flow code with a higher level of analysis (typically quasi-three-dimensional) is required. This loading is defined as the ratio of the difference in static pressures along the suction and pressure surfaces to the mean static pressure at a given streamline.

$$BBL(local) = \frac{P_{ss} - P_{ps}}{P_{mss}} \tag{14}$$

Maximum suggested value for this loading at the design point is 0.70 in Ref. 16 at any point along the blade. The maximum value of blade-to-blade loading is used as a design criteria to determine the number of impeller blades (and splitter blades) and the work distribution in the spanwise and streamwise directions. The limit at stall is not known, but minimizing this loading at the design point will decrease the blade-to-blade static-pressure gradient. This may lead to a reduction in the magnitude of the secondary flows within the blade and reduce the potential for separation.

Reference 9 has shown a correlation between the inducer incidence angle and inducer stall. Inducer stall was observed to occur at 13° incidence.

Other design parameters that influence stall include the ratio of the impeller exit blade-to-blade distance and the axial width (B_2). Compressor impellers with a ratio of nearly 1 have resulted in very wide characteristic maps.[17] The effect of decreasing the width (B_2) will, in itself, decrease the flow rate for compressor surge.[18]

The degree of blade-angle backsweep at the exit of the centrifugal impeller affects the exit-velocity triangle and, therefore, most of the above loading parameters. Impellers having more nearly tangential blade-angles typically have reduced levels of loading parameters and, therefore, are more resistant to stall.

DIFFUSER

Came, Herbert[16] and Elder, Gill[19] have shown that the performance of the semi-vaneless space in vaned diffusers plays a key role in determining the onset of compressor surge. A correlation was given between a simple geometric diffuser parameter, and the maximum achievable static pressure recovery at surge, in the semi-vaneless space of the vaned diffuser. The diffuser parameter is a function of the wetted-surface area of the semi-vaneless space divided by the vaned-diffuser throat area and the number of impeller blades. Qualitatively, the stage with a small diffuser throat area and an impeller having a high blade number will have improved stall margin. However, the maximum achievable pressure recovery coefficient in the semi-vaneless space at stall, even with an "optimized" vaned diffuser, is 0.45 and generally results in low stall margin.[19]

The pump in Ref. 6 (map in Fig. 4), which experienced severe pressure oscillations due partially to diffuser stall between 87 and 76 percent of the design flow, appears to confirm the correlations based on compressors. Therefore, this criteria for compressor surge appears to be applicable for predicting diffuser stall in pumps. In addition, pump diffuser stall may be a precursor to pump surge.

In general, eliminating the vanes in the diffuser will permit pumps to operate over a wider flow range.[18] Vaneless diffusers are not as efficient at recovering static pressure, but can perform stall-free over a larger range of flow coefficients compared to vaned (or channel) diffusers.

In compressors, vaneless diffusers can also experience local instabilities, as well as play a key role in stage surge. Correlations of the flow angle within the vaneless diffuser have been made to rotating stall[20,21] and stage surge.[9] Controlling the flow angle in the vaneless diffuser to a value above a critical value has been effective at preventing rotating stall in the operating range, as well as delaying the onset of surge. In compressors, rotating stall in the diffuser is a local unsteady phenomenon that may not initiate stage surge, but may be a precursor to it. It can reduce the ratio of time-averaged stage pressure and create a discontinuity in the map.

Vaneless diffuser rotating stall in compressors has been correlated with the diffuser length, which can be expressed by the ratio of inlet to exit radius. Qualitatively, the shorter the vaneless-diffuser length, the more resistant the diffuser is to instability. The vaneless-diffuser axial width is inversely proportional to the flow angle in the diffuser. Qualitatively, a narrower vaneless diffuser results in higher diffuser stability.

Flow reversals have been observed to take place in pump vaneless diffusers at low flow coefficients. But the existence of dynamic pressure pulsations in the circumferential direction, characteristic of compressors during vaneless-diffuser rotating stall, must be verified. Therefore, at this time, the analogy of the compressor correlations to pump vaneless-diffuser stall is unclear.

II. FLOW RECIRCULATION

Recirculating the flow around the pump stage has been used to delay the onset of stall and surge in some pumps.[10,22] It has been found that augmentation of the "prewhirl" upstream of the rotor by injecting high pressure fluid from the pump discharge in a tangential direction has a significant stabilizing effect.[6] Some disadvantages of this method are that excessive power may be lost, thereby reducing engine system efficiency. Another disadvantage of recirculating pump flow is the possibility of flashing the "heated" fluid that is injected upstream of the rotor. This can increase the level of cavitation that may already exist at the inlet. If the flashing can be avoided, recirculation may be successful in reducing the incidence-related separation experienced by the rotor. The designer must weigh the risk of possibly aggravating an already cavitating inlet at low flow coefficients for better system efficiency.

III. CASING TREATMENT

Various passive devices generally referred to as casing treatments have been successfully used in compressor and pump systems for extending the stall-free operating flow range. Limited or no flow-modeling correlations are available on the effects of some of these devices. The effects of most of these devices on pump stability and cavitation are currently limited to qualitative correlations based on the effects that they have had on the stability of some pumps and compressors.

Recessed Vanes

The onset of cavitation surge has been effectively delayed by means of recessed vanes (sometimes called backflow recirculators) within the pump casing, as illustrated in Fig. 6. This technique has been successfully demonstrated in pumps of various configurations[4] and axial compressors.[5] The backflow and vortices that appear upstream of the rotor at low flow coefficients are extracted in the radial direction into slots within the case and get reinjected into the main flow at an upstream location. The high swirl velocity component of the backflow causes it to radially eject out of the mainstream into the recessed vanes. The fluid then moves axially through a set of straightening vanes (axial diffusers, stators) and then is injected back into the mainstream. The recessed vanes effectively stabilize the swirling tangential component of the backflow. This type of device can significantly delay the onset of cavitation surge. This is a passive device that does not degrade pump performance under normal operation and appears quite promising for use in rocket-engine turbopumps.

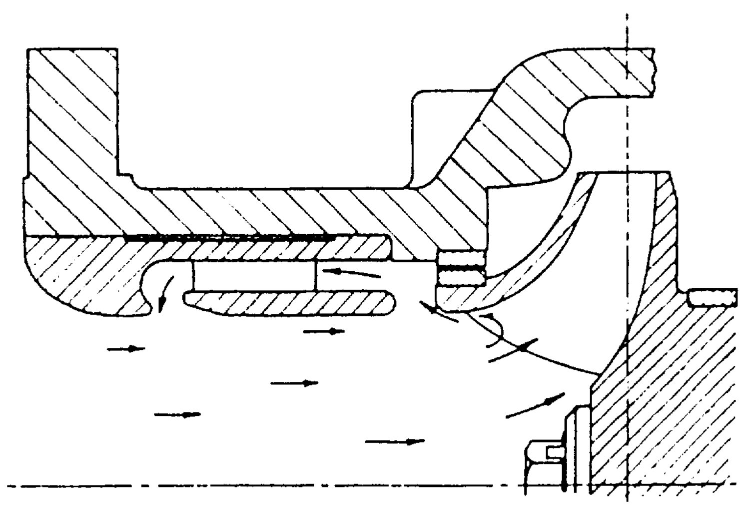


Figure 6. The swirling reverse flow encountered at low flow coefficients can be stabilized by means of deswirl vanes recessed within the casing[4].

Volute

The geometry of the volute effects the pump flow range. The absolute flow angle of the flow exiting the impeller is dependent on the impeller flow coefficient. At the design flow coefficient, the flow exiting the impeller typically experiences zero incidence with the leading edge formed by the volute tongue as shown in Fig. 7.

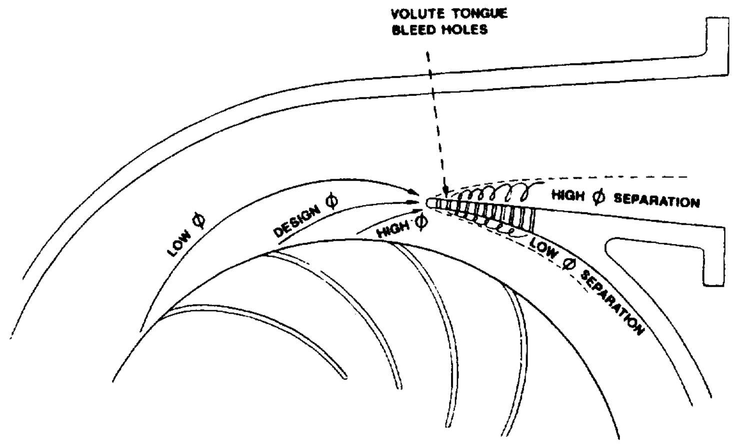


Figure 7. Severe incidences resulting from operation at off-design flow coefficients result in flow separation at the volute tongue (cutwater). Bleed holes located in the tongue may stabilize the boundary layer and delay separation.

At flow coefficients higher than the design value, the flow angle becomes more nearly radial, creating an incidence with the volute tongue. Likewise, at flow coefficients lower than the design value, the flow angle becomes nearly tangential and creates an incidence on the opposite side of the tongue. Increased levels of incidence due to pump operation on either side of the design flow coefficient ultimately results in flow separation and loss of performance. Laser measurements in Ref. 23 show the tongue stagnation point location moves from the suction surface to the pressure surface as the flow rate is decreased from 112 to 40 percent of design. Experimental investigation in Refs. 24 and 25 have shown the signifi-cance of the volute geometry on the shape and slope of the resulting pump characteristic head - flow map. The rate of diffusion in the scroll and the area at the volute throat also affect the head coefficient, efficiency and shape of the map.

One concept that may delay the onset of the flow separation from the tongue is to place bleed holes in the vicinity directly behind the tongue. Figure 7 shows the proposed location of the bleed holes on a typical volute tongue. This will provide communication between the suction and pressure surfaces at off-design operating conditions. The bleed holes may effectively control the boundary layer and delay flow separation from incidences encountered at off-design operation. This concept has yet to be verified experimentally.

The radial gap between the impeller exit and the volute casing also has been shown to affect the stable operating flow range of the pump.[26,27]

Plenums

Control of surge in compressors has been demonstrated by the use of plenums located downstream of the rotor. As illustrated in Fig. 8, a plenum recessed in the compressor casing near the trailing edge of the impeller was successful in extending the surge margin.[28]

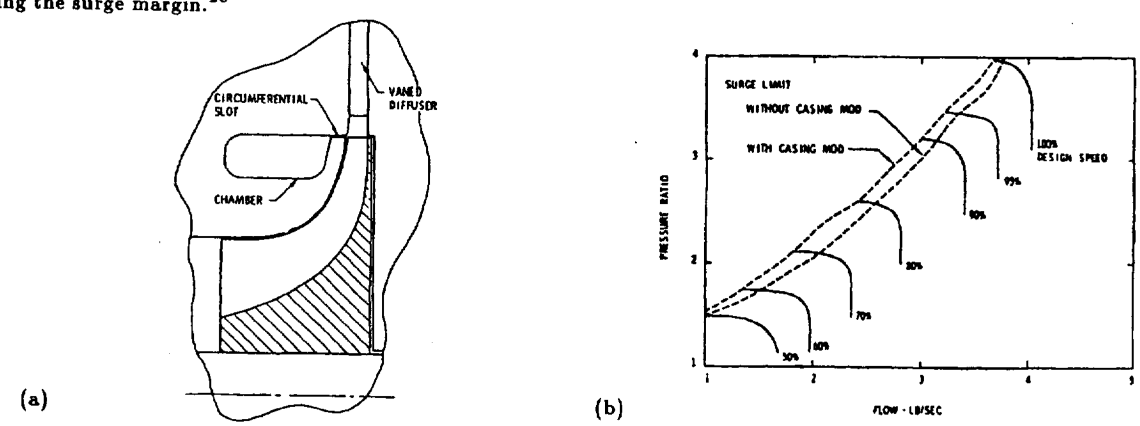
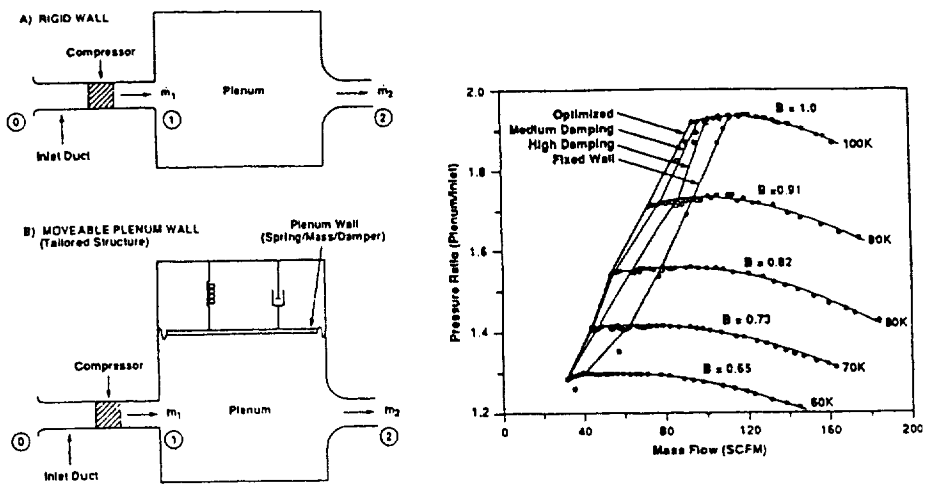


Figure 8. Plenum, or chamber in the casing outside the main flow stream (a). Effect of casing modification on compressor surge flow limit (b).[28]

Plenums and tailored structures located within the flowpath as illustrated in Fig. 9 have also been effective at delaying the onset of compressor-stage surge.[29]



(a) (b)

Figure 9. Plenum within the main flow stream downstream of the compressor (a). Increase in the stable flow region was achieved by tailoring the damping value of the wall (b).[29]

This approach modifies the dynamic behavior of the compressor system using structural feedback. More specifically, one wall of a downstream volume, or plenum, is constructed so as to move in response to small pertubations in pressure. The effectiveness of these techniques in extending the surge margin of pumps needs to be experimentally verified.

CONCLUDING REMARKS

To achieve the engine deep-throttling goals for future space vehicles, the design of the turbopumps will have to permit a wide surge-free flow-range capability. A turbopump with improved flow-range capabilities must be resistant to instabilities related to local separations and cavitation.

Current stability prediction methods consist largely of empirical correlations of lumped parameters to individual sub-components of the stage. The assumption, therefore, is that throttling range can be increased by biasing, or trading off, as much as feasible the loading levels based on the lumped parameters. Unloading the pump's rotor and diffusion system may result in improved resistance to local separations. Computer flow modeling of the pump sub-components at all operating conditions is an essential part of the design process to ensure that loading level limits are not being violated. The two- and quasi-three-dimensional loading limits should be analyzed with full three-dimensional computational fluid dynamics (CFD) codes. The currently available CFD codes are single-phase and, therefore, can only model noncavitating flow. Until three-dimensional two-phase flow codes are available for pumps, the compressor surge criteria may serve as an adequate guide to the designer. However, the compressor correlations to loading limits at surge must be verified for pumps. In addition, the influence of cavitation on pump surge must be better understood.

The sensitivity to instabilities can be reduced if key geometric features are incorporated into the pump early in the conceptual-design phase. These include inducer leading-edge backsweep, a vaneless diffuser followed by a volute, and an assortment of casing treatment devices. Vanes recessed within the pump casing ahead of the inducer appear to be effective at delaying cavitation surge.

Cavitation surge can be most damaging in high-head pumps. A low-head boost-pump can increase the NPSH of the high-head pump and thereby delay cavitation surge. The net effect may be a pump map with a wider surge-free flow range on the left side of the design flow coefficient. The increased NPSH to the high-head pump may also result in improved cavitation-free flow range to the right of the design flow coefficient. By gaining flow range the size of the pump can be optimized to result in improved engine-throttling range. However, this approach should be a last resort as it is at the expense of operational efficiency from the additional vehicle weight.

These concepts should be applied to a high-head research pump to validate their effectiveness on improving the stable flow range. Since cavitation plays an important role in determining the stable flow range of pumps, suction performance testing must be a vital part of the test plan in order to determine suction-specific-speed capability. If the test fluids are other than the actual rocket propellants (e.g., water), then the thermodynamic suppression head (TSH) has to be accounted for to properly simulate suction performance.

NOMENCLATURE

A	rotor inlet annular area
BBL	blade-to-blade loading
B	centrifugal blade height
b	centrifugal average blade passage height
C	absolute velocity
DF	diffusion factor
G_c	gravitational constant
H	rotor head rise
L	blade length
M	mass flow
P	pressure, static
R	radius from centerline
RPM	rotational speed
R_g	universal gas constant
r	radius of curvature
S	blade solidity
T	Temperature
U	rotor peripheral tip speed
W	relative velocity
Z	rotor blade number
ϕ	flow coefficient
μ	head coefficient
σ	cavitation number

Subscripts:

h	rotor hub; inducer, impeller
mss	mean stream surfaces
ps	pressure surface
rms	root-mean squared radius
s	rotor shroud; inducer, impeller
ss	suction surface
u	tangential component of velocity
1	inducer, impeller blade leading edge
2	inducer, impeller blade trailing edge

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13. ABSTRACT ***(Maximum 200 words)***

Design features and concepts that have primary influence on the stable operating flow range of propellant-feed centrifugal turbopumps in a rocket engine are discussed. One of the throttling limitations of a pump-fed rocket engine is the stable operating range of the pump. Several varieties of pump hydraulic instabilities are mentioned. Some pump design criteria are summarized and a qualitative correlation of key parameters to pump stall and surge are referenced. Some of the design criteria were taken from the literature on high pressure ratio centrifugal compressors. Therefore, these have yet to be validated for extending the stable operating flow range of high-head pumps. Casing treatment devices, dynamic fluid-damping plenums, backflow-stabilizing vanes and flow-reinjection techniques are summarized. A planned program has been undertaken at NASA Lewis Research Center to validate these concepts. Technologies developed by this program will be available for the design of turbopumps for advanced space rocket engines for use by NASA in future space missions where throttling is essential.

14. SUBJECT TERMS
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