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D. E. Jaekle, Jr.
PMD Technology
Lafayette, CA

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PROPELLANT MANAGEMENT DEVICE CONCEPTUAL DESIGN AND ANALYSIS: VANES

D. E. Jaekle, Jr.

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ABSTRACT

While surface tension devices have been used in liquid propellant tanks for almost thirty years, the conceptual design process and the analytical methods used to verify performance have been closely held by propellant management device (PMD) designers. With the proliferation of micro computers, the sophistication of the analytical techniques has greatly advanced. These advances have largely gone unpublished. To partially rectify this situation, this paper will address the process and the techniques developed and used by PMD Technology to design and verify the simplest PMD component, the vane.

All areas of concern inherent in vane design and implementation will be addressed - starting from the dictating requirements, proceeding into the design configuration choice, and ending with required performance analysis. The result is a cohesive process by which one may design and verify vane PMD components.

I. INTRODUCTION

Surface tension forces are negligible in most engineering problems. However, in the low gravity environment of orbiting vehicles, surface tension forces are significant and often dictate the location and orientation of liquid within vessels, conduits, etc. By carefully designing structures within a propellant tank, one can utilize these forces to ensure gas free propellant delivery. These structures have come to be known as propellant management devices or PMDs.

Traditionally PMDs are designed for each specific mission scenario and tank size. As a result PMDs can be found in numerous sizes and configurations. PMDs can be classified into three broad categories: partial control devices, total control devices, and total communication devices.[1] By definition, communication PMDs provide gas free propellant delivery by establishing a communication path between the bulk of the propellant and the outlet or another device component such as a sponge. The vane type PMD is such a device.

For this paper, a vane device is defined as a structure in proximity to a tank wall which creates an open path along which propellant can flow (the open path restriction excludes galleries and liners which create closed paths with screen or perforated sheet). Because all conventional propellants wet, propellant tends to cling to crevices and form fillets in the space surrounding the vane/tank wall intersection. Figure 1 illustrates the flow along a simple vane situated along an incline. A simple vane is defined as a thin solid sheet perpendicular to and traversing the boundary surface.

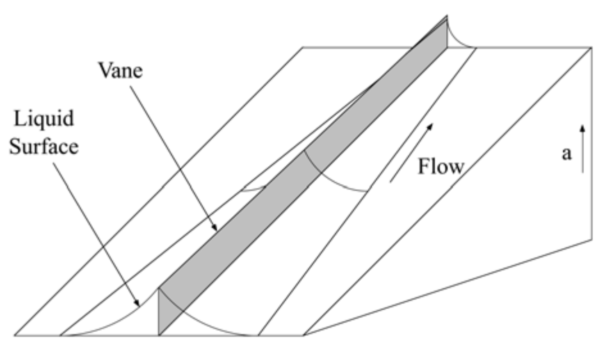

Figure 1. A Simple Vane on an Incline with Liquid Attached

The vane design process starts with the evaluation of the mission requirements to determine whether vanes are suitable. Once suitability is established, the design configuration and the design details must be explored. Finally with the design established, a thorough analytical investigation is conducted to verify performance. This last step is particularly important since vanes are not ground testable and performance verification relies completely on analysis.

This paper progresses along the same track as the design process. Section II addresses the physics of vane flow and presents the governing equations. Section III describes the uses of vanes and establishes the requirements leading to vanes. Section IV presents the major design choices and discusses the utility of each option. Finally, Section V presents the analytical techniques used by PMD Technology to verify the vane design.

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II. PHYSICS

The propellant illustrated in Figure 1 will flow up the incline against the hydrostatics only if the downstream radius is sufficiently smaller than the upstream radius. In the most basic terms, the surface driving pressure will be balanced by the dynamics, the viscous losses, and the hydrostatics. If the driving pressure is insufficient to overcome the hydrostatics, flow up the incline will not occur. A simplified set of equations follow.

The driving pressure within the liquid resulting from the surface tension forces is defined by the Laplace equation:[2]

$$\Delta P \equiv P_{gas} - P_{liquid} = \sigma \left(\frac{1}{R_1} + \frac{1}{R_2} \right) \qquad (1)$$

Thus the driving pressure within the liquid from the bottom to the top of the vane is approximately:

$$\Delta P = P_{up} - P_{down} = \left[P_{gas} - \sigma \left(\frac{1}{R_{up}} \right) \right] - \left[P_{gas} - \sigma \left(\frac{1}{R_{down}} \right) \right]$$

$$\Delta P_{drive} = \sigma \left(\frac{1}{R_{down}} - \frac{1}{R_{up}} \right) \qquad (2)$$

The radii R_{down} and R_{up} can be approximated as the fillet radii occupying the corner formed by the vane and the tank wall as illustrated in Figure 1 or more exactly as the mean Gaussian radii of curvature. For this presentation of the basic equations, the errors associated with one dimensionality are accepted.

Assuming steady flow up the incline and therefore given $R_{down} < R_{up}$, the surface tension driving pressure is opposed by the hydrostatic pressure, viscous losses, and the dynamic losses. Assuming fully developed laminar flow and mean flow velocities, these forces can be estimated as follows.[3]

$$\begin{aligned} \Delta P_{hydrostatic} &= \rho\, a\, \Delta z \\ \Delta P_{viscous} &= \frac{\tau_w \bar{s} L}{\bar{A}} = \frac{2 \mu Q \bar{s}^2 L}{\bar{A}^3} \qquad (3) \\ \Delta P_{dynamic} &= \rho \frac{\Delta (u^2)}{2} = \rho \frac{\Delta (Q^2 / A^2)}{2} \end{aligned}$$

As a rough approximation, one could implement Bernoulli's equation by equating the driving pressure to the sum of the hydrostatic, viscous, and dynamic pressures. The resulting equation could be solved for one of the three independent variables, R_{down} , R_{up}, or Q:

$$\sigma \left(\frac{1}{R_{down}} - \frac{1}{R_{up}} \right) = \rho\, a \left(z_{down} - z_{up} \right) + \frac{2 \mu Q \bar{s}^2 L}{\bar{A}^3} + \frac{\rho Q^2}{2} \left(\frac{1}{A_{down}^{\ 2}} - \frac{1}{A_{up}^{\ 2}} \right) \qquad (4)$$

Using equation (4), one could derive a curve relating upstream and downstream radii at a given flow rate. Each point on this curve would represent a different liquid volume attached to the vane. Given a downstream radius (the fillet radius at the outlet) and the steady flow rate demanded from the vane, one could approximate the volume of liquid which would have to adhere to the vane to meet the flow demand.

While the physics of flow up an incline can be explained by a relatively simply equation, the approximations resulting from this force balance are only rough order of magnitude for a variety of reasons including: a) order of magnitude viscous losses, b) neglected unsteady effects, c) superposition, etc. This basic force balance technique is a good tool for rough order of magnitude estimates and for feasibility studies but because of its inherent errors and problems it should not be used to validate a vane design.

The following aside presents an example of how this rough order of magnitude approach could be applied to a vane located within a propellant tank.

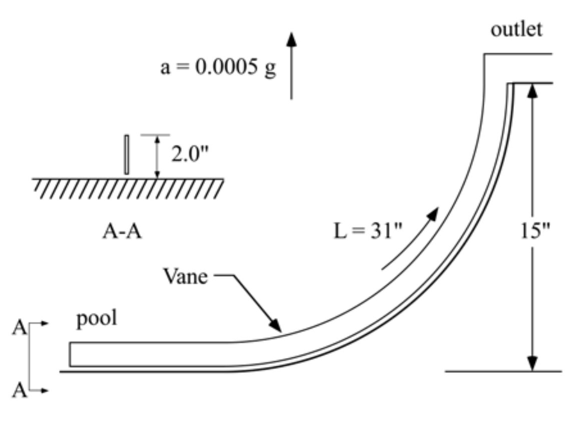
Aside - Section II

Assume:

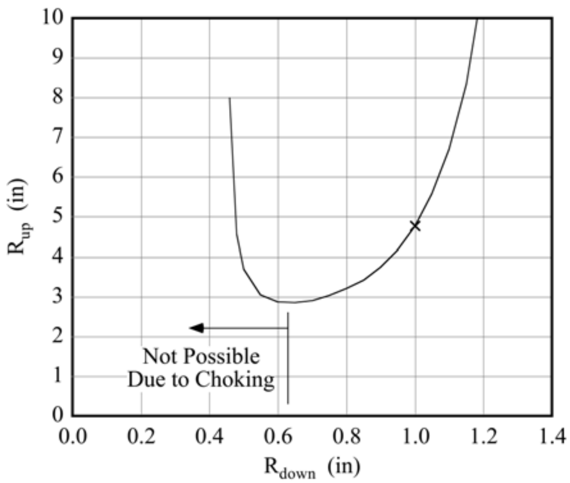
a) the propellant is hydrazine,
b) the acceleration is 5 x 10^{-4} g,
c) the vane configuration is illustrated in Example Figure 1,
d) the downstream radius is 1.0 inches,
e) the upstream radius is 4.76 inches, and
f) the viscous losses are based on the mean radius of 2.88 inches (A = 2.93 in^2, s = 9.5 in).

The rate at which propellant would flow up the vane under these conditions is 0.30 in^3/sec. Thus the vane illustrated in Example Figure 1 should be able to supply at least 0.30 in^3/sec and still maintain a radius of curvature of at least 1.0 inches at the downstream location (the top of the vane).

Example Figure 2 shows the relationship between upstream and downstream radii of curvature at a given a flow rate of 0.30 in^3/sec.



Example Figure 1. Flow Up a Simple Vane



Example Figure 2. R_{up} vs. R_{down} at $Q = 0.30\ in^3/s$

Example Figure 2 hints at the problems associated with a simple force balance. For each upstream radius there appears to be two downstream radii which could yield the required flow rate. The force balance method of analysis described distorts the true character of the solution (integrating the differential equations found in the section V shows that the left hand side of the curve in Example Figure 2 is generally not possible due to the choking which occurs between the upstream and downstream boundary conditions).

III. USES OF VANES

The principal advantages of vanes over diaphragms or positive expulsion devices are weight, reliability (no moving parts), and compatibility (100% Titanium designs are possible). However, diaphragms can deliver gas free propellant in any attitude and at almost any flow rate & acceleration. While vanes can deliver propellant in any attitude, they are limited by acceleration and flow rate. These limits have restricted their use.

Traditionally, the two principal uses of vanes are in hydrazine flexible demand systems and in bipropellant systems incorporating a sponge which must be refilled in zero g. Because vanes are limited by acceleration and flow rate, they are not used in bipropellant flexible demand systems. However, as vehicle mass grows, thrust created accelerations will decrease and vanes may become viable.

This section will address these two vane uses and describe how viability is determined for each system. Before embarking upon the design of a vane device, the requirements should be evaluated to determine if a vane is viable and any subsequent design effort is justified.

Hydrazine Flexible Demand Systems

Flexible demand systems require gas free delivery during thrusting in any direction and for any duration. This required flexibility forces the PMD designer to look at total communication devices - ones that can bring propellant from any region of the tank to the outlet. These include vanes, galleries, and liners. The vane PMD is by far the lightest, the simplest, the least costly, and the most reliable of PMDs. Unfortunately, vane PMDs are unable to provide propellant at flow rates and accelerations moderately high and thus viability is more limited. Figure 2 illustrates a vane device fitted into a typical propellant tank.

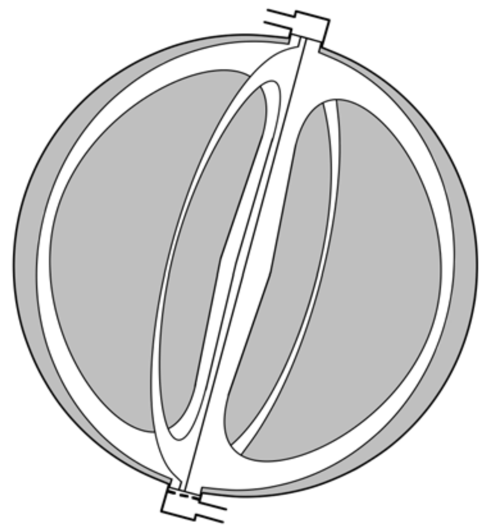


Figure 2. Vane Concept for a Flexible Demand System

Traditionally, hydrazine flexible demand systems incorporate small thrusters demanding a maximum combined flow rate on the order of 0.025 lbm/sec and producing accelerations in the range of $1x10^{-4}$ to $7x10^{-4}$ g. Vanes have often been able to meet hydrazine demand under these conditions and as a result have been employed on numerous vehicles.

The flow rate and acceleration limits of vanes in hydrazine flexible demand systems are related to one another and dependent upon three factors:

a) the tank geometry,
b) the type of vane employed, and
c) the allowable residual quantities.

Typically, the PMD designer is given these properties as well as acceleration and flow rate as mission or system requirements and then must determine if vanes are viable.

Vanes can immediately be rejected if the acceleration is too high and the surface tension forces are unable to lift the propellant from the pool to the outlet. The acceleration is too high if the hydrostatics exceed the surface tension forces given a sufficient safety factor. Thus,

$$\frac{1}{SF}\,\sigma\left(\frac{1}{R_{down}}-\frac{1}{R_{up}}\right) < \rho\, a\left(z_{down}-z_{up}\right) \qquad (5)$$

The minimum tolerable downstream radii of curvature, R_{down}, is determined by the outlet configuration. Usually, a porous element is positioned over the outlet as a gas arrestor. Hydrazine must pass through the element to exit the tank and gas is prevented from exiting by the surface tension forces in the pores. If the flow area is reduced by exposing the porous element to gas, the flow losses across the element will necessarily increase. Eventually the flow losses will overcome the surface tension forces and gas will be ingested. The bubble point of a porous element defines this pressure differential.

The flow area at which the losses are equal to the bubble point divided by a safety factor (three is commonly used) is taken as the minimum area required. The fillet radius of curvature necessary to obtain this flow area is the minimum R_{down} required and should be used in equation (5). In most PMDs, this radius lies between 0.1 and 0.5 inches (although it may differ and should be determined for each design). For conservatism, a radius of 0.5 inches may be used if the design details are unknown.

Once the minimum required radius of curvature at the outlet is established, the absolute limiting acceleration can be determined as a function of upstream radius. This limiting acceleration depends upon the height which propellant must be raised to reach the outlet and the radius of curvature at the propellant pool. The limiting acceleration is:

$$a_{limit} = \frac{\frac{1}{SF}\frac{\sigma}{\rho}\left(\frac{1}{R_{down}} - \frac{1}{R_{up}}\right)}{\Delta z} \tag{6}$$

The fill fraction dictates the variables R_{up} and Δz, and thus the limiting acceleration is dependent upon fill fraction. In blow down systems, where the thrust acceleration varies with fill fraction, all fill fractions must be evaluated to determine the worst case. In regulated systems where the required acceleration is constant, the worst case is end of life (EOL) where Δz is at its peak and R_{up} should be minimized for minimum residuals. Vanes can be excluded from consideration if the limiting acceleration at any fill fraction exceeds the thruster resultant acceleration.

Once the impact of the hydrostatics is evaluated and vanes are not excluded, an evaluation of the impact of the dynamics must be conducted. For this analysis one can use the force balance equation presented earlier or the modelling described in the Section V.

Vanes have been used extensively in hydrazine flexible demand systems because the flow rate and accelerations limits are often tolerable. The application of vanes in flexible demand systems has not been extended to bipropellant systems since thrust levels are often 10 times higher and nitrogen tetroxide's (NTO's) kinematic surface tension is only 27% of hydrazine's. This simple analysis, when applied to most bipropellant systems, will show that vanes are not capable of lifting NTO to the outlet under the thrust accelerations produced. For this reason, galleries and liners are used - or more often, sponges and traps coupled with limits on the mission flexibility. Sponge and trap PMDs generally are more reliable, less expensive, and lighter than galleries.

Refillable Component Systems

While not viable in bipropellant flexible demand systems, vanes can be used in systems that do not require unlimited flexibility in maneuver duration and direction. A typical geosynchronous communication satellite uses the majority of its on orbit propellant in fixed quantity station keeping maneuvers. As a result, refillable components, such as sponges, are employed to meet this intermittent demand. Typically several hours or days of zero g coast separate station keeping maneuvers. During this coast, vanes are used to refill the sponge. This is the second predominate use of vane devices: refillable component systems. In this case, vanes can be designed for use with NTO, monomethyl hydrazine (MMH), and/or hydrazine. Due to its low surface tension, NTO is generally the worst case.

A vane system designed to refill a sponge after a lateral E-W or N-S station keeping maneuver requires four vanes aligned with the thrust vectors (misalignment is generally not a serious drawback in zero g refilling since, in zero g, the propellant will climb the walls and eventually reach a vane). The vanes may or may not have to extend to the top of the tank depending upon condensation and fluid position stability considerations. A vane concept designed to resupply a sponge is illustrated in Figure 3.

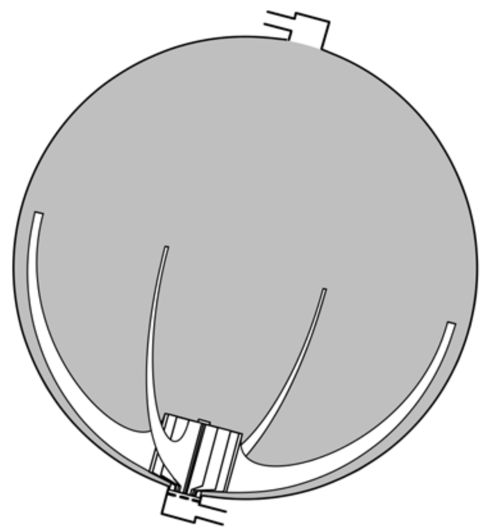


Figure 3. Vane Concept for a Refillable Sponge System

To obtain an order of magnitude estimate, one can use the force balance analysis, equation (4), to estimate a steady flow rate given upstream and downstream radii. The flow losses will not be conservative and a friction multiplier at least three is typically used as well as a safety factor on the surface tension (a safety factor of two is typically used). With safety factors, the resultant equation is:

$$\frac{1}{SF}\sigma\left(\frac{1}{R_{down}} - \frac{1}{R_{up}}\right) = F_{mult}\left(\frac{2\mu Q \bar{s}^2 L}{\bar{A}^3}\right) + \frac{\rho Q^2}{2}\left(\frac{1}{A_{down}^2} - \frac{1}{A_{up}^2}\right) \tag{7}$$

The downstream radius is determined by the sponge design or the gap between the sponge and the vane (a gap often exists to prevent the sponge from leaking during a burn). The remaining independent variable, the upstream radius, is defined by the fill fraction. With the upstream and downstream radii and the vane configuration defined, the steady flow rate and the time to refill a sponge can be estimated. The modelling in Section V provides more accurate refill times.

Another way to bound the refill time is to consider the time required for the liquid in a bare tank to reorient into its zero g configuration. Extensive drop tower tests of small liquid filled glass spheres have been conducted and analyzed.[4,5] From these data and analysis, one can estimate the time for liquid in a partially full tank to orient into a zero g configuration. This time should be conservative since vanes will aid reorientation. Unfortunately, the fill fractions adequately analyzed are not sufficiently low to be of value in modelling depletion - the slowest reorientation period and therefore the worst case. This method is only useful in estimating times to determine viability and not for performance verification.

The vast majority of vanes currently in use either provide flexible hydrazine demand or refill sponges in zero g. Other uses are possible but have yet to implemented.

<u>Other Uses for Vanes</u>

Vanes are used primarily in hydrazine flexible demand and refillable component systems. Vanes also have potential in other applications including flexible bipropellant demand, cryo-fluid tankage, material processing, and thermal applications.

Although the absolute limiting acceleration cannot be modified by vane design, the limiting flow rate can be increased with alternative vane cross sections as indicated in Section IV. This increased flow rate capability will allow vanes to be implemented in low surface tension systems which a) need to refill sponges in minutes instead of hours or days and b) are incorporated into very massive vehicles where thruster resultant accelerations are small but flow rates are high. Examples include space tugs and space based platforms. Vanes are expected to appear in many alternative uses in the future.

IV. VANE DESIGN

The simple vane illustrated earlier in Figure 1 is only one of numerous possible vane designs. This section will address qualitatively, and in some cases quantitatively, the various design issues. The design choices can be divided into four categories: vane position, vane cross section, center posts, and design details.

<u>Vane Position</u>

Vanes need not be placed, as previously illustrated, along the tank wall extending radially and linearly from the propellant outlet (although it is the predominate position in existing vane PMDs). A vane could be designed to spiral out from the outlet or connect two vanes in the mid-plane of a tank or in fact could be situated in any position in proximity to the tank wall. The two dictating factors are the path length to the outlet or sponge and the separation between vanes and the thrust vector.

The reason vanes typically extend from the outlet directly toward the opposite end of the tank is to minimize the path length to the outlet and thus minimize the flow losses and residuals. Proper positioning of vanes relative to the thrust vector is critical to ensure proper vane operation and minimal residuals in flexible demand systems.

One consideration in proper positioning of the vanes is whether they extend to the top of the tank. Generally, extending vanes to the top of the tank must occur if propellant access during or after non-settling accelerations is required. In addition, extension is also generally recommended in cylindrical section tanks where the propellant could occupy a stable zero g position in the upper tank head. In spherical tanks experiencing only lateral accelerations, vanes need not extend to the top of the tank, although additional flow rate capability or structural concerns may require such an extension (see following section on center posts for more on flow rate capability concerns).

Also important in vane positioning is the separation between the vane and the thrust acceleration. In flexible demand systems, it is likely that some thruster operations will produce acceleration vectors not in line with a vane but bisecting the region between two vanes. An acceleration vector bisecting two vanes will orient the propellant between the vanes. If the vanes are too far apart, a pool will form at the end of life and residual propellant quantities may become excessive. To avoid large residual propellant quantities, the number of vanes must be chosen to ensure that no pool or an acceptably small pool results at EOL. Figure 4 shows the liquid in a tank near EOL in a poor design and a good design.

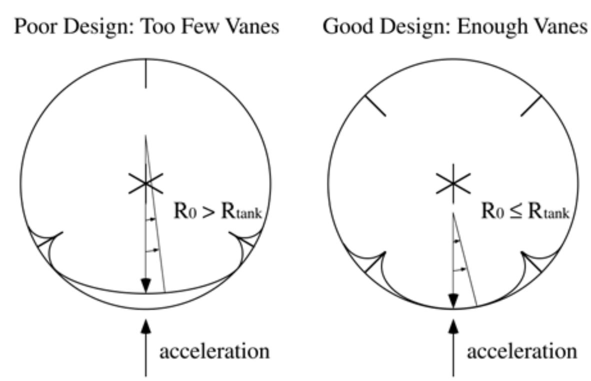


Figure 4. Insufficient and Sufficient Number of Vanes

The equilibrium fluid position in a tank during an acceleration is fully described by Laplace's equation for curvature coupled with hydrostatics or:

$$\frac{1}{SF}\,\frac{\sigma}{\rho}\left[\left(\frac{1}{R_1}+\frac{1}{R_2}\right)-\left(\frac{1}{R_{10}}+\frac{1}{R_{20}}\right)\right] = a\left(z - z_0\right) \quad (8)$$

While it is theoretically possible to determine the surface position in three dimensions, it is not required. It is far simpler and sufficient to compute the bounding surfaces by computing the axisymmetric surface and the planar or cylindrical surface at variety of potential energies. If an axisymmetric surface is assumed (as in a spherical tank), the surface predicted will be more curved than one would expect in the cylindrical section of a tank but would be close to the surface near the ends of tank. A more conservative approach is to assume the surface is two dimensional. This would result in less curved surfaces and correspondingly more conservative results. If a liquid pool will drain between two vanes assuming a planar surface, a conservative vane design will result.

A rigorous examination of axisymmetric and two dimensional surface generation and stability can be found in *Low-Gravity Fluid Mechanics* by Myshkis et al.[6] However, foregoing some of the rigor, the following integration scheme to determine the axisymmetric surface produces satisfactory results (see Figure 5 for a geometric representation of the proposed integration):

1) Assume a radii of curvature at the base of the surface (radii larger than the tank radius would allow a pool to form). The first and second radii of curvature are identical at this point on the axis of symmetry. Therefore, $R_{10} = R_{20} = R_0$.
2) Step $\Delta\theta$.
3) Calculate new point (r,z) based upon previous step's R_1 and a continuous first derivative.
4) Determine new R_2 which is equal to the distance from the surface to the axis of symmetry through the center of radius R_1 (perpendicular to the surface) as illustrated in Figure 5.
5) Using z and R_2, solve equation (8) for a new R_1.
6) Compute the new center of curvature maintaining continuity of the first derivative of the surface.
7) Repeat steps 2 through 7 until the surface is defined ($\theta = 180°$).
8) Fit surface to tank walls by moving the surface along the axis of symmetry until all boundary conditions are met (contact angle is zero).

For the planar case the integration is identical except R_2 is infinite everywhere and thus step 4 is omitted. This method is easily implemented in a spreadsheet on a micro computer.

The minimum tolerable vane separation and thus the number of vanes required is easily determined by fitting vanes into the tank so that they intersect the computed no-pool surfaces. In practice, vane placement is not exact (the vanes are often very thin and seldom straight) and a wide tolerance capability should be built into the design.

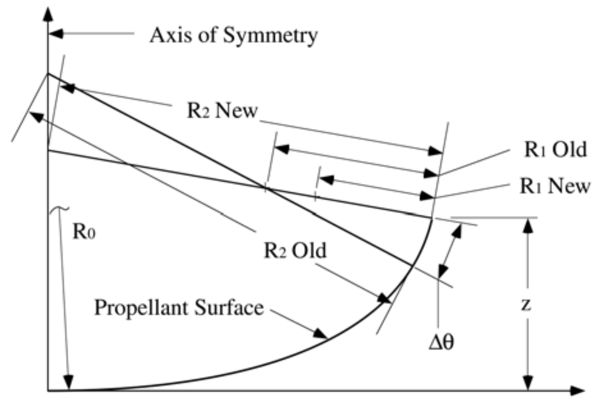


Figure 5. Schematic of Axisymmetric Surface Location Procedure

Vane Cross Section

Besides the vane position within the tank, the vane cross sectional shape and size is equally important. To date nearly all vanes in use are "simple vanes" consisting of a single sheet metal panel extending from and perpendicular to the tank wall. Alternative vane cross sections include ribbon vanes, parallel vanes, and complex combinations. Figure 6 illustrates some cross section possibilities. Fluid, shown in gray, adheres to vane structure to minimize surface area and maintain the contact angle.

The choice of a vane cross section is dictated by performance and manufacturing considerations. A simple vane is most often used because it is easy to build and provides adequate performance to meet the flow demands of the system. The ability of a vane to deliver propellant is directly proportional to the cross sectional area of attached fluid. Since all conventional propellants have a zero contact angle when in contact with clean metal, fluid tends to fill all crevices to minimize surface area and surface energy. Thus more complex vanes can supply a greater flow area at a given fillet radius and more capability.

In a curved wall tank with a finite height vane, the area as a function of radius can be determined but is complicated in

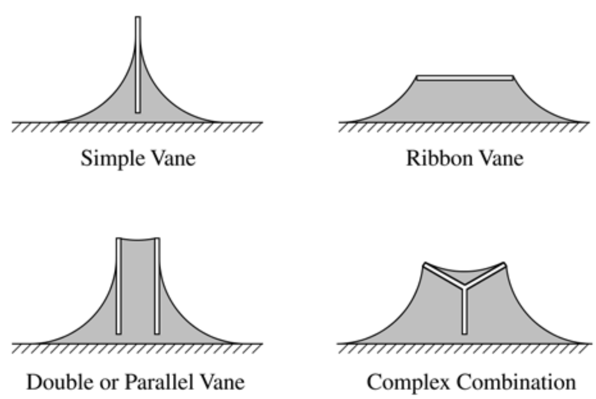


Figure 6. Possible Vane Configurations

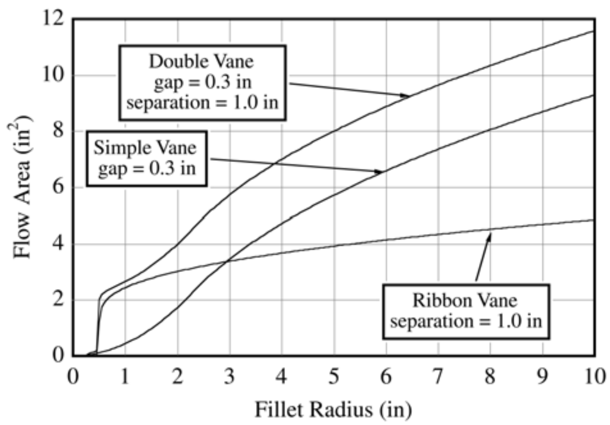


Figure 7. Cross Sectional Flow Area as a Function of Fillet Radius for a 2" High (Wide) Vane on a Planar Wall

explicit form. Figure 7 illustrates the area-radius relationships for a variety of example vane configurations. The areas shown assume a planar wall and a vane height (or width) of 2.0 inches. The gaps and separations used in the calculations are identified in Figure 7.

Figure 7 shows that the vane with the greatest flow area is the double vane and the ribbon vane is advantageous at the critical smaller radii. By varying the dimension of the vane, the area relationship can be altered to meet flow rate, and thus flow area requirements. For example, a large flow rate requirement leads one to use a ribbon or double vane. But in doing so one is burdened with manufacturing disadvantages associated with fabrication of more complicated assemblies.

A simple vane is by far the easiest to implement since there are no dimensions which must be held to close tolerances. With a ribbon vane, the distance to the tank wall must be maintained and with a double vane, the distance between vanes must be maintained. In addition, in the case of a ribbon vane mounted within a tank with curved walls, the vane must be curved or 'v' shaped to prevent gas bubbles from being trapped between the tank wall and the vane.

In summary, due to the added manufacturing difficulties of ribbon and double vanes, a simple vane should be implemented if it meets requirements and a double vane implemented in cases requiring more flow rate capability.

Center Post vs. No Center Post

Another design choice is whether to implement a tank center post. A center post is defined as a centrally located structure designed both as a structural support and as a functional flow path from the opposite end of the tank to the outlet. Figure 2 shows a PMD with a center post while Figure 3 shows one without a center post.

In those PMDs which implement vanes that follow the outer tank wall contour from the pressurant inlet to the propellant outlet, the vanes are generally supported at their ends and not along the wall. Without a center post, the vane must be attached to the pressurant hemisphere with a degree of freedom along the tank axis (allowing for tank growth) and must also have enough stiffness in this axis to ensure proximity to the tank wall at the top of the tank. A center post provides this stiffness without stiffening the individual vanes and thus eliminates vane structural integrity issues (stiff vanes must bear slosh loads while thin vanes simply bend out of the way of the sloshing fluid). Stiffer vanes provide slosh damping - an advantage in some cases.

Center posts are also desirable because they provide a more direct flow path from the pressurant inlet to the propellant outlet. This additional flow path increases the flow rate capability of the PMD.

For reasons previously stated, in many designs it is not necessary to run the vanes all the way to the top of the tank. In these cases, a center post is not advantageous nor required.

As stated in the previous section, increasing flow area increases flow rate capability. Figure 8 depicts several viable center post cross sections which allow flow along the center post. The choice of cross section is determined by manufacturing, structural support, and flow considerations.

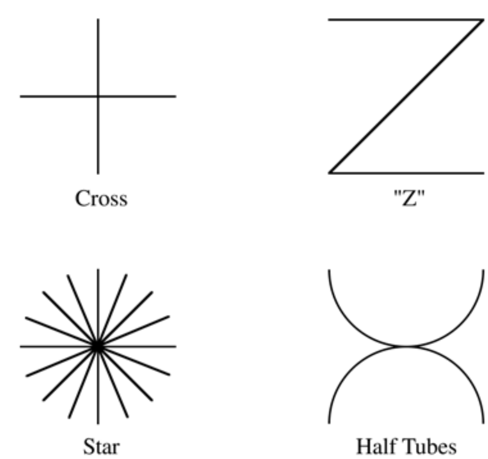


Figure 8. Possible Center Post Cross Sections

In general, a PMD using a center post can deliver propellant to the outlet at nearly twice the flow rate of a PMD without a center post. Typically, a center post is designed so that at any fillet radius, the flow area is many times larger than the flow area of a simple vane. Thus the flow losses along the center post are negligible compared to the flow losses along the vane. During a lateral acceleration, two flow paths to the outlet (or sponge) exist - one from the side of the tank directly up a vane to the outlet and the other from the side of the tank up a vane to the top of the tank and down the center post to the outlet. Since the flow losses are negligible down the center post, each flow path will carry roughly equal flow. Thus the flow rate capability of the PMD is twice that of a PMD without a center post.

The center post is also advantageous since it will contain propellant in zero g and can provide that propellant to the outlet upon demand. This ability makes the center post ideal to accommodate transient demand such as engine ignition and engine pulsing. In a flexible demand system, the center post and the vanes are surrounded by stationary propellant prior to a burn. At the onset of demand, the propellant must be accelerated from zero velocity to the demand velocity. The center post can provide propellant during this transient.

In general, a center post should be implemented if flow demand is greater than a single vane can deliver or if structural considerations demand a central support.

Details

Once the broader design issues are settled, design details such as vane mounting, structural support, and flow peculiarities must be addressed.

Vanes are typically mounted by welding to the tank outlet or to a sponge. Half height vanes are cantilevered and are allowed to bend in response to slosh loads. Full height and center posted vanes are supported at the top of the tank by a slide; allowing motion along the tank axis (for growth) but restraining the PMD laterally.

As mentioned previously, vane PMDs which feed the outlet directly implement a porous element between the vanes and the outlet to ensure gas free propellant. This porous element can be perforated sheet, screen, or perforated tube. The exact configuration is not important as long as adequate flow area is available at small fillet radii. In addition, it is critical to have a gap between any porous element hole and the vane structure greater than the porous element hole diameter. This will prevent the vane from pulling propellant from the porous element during high accelerations.

In terms of structural support, vanes can either be designed to withstand the loads or to bend upon application of the loads. Often it is easier to design a vane which bends elastically than it is to define the design loads. For example, if liquid is rotating within the tank, a vane must either be stiffened to sustain the drag loads or designed to roll up thus minimizing the loads. If no support exists, the vane may bend plastically and not return to its original position. If a support is used to limit the radius the vane bends around, the vane can sustain the loads and return to its original position.

Structural supports should be designed not to reduce the flow area along the vane; thus no support should be used in proximity to the "corner" formed between the vane and the tank wall or formed by the center post sections.

The last detail of importance is the cross flow at the ends of the vanes. In a vane PMD with a center post, the propellant flowing up the vane and down the center post must be able to reach all the fillets in the center post at the pressurant end. Thus a slot in the vanes is required at the top of the PMD. This slot should be large enough to allow cross flow with little if no pressure loss.

V. ANALYSIS

Besides simple bubble point tests verifying porous element integrity, no performance related quantitative testing in one g is possible. As a result, extensive analysis using large safety factors is required to verify performance.

PMD Technology has developed the techniques presented in this section to verify vane compliance with the operating requirements. Two types of analysis are presented: one for steady flow along a vane and one for unsteady flow.

Steady Flow Analysis

The flow in a channel with variable area, such as the fillet formed along a vane or along a center post is similar to flow in flexible tubes (such as arteries in mammals), flow in open channels (such as water conduits), and even compressible duct flows (such as high speed wind tunnels). Each of these flows is governed by similar equations.

The most important similarity is the non zero dA/dx terms in the continuity equation (dρ/dx in compressible flow). One similarity between compressible flow and other area-variable flows is the existence of limiting or choking velocities. In compressible flow, the speed at which density waves propagate defines the choking velocity (the speed of sound). In area-variable flows, the speed at which area waves propagate defines a similar limiting velocity. This velocity could limit the flow rate even though the driving forces are quite high.

Before deriving the flow equations and explaining the computational model, an example dealing with this choking velocity is presented as a possible upper limit on demand flow rate. The area wave speed is derived from propagation mechanics as:[7]

$$c^2 = \frac{A}{\rho}\frac{\partial\left(p - p_{gas}\right)}{\partial A} \tag{9}$$

But since, in a simple vane fillet,

$$p - p_e = -\sigma\left(\frac{1}{R}\right) \quad \text{and} \quad A \approx \left(\text{constant}\right)R^2 \tag{10}$$

the area wave propagation speed, c, down a simple vane or center post is approximately:

$$c = \sqrt{\frac{1}{2}\left(\frac{\sigma}{\rho}\right)\frac{1}{R}} \tag{11}$$

Table 1 shows the maximum flow rate along a simple vane as a function of fillet radius assuming that the flow is choked and the fluid is hydrazine. In a center posted vane design, the maximum flow rate deliverable to the outlet by the PMD is twice the vane maximum flow rate because the flow can travel along two paths - one up the vane directly to the outlet and the other up the vane to the pressurant end of the center post and down the center post to the outlet. The maximum

Table 1
Flow Rate Assuming Choked Flow Along Simple Vane

R (in)	c (in/sec)	Q (in^3/sec)
0.10	4.52	0.02
0.20	3.20	0.05
0.30	2.61	0.10
0.40	2.26	0.16
0.50	2.02	0.22
0.60	1.85	0.29
0.70	1.71	0.36
0.80	1.60	0.44
0.90	1.51	0.52
1.00	1.43	0.61

center post flow rate is large compared to the vane maximum flow rate and therefore it is not limiting.

The flow rates in Table 1 are not definitively limiting because the flow need not be steady and can be supercritical. These choking flow rates are only of passing interest. More significant is steady and unsteady flow modelling which follows.

The steady flow equations can be derived from the continuity and momentum equations:

continuity:

$$Q = A\,u \quad \text{or} \quad du = -\left(\frac{Q}{A^2}\right) dA \qquad (12)$$

momentum:

$$-A\,dp - \tau_w\, s\, dx - \rho\, A\, dz = \rho\, A\, u\, du \qquad (13)$$

equation of state (surface tension):

$$p - p_e = -\sigma\left(\frac{1}{R}\right) \quad \text{or} \quad dp = \sigma\left(\frac{1}{R^2}\right) dR \qquad (14)$$

Combining and reducing:

$$\frac{dA}{dx} = \left(\frac{A}{Q}\right)^2 \left[\left(\frac{\sigma}{\rho}\right)\frac{A}{R^2}\frac{dR}{dx} + \frac{\tau_w\, s}{\rho} + a\,\frac{dz}{dx} A\right] \qquad (15)$$

The friction term is roughly estimated assuming steady, fully developed laminar flow as:

$$\frac{\tau_w\, s}{\rho} \cong 2\,\nu\, Q\left(\frac{s}{A}\right)^2 \qquad (16)$$

Because the flow is not as simple as the above friction term would tend to indicate (due to varying flow area), the true losses can be conservatively estimated to be as high as twice the above approximation. When computational simulations are conducted, two friction estimates should be used: the one above and one twice as large.

The governing equation is reduced into the form:

$$\frac{dR}{dx} = \frac{\left(\frac{2\,\nu}{Q}\right) s^2 + \left(\frac{a}{Q^2}\frac{dz}{dx}\right) A^3}{\frac{dA}{dR} - \left(\frac{\sigma}{\rho}\right)\frac{1}{Q^2}\frac{A^3}{R^2}} \qquad (17)$$

If the denominator equals zero, the flow becomes choked as the fluid velocity reaches the area wave propagation speed. This equation is not strictly valid if the denominator equals zero. If the denominator goes to zero and the numerator is non zero, the flow is truly choked with no possible steady solutions. Unsteady behavior or shocks will appear. On the other hand, if the numerator goes to zero and the denominator is zero, a singular point is encountered. This singular point may or may not allow smooth transition from subcritical to supercritical flow and is dependent upon the nature of the singularity (saddle, nodal, spiral, etc.).

Equation 17 can be numerically integrated using any one of a number of techniques including the fourth order Runge-Kutta method. However, since the equation can describe non real solutions (such as spirals), integration by stepping along the integration path (and not in x) is more accurate and more telling since transition points can be identified. Classification of singular points is also possible through linearization near the singular point but is beyond the scope of this paper.[8] For conservatism, numerical transitions are not allowed and the vane is assumed to choke.

To integrate a specific vane configuration, several definitions are required: dz/dx as a function of x, the flow area & wetted circumference as a function x and R, and the flow path. The effect of a) the finite width of the vane, b) the curvature of the tank wall, and c) the components of a center post must be taken into account by the model. The three dimensional effects at the vane/center post intersection are not included. This is conservative because the defined cross sectional area is lower than what will occur in reality.

In most cases, more than one flow path exists from the pool in the tank to the tank outlet and the solution is iterative. First, assuming a flow rate along each path, the radii of curvature can be determined along each path. Second, if the radii at the end of each integration path are not identical, then the flow division is adjusted and the process is repeated until the radii are identical at the pool.

The integration can be accomplished at a variety of downstream fillet radii (at the outlet) with the demand flow rate. The result is the radii along the entire flow path including the upstream radius. Integrating the area along the flow path yields the volume and the fill fraction. Thus the characteristics of steady flow along a vane can be determined as a function of fill fraction in the tank. In addition, it is fairly straightforward to evaluate the impact of vane cross section and other details on vane performance. Thus coupling the design and analysis yields an optimal design.

During long thruster burns, a steady state solution is accurate after the initial thrust ignition transient but during the ignition transient (or during the depletion transient), a steady model is not adequate. An unsteady model is more appropriate and can help answer questions such as “Is a center post required to start the engines?” and “How much propellant residual volume is present when gas is first ingested?”.

Unsteady Flow Analysis

To attain the steady flow modelled in the preceding section, the propellant must be accelerated from its static equilibrium position in zero g to the steady flow condition. This engine ignition transient is an unsteady phenomenon of particular interest because if the vanes or center post do not respond quickly to meet demand, the fluid around the outlet will be consumed and gas ingested into the outlet. In addition to this unsteady phenomenon, thruster pulsing, depletion, and sponge refilling must be analyzed in terms of unsteady flow.

The unsteady model is similar to the steady flow model. The one dimensional assumption along the given flow path is employed.

The assumptions used in the steady model apply here. The differences are a) the equations contain unsteady terms and b) the solution is by finite differences and not straightforward integration. The unsteady flow equations can be derived from the continuity and momentum equations:

continuity:

$$\frac{\partial A}{\partial t} + \frac{\partial}{\partial x}\left(A u\right) = q_L \tag{18}$$

momentum:

$$\frac{\partial u}{\partial t} + u\frac{\partial u}{\partial x} + \frac{1}{\rho}\frac{\partial p}{\partial x} + a\frac{\partial z}{\partial x} + \frac{4\tau_w}{\rho D_e} + \frac{q_L\left(u - u_{Lx}\right)}{A} = 0 \tag{19}$$

equation of state (surface tension):

$$p - p_e = -\sigma\left(\frac{1}{R}\right) \quad \text{or} \quad \frac{\partial p}{\partial x} = -\sigma\frac{\partial}{\partial x}\left(\frac{1}{R}\right) \tag{20}$$

In addition, the friction term can be roughly estimated (as steady, fully developed, laminar flow in a constant area conduit) to be:

$$\frac{4\,\tau_w}{\rho\, D_e} \cong 2\,\nu\left(\frac{s}{A}\right)^2 u \tag{21}$$

Because the flow is not as simple as the above friction term would seem to indicate (due to varying flow area and boundary layer development), the losses were estimated for a few specific cases. It can be shown that friction can be as high as five times the above approximation. When the computational simulations are conducted, two friction estimates should be used: the one above and one five times larger. Although the worst case in steady flow is high friction, exactly the opposite is true during the transient. This is because the friction damps any oscillation created in the transient. Similarly, using half the surface tension effectively reduces the 'spring force' and may not produce a worst case. Several cases with and without safety factors should be examined to completely bound the problem.

Substitution and manipulation yields:

$$\frac{\partial u}{\partial t} = -\left[u\frac{\partial u}{\partial x} - \frac{\sigma}{\rho}\frac{\partial}{\partial x}\left(\frac{1}{R}\right) + a\frac{\partial z}{\partial x} + \frac{2\,\nu\, s^2}{A^2}u + \frac{q_L}{A}\left(u - u_{Lx}\right) = 0\right] \tag{22}$$

$$\frac{\partial R}{\partial t} = -\frac{1}{\left(\frac{\partial A}{\partial R}\right)}\left[\frac{\partial\left(A u\right)}{\partial x} - q_L\right] \tag{23}$$

These can be solved using a two step Lax-Wendroff type scheme: one of the more popular methods for solving compressible flow problems with friction. For example, the method of Rubin and Burnstein is used successfully.[9] Velocity and radius are calculated at half spaces and full time steps by averaging:

$$U_{i+½}^{n+1} = \frac{1}{2}\left(U_i^n + U_{i+1}^n\right) - \Delta t\left[\frac{F_{i+1}^n - F_i^n}{\Delta x}\right]$$

$$U_{i-½}^{n+1} = \frac{1}{2}\left(U_i^n + U_{i-1}^n\right) - \Delta t\left[\frac{F_i^n - F_{i-1}^n}{\Delta x}\right] \tag{24}$$

Note that U is u or R depending upon which equation is being evaluated, 22 or 23, and F is the right hand side of equation 22 or 23 multiplied by - dx. The second step in the solution process is:

$$U_i^{n+1} = U_i^n - \Delta t\left\{\frac{1}{2}\left[\frac{F_{i+1}^n - F_{i-1}^n}{2\,\Delta x} + \frac{F_{i+½}^n - F_{i-½}^n}{\Delta x}\right]\right\} \tag{25}$$

The scheme is explicit and provides no artificial viscosity. Where shocks exist, artificial viscosity may be added to damp the overshoot created by the numerics but is generally not required. The stability condition is the classic Courant number restriction as both necessary and sufficient for stability.

Applying this method to a specific vane requires that the boundary conditions be addressed. In a vane PMD with a center post, the flow path is a loop. The boundary conditions can be handled by simply joining the 0 and Nth positions just as all other positions were joined. In addition, the flow rate out of the tank can be simulated by removing the appropriate volume of liquid from the segment centered about x = 0 and with a length Δx (i.e. half from the 1st segment and half from the Nth segment). The flow out can be assumed to diffuse out - neither adding nor subtracting from the momentum ($u - u_{Lx} = 0$).

In a vane which does not lend itself to circular flow paths, the boundary condition must be addressed by ignoring either conservation of momentum or continuity at the end of the flow path. The choice depends upon the nature of the flow and often it is educational to simulate both assumptions.

Using the analytical techniques outlined in this section, a vane PMD's performance can be verified whether the vane is refilling a sponge in zero g or steadily supplying propellant during a thruster induced acceleration. This analysis coupled with an examination of such problems as filling, draining, slosh loading, and center of gravity position can be used to validate a design completely.

Aside - Section V

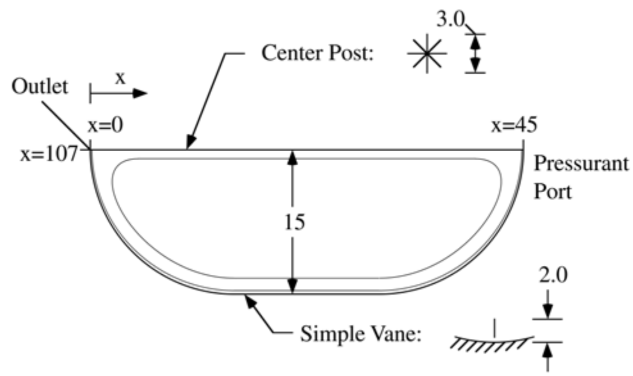
Example Figure 3 shows a tank similar to that previously presented in Aside - Section II but with a center post installed.

Example Figure 4 shows the radius of fillet curvature as a function of distance along the center post and vane at 0.30 in^3/sec and during a 0.0005 g lateral acceleration as determined by the steady model. The outlet is at x=0 & at x=107. The center post extends from x=0 to x=45, and the vane from x=45 to x=107. Several possible curves are presented - each corresponding to a different volume along the vane. When compared to the analysis of Aside - Section II, the results show that the rough order of magnitude analysis over-estimates the flow capacity of the vane by nearly a factor of two and thus modelling should be used to validate a design. One will also note that choking occurs as predicted by Table 1 - at a radius of 0.39 inches (corresponds to a flow rate of 0.15 in^3/sec - the flow rate along each vane).

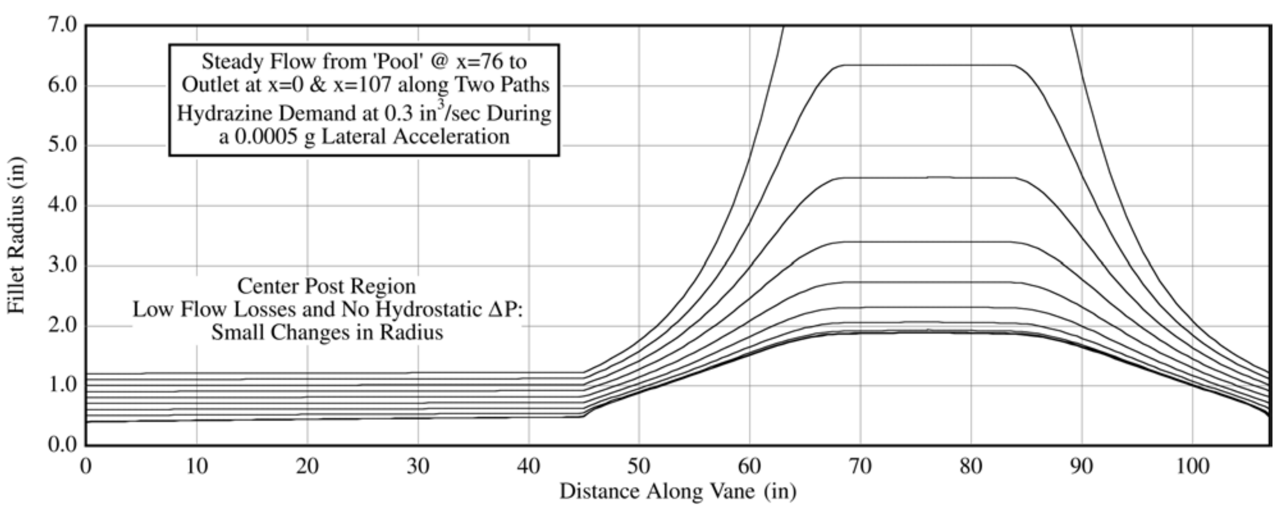
Example Figure 5 shows the result of an unsteady simulation showing the radius as a function of distance and times during an engine ignition. Initially the fillet radius is constant everywhere as in zero g. As demand continues, the acceleration causes the fluid to flow down the vanes from the center post. The surface tension forces stop and eventually reverse this downward flow establishing a steady flow up the vanes (and down the center post) to the outlet.

Finally, Example Figure 6 shows the depletion transient where the flow chokes and the radius decreases rapidly. The vane can no longer supply the center post at the required flow rate. The center post depletion occurs thereafter and could be modelled by the same technique (not illustrated). A conservative estimate of the residual propellant can be made by integrating the area along the vane.

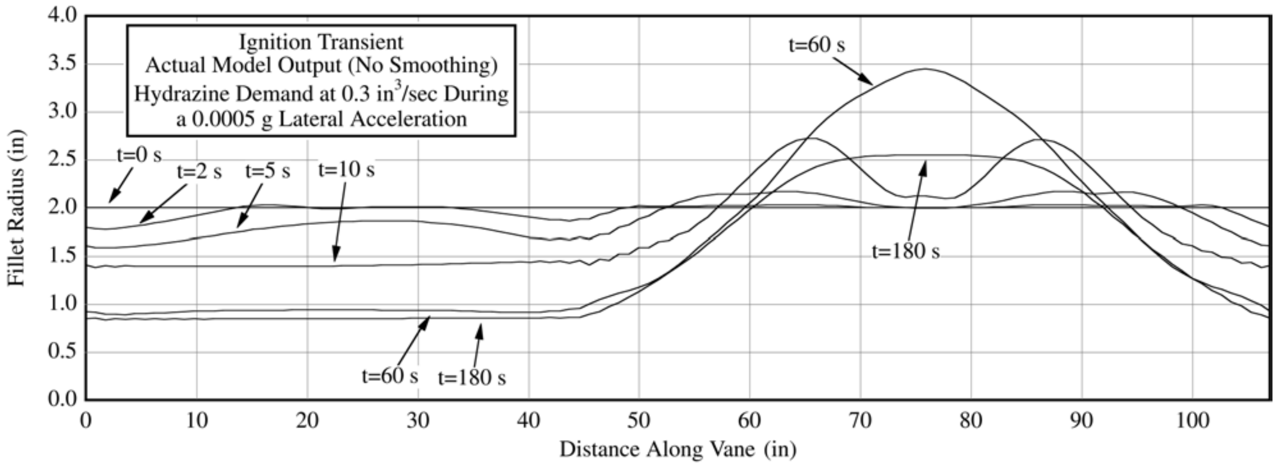
Please note that the results shown in Example Figures 5 and 6 are intermittent and computer simulations were conducted at smaller Δt (and were easily displayed as an animation).



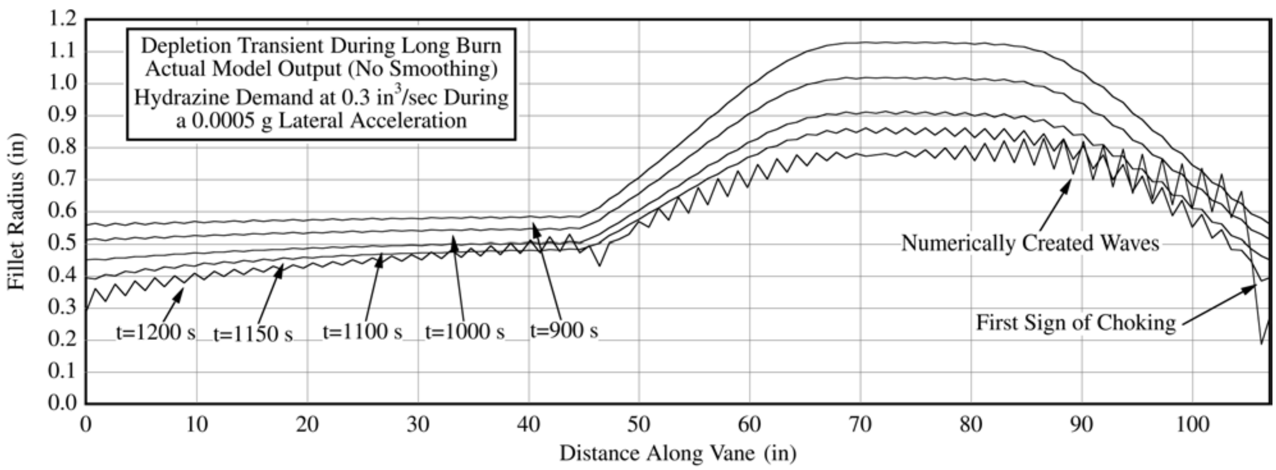
Example Figure 3. Center Posted Vane PMD Configuration

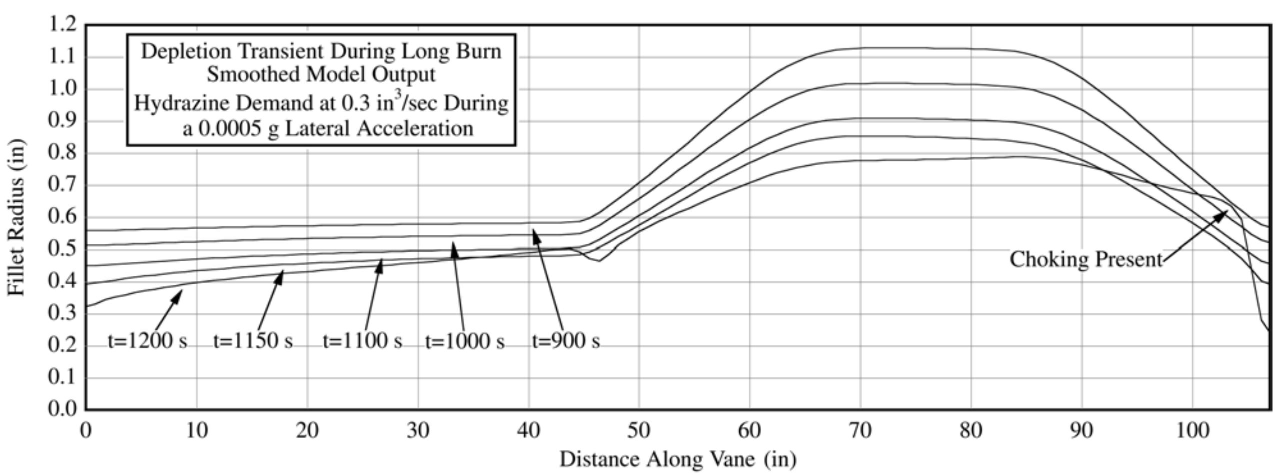


Example Figure 4. Steady Flow Along a Center Post and Simple Vane PMD



Example Figure 5. Unsteady Flow During Engine Ignition Along a Simple Vane PMD with Center Post





Example Figure 6. Unsteady Flow To Depletion Along a Simple Vane PMD with Center Post

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NOMENCLATURE

Greek

Δ ≡ change
μ ≡ absolute viscosity
ν ≡ kinematic viscosity
ρ ≡ liquid density
σ ≡ absolute surface tension
τ_w ≡ shear stress

Arabic

a ≡ acceleration
c ≡ the area wave propagation speed
q_L ≡ flow rate per unit length into the fillet
s ≡ wetted circumference
t ≡ time
u ≡ flow velocity
u_{Lx} ≡ the velocity at which q_L is added
x ≡ distance in flow direction
z ≡ height relative to acceleration vector

A ≡ flow area
D_e ≡ equivalent diameter
F ≡ function
F_{mult} ≡ friction multiplier
L ≡ length of the flow channel
P ≡ pressure
Q ≡ volumetric flow rate
R ≡ principal radius of curvature
SF ≡ safety factor
U ≡ dependent variable

Subscripts

drive ≡ driving
down ≡ downstream
gas ≡ pressurant gas
limit ≡ limiting
up ≡ upstream

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