LIQUID OXYGEN/LIQUID HYDROGEN
BOOST/VANE PUMP
for the
Advanced Orbit Transfer Vehicle
Auxiliary Propulsion System

ENGINEERING REPORT

SUNDSTRAND CORPORATION
SUNDSTRAND AVIATION FLUID PUMPING
PREPARED FOR
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

NASA LEWIS RESEARCH CENTER
CONTRACT NAS3-20401
A rotating, positive displacement vane pump with an integral boost stage was designed to pump saturated liquid oxygen and liquid hydrogen for APS of orbit transfer vehicle. This unit is designed to ingest 10% vapor by volume, contamination-free liquid oxygen and liquid hydrogen. The final pump configuration, predicted performance, and the major design work performed under this contract are included in this publication.
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1. SUMMARY

A rotating, positive displacement vane pump with an integral boost stage was designed to pump liquid oxygen and liquid hydrogen with the following requirements:

<table>
<thead>
<tr>
<th></th>
<th>LIQUID HYDROGEN</th>
<th>LIQUID OXYGEN</th>
</tr>
</thead>
<tbody>
<tr>
<td>Overall Efficiency</td>
<td>75%</td>
<td>75%</td>
</tr>
<tr>
<td>&quot;O&quot; - 10% Vapor</td>
<td>&quot;O&quot; - 10% Vapor</td>
<td></td>
</tr>
<tr>
<td>Life</td>
<td>125 Hours</td>
<td>125 Hours</td>
</tr>
<tr>
<td>Cycles (starts)</td>
<td>6000</td>
<td>6000</td>
</tr>
<tr>
<td>Fluid Inlet Temp. (°K)</td>
<td>20.2° (37°R)</td>
<td>90.2° (163°R)</td>
</tr>
<tr>
<td>Flow Rate (gr/sec)</td>
<td>28.6 (.063 lb/sec)</td>
<td>86.2 (.19 lb/sec)</td>
</tr>
<tr>
<td>Pressure Rise (N/M²)</td>
<td>1.49 x 10⁶ (216psi)</td>
<td>2.23 x 10⁶ (323psi)</td>
</tr>
</tbody>
</table>

Preliminary studies led to the conclusion that engineering effort was to concentrate on the liquid hydrogen pump and adapt the configuration to pump liquid oxygen. The two pumps were to be built from different material to meet the material compatibility with the fluids.

Analyses indicated that the LH₂ vane pump required two matching boost stage while the LOX vane pump took only one.

Analysis work done included vane stage cam profile, vane dynamics, thermal leakages, stresses, and various design calculations.
2. INTRODUCTION

Cryogenic fluid pumping is a newly developing technology in the field of fluid transfer. Up to the present, no successful attempt in building a cryogenic vane pump has been recorded. Both PESCO and General Motors have ventured into this field but their effort have been unrewarding. Nevertheless, PESCO liquid hydrogen vane pump test data has been proven to be an invaluable calibration tool for the mathematic model developed by Sundstrand's engineers in this conquest.

The objective of this program is to design and fabricate a boost vane pump package which will serve as a test vehicle for the technology development of small positive displacement pumps for liquid oxygen and liquid hydrogen for APS Feed System. Since this is a technology development program, neither weight nor size has been treated as a design constraint. In order to reduce the manufacturing cost, the displacement for the liquid hydrogen pump was used for both fluids.
3. DESIGN GUIDELINES

1. The preliminary design effort was to be guided toward, but not limited to:

   a. Stress analysis
   b. Bearing loads, life, and cooling requirements
   c. Seal leakage and life
   d. Internal leakage analysis
   e. Thermal analysis
   f. Rubbing speeds and loads
   g. Materials selection and heat treatment
   h. Oxygen and hydrogen compatibility of materials.

2. The boost/vane pump package was a self contained unit with its own seals and bearings. A mounting flange and coupling was to be provided for mounting and driving the unit from an external power source.

3. The pump displacement was to be selected so that the flow conditions was to be met at nominal pump speeds of either 4000, 6000, 8,000, 12000, or 24000 RPM. The oxygen pump and hydrogen pump requirements were such that, for the same displacement, they would require different operating speeds, but the speed selected for each fluid was to be one of those listed.

4. If a single pump design could not meet the flow rate requirements of both fluids, with the speed constraints listed above, a compromise must be made in the selection of the pump displacement. The displacement selected would be that which would provide for the greatest technology gain when considering both material selection and boost pump performance.

5. The ability of the pump to expel gas and recover nominal performance would be more critical than a momentary reduction in flow or pump efficiency.

6. The pump would be designed so that both the vane pump and boost pump components could be tested separately.

7. The vane pump would be designed so that material changes could be made in the area that rubbing contact occurs between the vanes and the pump housing.
4. MATERIAL SELECTION
MATERIAL SELECTION

LH₂ AND LO₂ CRYOGENIC VANE PUMPS

Evaluation of materials for usage at cryogenic temperatures was conducted taking into account the basic low temperature characteristics of materials. This constituted the rating of materials using the following basic parameters as applicable:

1) Low temperature mechanical strength
2) Low temperature impact strength
3) Low thermal contraction differential
4) Wear resistance

More importantly, material compatibility with LO₂ was established using what is termed as the "burn factor"(1) for selected materials.

The material selections are summarized in Table 4.1a and 4.1b and described below.

An attempt has also been made to fabricate the vane stage parts from similar, if not the same, materials in each respective pump. This is due to areas in the vane stage requiring tight tolerances. Similar materials thus similar contraction rates alleviate loss of these tolerances at cryogenic temperatures.

LH₂ Pump Vane Stage

The low lubricity characteristics of LH₂ dictates a wear resistant base material or coating for the rubbing surfaces. Most hardenable steels have been discounted due to their low temperature embrittlement effects or poor impact strength at cryogenic temperatures as exhibited by 4340 steel in Fig.4.1.(2) The 300 series stainless steels would not possess adequate wear resistance even though their hardness and strength increases at low temperatures (Fig.4.2). (2)

Investigation into suitable wear resistance led to the Ferro-TiC materials (Div. of Sintecast). The Ferro-TiC materials are a family of steel or alloy bonded carbides utilizing extremely hard TiC grains uniformly distributed through a hardenable metal matrix. A variety of metal matrices are available. The selected grade was HT-6 which is composed of a nickel alloy (Inconel 718) matrix with a 45% TiC particle content.

The nickel alloy matrix provides excellent low temperature strength (Fig.4.3)(2) while the TiC particles provide the essential wear resistance. (Similar Ferro-TiC grades (Ferro-TiC SK, CM) have exhibited excellent wear resistance for vane stage parts). Alternate material selection for the vane stage parts necessitated the use of dissimilar metals due to the low temperature embrittlement of hardenable conventional steels. The dissimilar combination (Table4.1a) utilizes a hard on soft material approach. A-286 stainless steel exhibited the optimum low temperature properties for the rotor and vanes while maintaining similar thermal contraction rates with the leaded bronze port plates and liner. To insure adequate wear resistance, the rotor ends and vanes will be electroless nickel plated. This combination insures that all rubbing contact will be composed of hard (60 RC) electroless nickel against the self lubricating leaded bronze.
The inducer, impeller and housings were originally to be cast from C355 aluminum although to provide for interchangeability with the LO2 pump, K Monel will be used. Similarly the shaft has been changed from 304L CRES to K Monel. These changes also provide for a closer match of thermal contractions on the inducer, impeller and shaft.

The labyrinth seal rings for the LH2 pump will be P5N carbon and 70/30 Brass will be an alternate choice.

The ball bearings will be made from 440C stainless steel with one piece rulon retainers which have shown excellent performance on the space shuttle LH2 and Centaur LO2 pumps.

LO2 Pump Vane Stage

Compatibility with LO2 was used as the primary basis of material selection for the LO2 pump. Rubbing metal to metal contact of the vane stage elements require materials possessing high ignition temperature, high thermal diffusivity and low heat of oxidation while maintaining wear resistance.

A few of the alloys used as preliminary candidates in the LO2 pump study are listed in Table 4.1 (1) Some thermodynamic properties of each of the alloys are also shown. The first column lists the density of the material. The second column presents the calculated heat evolved from each alloy assuming complete combustion of 100 grams of the material. The heat of oxidation was calculated by taking the concentration of each of the major constituents in the alloy multiplying the weight present by the standard heat of oxidation for each constituent and summing the component values to obtain a calculated heat of alloy oxidation. A large value for $\Delta H_f$, the standard heat of oxidation, implies large amounts of heat evolved in the oxidation of 100 grams of the alloy.

The third column in the table presents a calculated value of the diffusivity of each material. The diffusivity is defined by the following expression:

$$\alpha = \frac{k}{\varrho c}$$

where $\alpha$ = thermal diffusivity  
$k$ = thermal conductivity  
$\varrho$ = density  
c = specific heat

The diffusivity is a value representing the ability of a material to diffuse heat away from a heat source.

The fourth column in the table is the ratio of the heat of oxidation divided by the diffusivity and is referred to as the "burn factor". This number provides a relative ranking of each material in terms of the amount of heat produced when a fixed amount of material oxidizes divided by the diffusivity, the ability of the base metal to diffuse heat away. The high value of $\frac{\Delta H_f}{\alpha}$ implies either a high heat of oxidation or a low diffusivity, or a high ratio of the two factors.
Table III lists the materials of interest according to the burn factor, $\Delta H_f$. The materials at the top of the list are considered to be more resistant to burning because of either a low heat of oxidation or a high diffusivity or both. Materials appearing lower in the list have either a high heat of oxidation or low diffusivity and are expected to be less resistant to both ignition and combustion.

The prime and alternate materials selected for the LO$_2$ pump are shown in Table 4 lb. The Ferro-TiC CN-5 material to be used as the rotor, port plates, liner and vanes is composed of 45 V/O, tungsten carbide and 55 V/O, 70/30 brass. The tungsten carbide is present as interspersed particles in the 70/30 brass matrix. This combination of hard and soft constituents has provided excellent wear resistance in conventional aircraft fuel pumps. Compatibility of the Ferro-TiC CN-5 with LO$_2$ was estimated by calculating the "burn factor" for the alloy (Appendix).

The alternate materials selected for the LO$_2$ vane stage all appear high on the list in Table 4.11. This combination, as in the LH$_2$ version will provide a hard on soft approach and utilizes electroless nickel against leaded bronze as the mating materials.

The shaft, inducer, impeller and housings will be fabricated from K Monel due to its low burn factor and excellent mechanical properties at cryogenic temperatures. S Monel will be used if the parts are to be casted.

The bearing, as with the LH$_2$ pump, will be made from 440C stainless steel as used on the Centaur LO$_2$ and space shuttle LH$_2$ pumps.
The following calculation is performed to estimate the "burn factor" of Ferro-TiC CN-5. The heats of oxidation ($\Delta H_f$) for 100g of each constituent material (Cu, Ni, W, + C) is calculated initially. From the percentages of constituents present, a heat of alloy oxidation is arrived at $\Delta H_{f,CN-5}$ for 100g of Ferro-TiC CN-5. The diffusivity ($\alpha$) is then calculated similarly.

The ratio of the heat of oxidation and the diffusivity yields the "burn factor". ($\frac{\Delta H_f}{\alpha}$).

Property data for Ferro-TiC CN-5 is given below:
Density: 11.8g/cm$^3$
Composition: 45 V/O, WC

55 V/O, Cu-Ni alloy (similar to CA 715)
Since the Cu-Ni alloy matrix is similar to copper alloy 715, property data for this alloy will be used in calculation of the burn factor.

CA 715

\[
K = \frac{17 \text{ BTU (ft)}}{(ft^2)(hr)(^\circ F)} = \text{thermal conductivity}
\]

\[
C = \frac{.09 \text{ BTU}}{lb^\circ F} = \text{specific heat}
\]

WC

\[
K = \frac{57.8 \text{ BTU (ft)}}{(ft^2)(hr)(^\circ F)} = \text{thermal conductivity}
\]

\[
C = \frac{.049 \text{ BTU}}{lb^\circ F} = \text{specific heat}
\]

Since the density of Ferro TiC CN-5 is 11.8g/cc the amount of the constituent elements in 100g of the alloy is calculated as follows, keeping in mind that there is 45 V/O WC and 55 V/O CA 715.

\[
100g \text{ CN-5} \times \frac{cm^3}{11.8g} = 8.47cm^3 \text{ CN-5}
\]

45 V/O of 8.47cm$^3$ = 3.81cm$^3$ WC

55 V/O of 8.47cm$^3$ = 4.66cm$^3$ CA 715

Using the following densities the amount of constituent materials is found.
\[ \rho_{\text{CA 715}} = 8.94 \text{g/cm}^3 \]
\[ \rho_{\text{WC}} = 14.99 \text{g/cm}^3 \]

\[ \text{CA 715} \]
\[ 4.66 \text{cm}^3 \times \frac{8.94 \text{g}}{\text{cm}^3} = 41.66 \text{g CA 715} \]

\[ \text{WC} \]
\[ 3.81 \text{cm}^3 \times \frac{14.99 \text{g}}{\text{cm}^3} = 57.11 \text{g WC} \]

Therefore in 100g of Ferro-TiC CN-5 we have ~40 W/O CA 715 and 60 W/O WC.

We can now calculate the heat of oxidation for a 100 gram sample of the material. To do so we start with calculating the amount of each constituent element in the material:

For 40g CA 715 (approx. 70% Cu, 30% Ni)

\[ 70\% \text{ of } 40g = 28 \text{g Cu} \]
\[ 30\% \text{ of } 40g = 12 \text{g Ni} \]

For 60g WC

\[ 60g \times \frac{1 \text{ mole WC}}{195.86g} \times \frac{183.85gW}{\text{mole WC}} = 56.32 \text{g W} \]

\[ 60g \times \frac{1 \text{ mole WC}}{195.86g} \times \frac{12.01gC}{\text{mole WC}} = 3.68 \text{g C} \]

If we now calculate the amount of heat evolved in oxidizing 100 grams of the constituents, we will arrive at the heat of oxidation of the Ferro-TiC CN-5 by multiplying these results by the percent of constituent present and then adding them.

For complete oxidation of 100g of Cu the amount of heat evolved is calculated below. Since the oxidation of Cu\( \rightarrow \)CuO (-37.1Kg cal/mole) evolves more heat than any oxide product of copper, it will be used.
For complete combustion of 100g Ni the oxidation reaction: Ni→NiO (-58.4Kg cal/mole) will be used again because it produces the most heat when formed.

$$\text{mole} \quad \frac{100\text{g Ni}}{65.37\text{g}} \times \frac{-58.4\text{Kg cal}}{\text{mole}} = \text{-89.34Kg cal}$$

Similarly for W and C the heat evolved in the following reactions are used respectively: W→WO_{3} (-200.84Kg cal/mole) and C→CO_{2} (-94.05Kg cal/mole).

$$\text{mole W} \quad \frac{100\text{g W}}{183.85\text{g}} \times \frac{-200.84\text{Kg cal}}{\text{mole}} = \text{-109.24Kg cal}$$

$$\text{mole C} \quad \frac{100\text{g C}}{12.01\text{g}} \times \frac{-94.05\text{Kg cal}}{\text{mole}} = \text{-783.10Kg cal}$$

For 100g of Ferro-TiC CN-5 the expected heat evolved is calculated:

$$\text{28g Cu} \quad \frac{58.39\text{Kg cal}}{100\text{g Cu}} = \text{16.35Kg cal}$$

$$\text{12g Ni} \quad \frac{89.34\text{Kg cal}}{100\text{g Zn}} = \text{-10.72Kg cal}$$

$$\text{56.32g W} \quad \frac{-109.24}{100\text{g W}} = \text{-61.52Kg cal}$$

$$\text{3.68g C} \quad \frac{-783.10\text{Kg cal}}{100\text{g C}} = \text{-28.82Kg cal}$$

Total = \text{-117.41Kg cal}

i.e. heat of oxidation for Ferro TiC CN-5

$$\Delta H_{fCN-5} = \text{-117.41Kg cal/100g}$$

The thermal diffusivities of CA 715 and WC are now calculated as below:
CA 715 → diffusivity \[ k = \frac{17 \text{ BTU ft}}{\text{ft}^2 \text{ hr} \ ^\circ F} \times 4.135 \times 10^{-3} = \frac{.07 \text{ cal cm}}{\text{sec cm}^2 \ ^\circ K} \]

\[ C = \frac{.09 \text{ BTU}}{\text{lb} \ ^\circ F} = \frac{.09 \text{ cal}}{\text{g} \ ^\circ K} \]

\[ \rho = \frac{.323 \text{ lb}}{\text{in}^3} \times \frac{454g}{\text{lb}} \times \left( \frac{\text{in}}{2.54cm} \right)^3 = 8.949 \text{g/cm}^3 \]

\[ \alpha = \frac{K}{\varphi c} = \frac{.07 \text{ cal cm} \ ^\circ C}{\text{sec cm}^2 \ ^\circ K} \left( \frac{8.949 \text{g/cm}^3}(.049 \text{ cal/g} \ ^\circ K) \right) \]

\[ \alpha_{CA \ 715} = \frac{8.6 \times 10^{-2} \text{cm}^2}{\text{sec}} \]

WC diffusivity at (212 \ ^\circ F)

\[ K = \frac{57.8 \text{ BTU}}{\text{hr ft ft}^2 \ ^\circ F} \times 4.135 \times 10^{-3} = \frac{.239 \text{ cal cm}}{\text{sec cm}^2 \ ^\circ K} \]

\[ C = \frac{.049 \text{ BTU}}{\text{lb} \ ^\circ F} = \frac{.049 \text{ cal}}{\text{g} \ ^\circ K} \]

\[ \rho = \frac{.54 \text{ lb}}{\text{in}^3} \times \left( \frac{\text{in}}{2.54cm} \right)^3 \times \frac{454g}{\text{lb}} = 14.99 \text{g/cm}^3 \]

\[ \alpha_{WC} = \frac{.239 \text{ cal cm}}{\text{sec cm}^2 \ ^\circ K} \left( \frac{14.99 \text{g/cm}^3}(.049 \text{ cal/g} \ ^\circ K) \right) = \frac{.325 \text{ cm}^2}{\text{sec}} \]
Since there is 45 V/O WC and 55 V/O 70/30 brass the thermal diffusivity for the alloy is calculated as follows:

\[
0.45 \times \frac{0.325\text{cm}^2}{\text{sec}} = 0.146\text{cm}^2/\text{sec}
\]

\[
0.55 \times \frac{8.6\times10^{-2}\text{cm}^2}{\text{sec}} = \frac{0.047}{0.193} \text{cm}^2/\text{sec}\n\]

Total

\[
\alpha_{\text{CN-5}} = 0.193\text{cm}^2/\text{sec}
\]

i.e. the thermal diffusivity of Ferro TiC CN-5 is:

\[
\alpha_{\text{CN-5}} = \frac{0.193}{0.193}\text{cm}^2/\text{sec}
\]

The burn factor can now be calculated for the alloy.

\[
\text{Burn factor} = \frac{\Delta H_f}{\alpha_{\text{CN-5}}} = \frac{121.96}{193}
\]

\[
\text{Burn factor} = 631.92
\]

Comparison of this number with Table II indicates that the Ferro-TiC CN-5 has a burn factor in the neighborhood of the bronze alloys + pure aluminum as anticipated and should be quite safe in liquid oxygen. Concurrently, the material will provide excellent wear characteristics based on the 45 V/O WC content.
Fig. 4.1 - IMPACT STRENGTH OF 4340 STEEL
Fig. 4.2 - YIELD STRENGTH OF 304 STAINLESS STEEL
Fig. 4.3 - YIELD STRENGTH OF INCONEL 718

NOTE: ANNEALED AND AGED, LONGITUDINAL.
(124) 1800°F + 1325°F/8 HR, FC TO 1150°F,
HOLD 18 HR, AC.
(205) SAME AS (124)
(206) A 1950°F + 1400°F/10 HR, FC TO 1200°F,
HOLD 20 HR, AC.
(206) D SAME AS (124).
**CRYOGENIC VANE PUMP MATERIAL SELECTION**

<table>
<thead>
<tr>
<th>Table 4.1a</th>
<th>L(_2) Prime Candidates</th>
<th>Alternates</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rotor</td>
<td>Ferro-TiC</td>
<td>A-286 (Ni Plated Ends)</td>
</tr>
<tr>
<td>Liner</td>
<td>H1T-6</td>
<td>Leaded Bronze</td>
</tr>
<tr>
<td>Port Plates</td>
<td></td>
<td>Leaded Bronze</td>
</tr>
<tr>
<td>Vanes</td>
<td></td>
<td>Nickel Plated A-286</td>
</tr>
<tr>
<td>Shaft</td>
<td></td>
<td>K Monel *</td>
</tr>
<tr>
<td>Inducer</td>
<td></td>
<td>S Monel</td>
</tr>
<tr>
<td>Impeller</td>
<td></td>
<td>S Monel</td>
</tr>
<tr>
<td>Labyrinth</td>
<td></td>
<td>P-SNR</td>
</tr>
<tr>
<td>Bearings</td>
<td></td>
<td>440C Rulon Cage</td>
</tr>
<tr>
<td>Housings</td>
<td></td>
<td>K Monel</td>
</tr>
<tr>
<td>Bolts</td>
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<td>Nitronic 60</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Table 4.1b</th>
<th>LO(_2) Prime Candidates</th>
<th>Alternates</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rotor</td>
<td>Ferro-TiC</td>
<td>Nickel Plated (End Faces) Be-Cu</td>
</tr>
<tr>
<td>Liner</td>
<td>CN-5</td>
<td>Leaded Bronze</td>
</tr>
<tr>
<td>Port Plates</td>
<td></td>
<td>Leaded Bronze</td>
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<tr>
<td>Vanes</td>
<td></td>
<td>Ni Plated Be-Cu</td>
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<tr>
<td>Shaft</td>
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<td>K Monel</td>
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<td>Inducer</td>
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<td>Bolts</td>
<td></td>
<td>Nitronic 60</td>
</tr>
</tbody>
</table>

*Monel material will be Ni Strike ~ .001" Cu Plated due to the hydrogen sensitive nature of the material.*
### TABLE 4.11

Calculated Heats of Oxidation and Ratios of Those Heats and the Thermal Diffusivity (n) of Alloys of Interest (at Room Temperature)

<table>
<thead>
<tr>
<th>Type of Alloy</th>
<th>Heat of Oxidation $\Delta H_f$ kg/cals/100 gms. alloy</th>
<th>Diffusivity $a$, cm$^2$/sec</th>
<th>Burn factor $\Delta H_f/a$</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Iron Alloys</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Cast iron, plain</td>
<td>7.20</td>
<td>210.2</td>
<td>.129</td>
</tr>
<tr>
<td>Cast iron, alloyed</td>
<td>7.20</td>
<td>207.8</td>
<td>.117</td>
</tr>
<tr>
<td>Ductile iron</td>
<td>7.11</td>
<td>209.3</td>
<td>.085</td>
</tr>
<tr>
<td>C-Steel, cast</td>
<td>7.83</td>
<td>179.9</td>
<td>.136</td>
</tr>
<tr>
<td>1025 steel</td>
<td>7.83</td>
<td>179.9</td>
<td>.136</td>
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### TABLE 4.11

**Ranking of Alloys According to Burn Factor - $\Delta H_f/\alpha$**

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<tr>
<th>Alloy</th>
<th>$\Delta H_f/\alpha$</th>
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<tr>
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<td>Leaded Bronze</td>
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</tr>
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<td>8630</td>
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5. THERMAL LEAKAGE ANALYSIS
Fictitious Station

Vane Pump shown in circle for representation only. All work in compression, in leakage, in friction etc. is designated by W_actual.
A vane pump can be designed successfully for pumping cryogenic fluids, if clearances between moving and stationary components are maintained as low as possible. In cryogenic operation, leakage affects the pump performance adversely in two ways: (a) reduction in the volumetric efficiency, and (b) possibility of creating a two phase mixture at the inlet zone of the pump.

Two types of leakage occur in a vane pump: (1) internal, and (2) external. In internal leakage the fluid from the exit (or high pressure) port side is dumped into the inlet (or low pressure) port side of the pump. In external leakage the fluid flows from the exit (or the high pressure) port side to low pressure zones but not to the inlet port of the pump itself.

In the present analysis internal leakage has been considered at the following points:

(a) vane tips
(b) vane sides
(c) rotor sides
(d) kidney-to-kidney in port plates, and
(e) pressure plate sides;
and external leakage has been considered from kidneys to shaft spline.

Clearance or dead volume also reduces the pump performance. Figure 1 shows the flow diagram for a vane pump, and includes the boost pump components at the upstream as inlet side.

Analysis

Using time average values, the flow through the pump may be treated as steady-state. The pump is submerged in saturated liquid. Analysis shows that the variation in fluid temperature as it flows through various passages is small enough to preclude significant heat transfer. Hence the flows are assumed adiabatic to err or a safer side and obtain a conservative
design. Variation in the kinetic and potential energy terms is in general negligible. The fluid density (particularly of hydrogen) may vary somewhat. However, momentum equations for incompressible fluids can be used here with an average density to account for its variation.

Continuity

\[ \dot{m}_a = \dot{m}_b = \dot{m} \]  \hspace{1cm} \text{(1)}

where \( a \) and \( b \) are the inlet and the exit stations of a control volume, and

\[ \dot{m} = \text{fluid flow rate, kg per sec.} \]

Momentum

\[ \dot{m}(u_b - u_a) = \left\{ p_a A_a - p_b A_b + F_{sx} \right\} \]  \hspace{1cm} \text{(2)}

where

\[ p = \text{pressure, N/m}^2 \]
\[ A = \text{area, m}^2 \]
\[ F_{sx} = "\text{shaft" force exerted upon the fluid, Newtons} \]
\[ u = \text{fluid velocity, ft per (sec), m/sec} \]

Energy

Considering adiabatic flow, and neglecting the kinetic and the potential energy terms,

\[ \frac{\dot{W}_m}{J} = \dot{m} (h_b - h_a), \hspace{1cm} \text{(3)} \]
where
\[ W_m = \text{work input rate, } \text{n. m per sec.} \]
\[ J = 778.16 \text{ n. m / kcal} \]
\[ h = \text{specific enthalpy, } \text{kcal per Kg} \]

**Entropy**

\[ \dot{m}(S_b-S_a) = \Sigma \geq 0, \]

--- (4)

where
\[ S = \text{specific entropy, Kcal per Kg. } ^0K \]
\[ \Sigma = \text{entropy production rate, kcal per sec} \]

We also have
\[ TdS = dh - \frac{dp}{\dot{J} \rho} \]

--- (5)

which may be integrated to give

\[ T_{ab} (S_b-S_a) = (h_b-h_a) - \frac{1}{J \rho_{ab}} (p_b-p_a) \]

--- (6)

where
\[ T_{ab} = \"Average\" \text{ of } T_a \text{ and } T_b \]
\[ \rho_{ab} = \"Average\" \text{ of } \rho_a \text{ and } \rho_b = \frac{1}{2} (\rho_a + \rho_b) \]

If a-b is an ideal pump, \( \Sigma \) is zero and from equations 3, 4 and 6 have

\[ \frac{(W_m)}{\text{ideal}} = \frac{\dot{m}}{\rho_{ab}} (p_b-p_a) \]

--- (7)
\[
\left( \dot{N}_m \right)_{\text{ideal}} = \dot{Q} \left( P_b - P_a \right) \quad \text{(8)}
\]

where

\[
\dot{Q} = \text{volume flow rate, m}^3 \text{ per sec}
\]

**Internal Leakage**

Since the leakage paths have small gaps and a good design requires low leakage flows, these flow can be treated as laminar. A detailed analysis of such flow between parallel rectangular plates (with relative motion between them) shows that we have:

\[
\dot{Q}_j = \frac{\nu_0 b t_j^3}{12} \left( \frac{P_{ij} - P_{out}}{t_j} \right) \quad \text{(9)}
\]

in the direction of \( \nu_0 \),

where the subscript \( j \) is for the leakage path \( j \), and

\[
\nu_0 = \text{velocity of the moving plate, m per sec.}
\]

\[
b = \text{clearance between plates, m.}
\]

\[
t = \text{"thickness" of flow path (at right angles to flow), m}
\]

\[
l = \text{length of flow path, m}
\]

\[
\mu = \text{fluid viscosity, average value, (lbm) per (ft) (sec) (kg) per (m)(sec)}
\]

Now

\[

\nu_{oj} = \frac{2 \pi r_j N}{60} \quad \text{(10)}
\]

where

\[
r_j = \text{"radius" or "location" of the moving element } j \text{ from shaft center line, ft., m.}
\]

\[
N = \text{RPM}
\]
Since internal leakage includes only flows from high pressure to low pressure side of the pump, we also have

\[ \dot{Q}_j = \alpha_j \dot{Q}_j \]  

\text{---------------------------(11)}

where

\[ \alpha_j = 1.0, \text{ if } P_{inj} > P_{outj} \]
\[ \alpha_j = 0.0, \text{ if } P_{inj} = P_{outj} \]
\[ \alpha_j = 0.0, \text{ if } P_{outj} > P_{inj} \text{ and } \dot{Q}_j > 0 \]
\[ \alpha_j = -1.0, \text{ if } P_{outj} > P_{inj} \text{ and } \dot{Q}_j < 0 \]

Thus some leakage paths may be "active" while the others are "inactive".

It can also be shown that the work transferred to the fluid by the moving element of the leakage path \( j \) is given by

\[ W_j = \left| \alpha_j \right| \left( \frac{2\pi N}{60} \right) \left[ \frac{2\pi \eta h L^4 r_j^2}{60 \beta_{j0}} + \frac{b_j r_j t_j (P_{outj} - P_{linj})}{2} \right] \text{ ft.lbf/sec} \]  

\text{---------------------------(12)}

Following subscripts in Figure 1 (flow diagram), the total internal leakage flow \( \dot{m}_7 \) is given by

\[ \dot{m}_7 = \rho_{67} \sum_j \left| Q_j \right| \text{ kg per (sec)} \]  

\text{---------------------------(13)}

where \( \rho_{67} = 1/2 (\rho_6 + \rho_7) \),

and the total work input to internal leakage flows is given by:
\[ \dot{m}_{67} = \sum_j \dot{m}_j \]  

\textbf{Clearance, Dead or Carry-around Volume}

The flow returning to the inlet zone of the pump due to clearance volume is \( \dot{m}_8 \). It can be shown that

\[ \dot{m}_8 = \rho_6 \frac{V_c N}{60} \text{ kg per (sec)} \]  

where

\[ V_c = \text{clearance volume per revolution, } m^3 \]

The clearance volume fluid expands from \( p_6, h_6 \) to \( p_8 \) (or \( p_4 \), \( h_8 \)), and in so doing transfers work to the moving metals of the pump. To err on the safer side, it will be assumed that the work transfer is zero. From equation 3, we thus have

\[ h_8 = h_6 \]  

\textbf{External Leakage}

This occurs between the kidneys in the rotor and the shaft spline. The flow may be assumed to have a line source at the kidney edge (bottom) radius and a line sink at the spline radius. The difference between these radii is not very large, and hence the flow equation in this case at zero rotor speed should reduce to equation 9 with \( v_p \) equal to zero. Taking this into account and assuming the average tangential velocity of the fluid as equal to half of the rotor velocity, solution of the radial component of the momentum equation gives:

\[ \dot{m}_{11} = \frac{2p_m v_r m_b k_b}{6 \mu_m \nu_m} \left[ g_0 (p_6 - p_4) - \rho_m \left( \frac{\pi}{60} \right)^2 l_m r_m \right] \text{ kg per (sec)}, \]
with kidneys on either side of rotor. In the above equation

\[ \rho_m = \frac{1}{2} (\rho_6 + \rho_{11}), \text{ lbm per (cu ft)} \]

\[ r_m = \frac{1}{2} (r_{\text{spline}} + r_{\text{kidney}} - \frac{1}{2} t_{\text{kidney}}), \text{ m} \]

\[ \nu_m = \frac{1}{2} (\nu_6 + \nu_{11}), \text{ kg per (sec) m} \]

\[ l_m = (r_{\text{kidney}} - \frac{1}{2} t_{\text{kidney}} - r_{\text{spline}}), \text{ m} \]

\[ b_{ks} = \text{kidney-side clearance, m} \]

The work transfer to \( \dot{m}_{11} \) was found to be a negligibly small quantity. Hence from Equation 3, we have

\[ \dot{h}_{11} = \dot{h}_6 \quad \text{----------------------------------------(18)} \]

**Mixing Zone at Vane Pump Inlet**

Applying continuity and the energy equations, we have

\[ \dot{m}_5 = \dot{m}_4 + \dot{m}_7 \quad \text{----------------------------------------(19)} \]

\[ \dot{m}_5 h_5 = \dot{m}_4 h_4 + \dot{m}_7 h_7 \quad \text{----------------------------------------(20)} \]

We also have

\[ p_5 = p_4 = p_7 \quad \text{----------------------------------------(21)} \]

**Junction 5-8-9**

Here we have

\[ p_5 = p_8 = p_9 \quad \text{----------------------------------------(22)} \]
\[
\dot{m}_9 = \dot{m}_5 + \dot{m}_8 \tag{23}
\]

and

\[
\dot{m}_9 h_9 = \dot{m}_5 h_5 + \dot{m}_8 h_8 \tag{24}
\]

**Pump Efficiencies**

The overall efficiency \(\eta_{ovp}\) is defined by

\[
\eta_{ovp} = \frac{(\dot{m}_h)_{ideal}}{(\dot{m}_h)_{actual}} \tag{25}
\]

at the same fluid flow rate. Thus from equations 3, 7 and 25,

\[
\eta_{ovp} = \frac{(p_6 - p_4)}{\sqrt{\rho_{46s}(\eta_6 - \eta_4)}} \tag{26}
\]

where

\[
\rho_{46s} = \frac{1}{2} (\rho_4 + \rho_6s)
\]

\[
\rho_6s = \rho \text{ at } p = p_6 \text{ and } S_6 = S_4
\]

The volumetric efficiency \(\eta_v\) of the pump is given by

\[
\eta_v = \frac{(60) \dot{m}_9}{\rho_4 V_d N} \tag{27}
\]

where \(V_d\) is the displacement volume in cuft per revolution. Alternatively \(\eta_v\) is given by

\[
\eta_v = \frac{60}{\rho_5 V_d N} \left\{ \frac{\rho_4 (V_d + V_{11}) N}{60} - \frac{\rho_5 \dot{m}_7 - \dot{m}_7}{\rho_5} \right\} \tag{28}
\]
The terms can be obtained from equations 13, 15 and 17. Substituting for \( \dot{m}_8 \) equation 15, we have

\[
\eta_v = 1 - \frac{V_c}{V_d} \left[ \frac{p_6}{p_5} - 1 \right] - \frac{60}{\rho_4 V_d} \left( \dot{m}_7 + \dot{m}_{11} \right)
\]

as an alternative to equation 27. With a constant density fluid, \( V_c \) has no effect of \( \eta_v \) (which of course should be the case).

The torque efficiency \( \eta_t \) of the pump is defined by

\[
\eta_t = \frac{T_i}{T_a} = \left\{ \frac{p_4 V_d N (V_2 - p_3)}{2 \pi N p_4 c_s} \right\} \left( \frac{\dot{m}_6 (h_c - h_4) 60}{2 \pi \eta_s} \right)
\]

using equations 3 and 7. Thus from equations 26, 29 and 30

\[
\eta_{vop} = \eta_t \eta_v
\]

as usual. Thus if \( \eta_t \) is known, \( \eta_{vop} \) can be obtained from \( \eta_v \).

**Fluid State at Various Locations**

The state of the fluid at 4 is completely defined. Also \( p_6, \eta_t, N \) and pump geometry are prescribed.

Equation 26 can be used to determine \( h_6, T_6, p_6 \) etc.

From equations 3 and 26, we have

\[
h_7 = h_4 + \frac{1}{\gamma} \left\{ \frac{(p_6 - p_3)}{\rho_4 c_s^2 \eta_{vop}} + \frac{\dot{m}_7}{m_7} \right\}
\]
to obtain the properties at point 7. For \( \dot{m}_7 \) and \( \dot{h}_{67} \) use equations 13 and 14.

From equations 19, 20, 26 and 32 we get

\[
h_5 = h_4 + \frac{1}{\dot{m}_4 + \dot{m}_7} \left\{ \dot{m}_7 \left( h_6 - h_4 \right) + \frac{\dot{h}_{67}}{J} \right\}
\]

(33)

to obtain the properties at location 5.

From equations 19, 23, 24 and 33 we have

\[
h_9 = h_4 + \frac{1}{\dot{m}_4 + \dot{m}_7 + \dot{m}_8} \left\{ \left( \dot{m}_7 + \dot{m}_8 \right) \left( h_6 - h_4 \right) + \frac{\dot{h}_{67}}{J} \right\}
\]

(34)

to obtain the properties at location 9.

From equations 26 and 34, we can also write

\[
h_6 - h_9 = \frac{\dot{m}_4}{\dot{m}_4 + \dot{m}_7 + \dot{m}_8} \left\{ \frac{P_6 - P_4}{\rho_{46s} \eta_{vop}} - \frac{\dot{h}_{67}}{J \dot{m}_4} \right\}
\]

(35)

From figure 1 and equation 3, it is obvious that \( h_6 > h_9 \), and so

\[
\frac{\dot{m}_4}{\rho_{46s} \eta_{67}} > \eta_{vop}
\]

(36)
which can be used as a check in the calculation of various terms.

The location 10 in Figure 1 is a point at the proximity of the vane where the fluid at 9 has undergone an isentropic and adiabatic acceleration. Thus we have

\[ u_g = 0 \text{ but } u_{10} > 0, \text{ and so} \]

\[ h_g = h_{10} + \frac{u_{10}^2}{2S^g} \]

and

\[ S_g = S_{10} \]

where

\[ u_{10} = \frac{2\pi R_{maj} N}{60} \text{ m per sec} \]

\[ R_{maj} = \text{Cam Ring major radius, m} \]

Thus \( P_{10} \) and \( T_{10} \) can be obtained.

In a pump with a large number of vanes, it is possible that a leakage path \( j \) may actually be made up of more than one "leakage channel" in series. Calculations with "typical" cases show that a series combination of two channels can be considered as a single channel with twice the length and 0.98 of the thickness of one channel.

Programs for properties of hydrogen and oxygen (in liquid as well as the vapor phase) were obtained from NASA. After eliminating some errors from this material, subroutines have been made to obtain the fluid properties.

**Evaluation Procedure**

Dimensions of components and leakage paths are listed. Inlet conditions (location 4), \( P_6, \eta_t \) are \( N \) are specified.
From a sketch or a drawing showing vane positions, ports, kidneys etc., Pinj and Poutj for each leakage path j is determined.

Assuming \( \dot{m}_4 \) and \( \dot{m}_{11} \), \( \eta_y \) and \( \eta_{vwp} \) are calculated from equations 27 and 31. Properties at locations 6, 8 and 11 are obtained from equations 26, 16 and 18 respectively.

Using equation 17, \( \dot{m}_{11} \) is calculated and checked against the assumed value. Assumed \( \dot{m}_{11} \) is altered until agreement is obtained.

Assuming \( h_7 \), \( \dot{m}_7 \) and \( \dot{m}_{67} \) are obtained from equations 13 and 14. Using equation 32, \( h_7 \) is calculated. Iteration is necessary to make the assumed \( h_7 \) equal to the calculated \( h_7 \).

Properties at 5 are calculated using equation 33. The \( \eta_y \) is calculated from equation 29 and checked against the value based on assumed \( \dot{m}_4 \) and \( \dot{m}_{11} \). Iteration is necessary to obtain correct \( \dot{m}_4 \) resulting in the same \( \eta_y \) from equations 27 and 29.

Properties at 9 and 10 are evaluated from equations 34 and 37. Equation 36 is used as a check.

The quality of the fluid is checked at various locations. In a properly designed pump, the fluid should be subcooled at location 10.

In the event of a two phase flow through a leakage path, the flow is assumed homogeneous. The density of the fluid is the density of the mixture; whereas the liquid viscosity is used in the flow equations. Choking of a two phase flow through a passage is possible. The problem is complicated by the two phase fluid and the passage with moving boundaries. Choking, if it should occur, limits the flow and as a result, the error caused by not taking it into account will be on the safer side.

**Conclusion**

A computer program of the analysis has been made. A cryogenic pump made and tested a few years ago was used as a sample to check the program. The agreement between the measured and the calculated values has been found to be quite good. As a result liquid \( \mathrm{H}_2 \) and \( \mathrm{O}_2 \) vane pumps may be designed with the help of this program. The program may also be used for non-cryogenic operations such as fuel pumps.

The computer program output consists of the fluid condition (including its quality) at all locations, the volumetric efficiency, and the pump capacity, as a function of the inlet condition, delivery pressure, speed, torque efficiency and the pump geometry. The computer printout is shown in Fig. 5.2.
**WRITE PRINT,ZAD10**

**DATE SET ZAD10 AT LEVEL 004 AS OF 11/10/78**

**ANALYSIS OF THE PERFORMANCE OF VANE PUMPS WITH REFERENCE TO LEAKAGE IN PUMPING LIQUID HYDROGEN AND LIQUID OXYGEN**

**REAL LMK,NR,PRP,PD1,PK,MLT,MLTIL,MU,MU8**

**LOGICAL VAPOR**

**INTEGER ERROR,ENTRY**

**DIMENSION P=OPS(8)**

**DIMENSION POUTV(200),PINVT(200)**

**DIMENSION TITLE(20)**

**COMMON L(200),R(200),T(200),P(200),P4,P6**

**COMMON/FLUIDC/GAMMA,WL,KG,DENB,DENL,ENTL,ENTG,ENTH,ENTL,ENT**

**PI0000 CONTINUE**

**ICRD=5**

**IPRT=3**

**P13=1.159**

**C $101**

**READ(ICRD,11)**

**END=5102 TITLE**

**READ(ICRD,10)**

**NVANES,PRORM,NKID,NCAM,NAMGAS**

**READ(ICRD,30)**

**T4,P4,P6,RPRP,PD1,PNED,DELP4,YTHROW**

**READ(ICRD,35)**

**RMAJ,RMIN,RROT,WIDTH,VMGT,WTHICK,TROT**

**READ(ICRD,20)**

**1S,VST,VST=VHTH~THICK**

**READ ICRD,20**

**(PINVT(J)=J=1,NVANES)**

**WRITE(IPRT,12)**

**FORMAT(20A4)**

**FORMAT(15A4)**

**IF(RKS.EQ.0.0) RKS=(RROT+ROSPLN)/2.0**

**P7=P4**

**P9=P4**

**P9=P9**

**PINPV1=PV**

**VOLD=2.0*PI*(RMAJ+RMIN)/2.0**

**VOLC=2.0*PI*(RMAJ+RMIN)/2.0**

**TVT=WIDTH**

**TVS=YTHROW**

**TRS=RROT-(RKS+0.5)**

**LVS=WTHICK**

**LRS=12.0*PI**

**IC5=PI*(RMAJ+RMIN)**

**RVS=VTHROW**

**RKS=RROT-0.5**

**RMAJ-ROSPLN**

**IF(NAMGAS.EQ.1) GO TO 1**

**IF(NAMGAS.EQ.2) GO TO 2**

**CALL 02RCM**

**WRITE(IPRT,5)**

**WHITE(IPRT,5)**

**GO TO 8**

**CALL PN2**

**WRITE(IPRT,6)**

**CONTINUE**

**IF SOME VALUE OF B SHOULD BE ZERO THE PROGRAM WILL MAKE IT NON-ZERO AND SET THE CORRESPONDING T VALUES EQUAL TO ZERO. THIS WILL AVOID MEANINGLESS QUOT AND WDOUT EQUATIONS.**

**IF(BVNANE(0.0)) GO TO 40**
THE NUMBERING SCHEME FOR VARIABLES WITH SUBSCRIPT J IS AS FOLLOWS

NVAES = NUMBER OF VANES MAX. NUMBER OF VANES IS 20
NROTOR = NUMBER OF ROTOR SIDE LEAKAGE PATHS MAX IS 10
NKID = NUMBER OF KIDNEY SIDE LEAKAGE PATHS MAX IS 10

J=1, 2, ..., NVANES FOR VT
J=1, 2, ..., (20+NVAES) FOR VS SIDE
J=1, 2, ..., (60+NVAES) FOR VS SIDE
J=1, 2, ..., (60+NROTOR) FOR ROTOR SIDES
J=1, 2, ..., (70+NKID) FOR KIDNEY SIDE
J=1, 2, ..., (80+NCAM) FOR CAM SIDE

DO 100 J=1, NVANES
L(J)=LVT
R(J)=RVT

CONTINUE
CALL CM(1, T4*OUT)
TH4=OUT
CALL CM(2, P4*OUT)
P4=OUT
ENTRY=3
NP=1
CALL FLUID(TM, PM, DUPR, NP, ENTRY, VAPOR, ERROR)
CALL CB1(3, D*H4)
CALL CB1(4, PROPS(2), S4)
CALL CB1(5, PROPS(3), H4)
CALL CB1(6, PROPS(7), H4)
C
C ITERATION ON MOOT4 (ETAV1T)
ICON=1
C
C INITIALIZE MOOT4
TH4=0.0
XM4=V0.0*RPM*RH04/60.4
C
C CONTINUE
MOOT4=(XM4L*XM4R)/2.0
C
C CALCULATION OF PROPERTIES AT 6 AND 8
CALL CM(2, P6*OUT)
PM6=OUT
PM6=PS6
CALL CM(4, S5*PS6M)
PROPS(2)=PS6M
NP=7
ENTRY=4
CALL FLUID(TEMP, PM6, D*PROPS, NP, ENTRY, VAPOR, ERROR)
IF (ERROR.NE.0) WRITE(1PRT,555) ERROR
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PREM=0

CALL CB1(1,TEMP,PT6)

CALL CB2(1,PROP5,PROP6)

RH048*(RH04*PR6)/2.0

C

ITERATION ON MOOT11

XM11=0.0

XM11=MOOT4

ICOUNT=1

225 CONTINUE

MOOT11=(XM11+XM11)/2.0

MOOT6=MOOT4-MO0T11

ETA0V=ETA*ETA

H06=H04+(144.5*(P6-P4)/((778.16*RH046*ETA0V)

H06=H06

CALL CM1(5,H06OUT)

H06OUT=PROP3*H06

NPROPS=7

ENTRY=5

CALL FLUID(TEMP,PM6,D,PROPS,NPROPS,ENTRY,VAPOR,ERROR)

IF ERROR NE 0 WRITE (IPRT,555) ERROR

VP6=VAPOR

CALL CB1(1,TEMP,TVNEW)

IF(VP6) GO TO 215

CALL CB3(3,D,RH06)

GO TO 216

215 CALL VAPR(1,XVAL6,S6,RH06,VOID6)

NP7=1

CALL FLUIDL(PROPS,NP,ERROR)

216 CONTINUE

CALL CB1(6,PROPS(7),XM06)

T6NEW=ENTRY=5

CALL FLUID(TEMP,PM6,REM,PROPS,NPROPS,ENTRY,VAPOR,ERROR)

CALL CB1(1,TEMP,T6SAT)

C

CALL CM1(2,P6,P8H)

H06=H06

CALL CM1(5,H6,H8M)

PROPS(3)=H06M

ENTRY=5

NPROPS=7

CALL FLUID(TEMP,PM6,D,PROPS,NPROPS,ENTRY,VAPOR,ERROR)

IF VAPOR GO TO 225

CALL CB1(3,D,RH08)

GO TO 225

225 CALL CB1(4,PROPS(7),SM8)

NP7=5

225 CALL FLUIDL(PROPS,NP,ERROR)

CONTINUE

CALL CB1(6,PROPS(7),XM08)

H08M=H08

CALL CM1(5,H8,H8M)

PROPS(3)=H8M

ENTRY=5

NPROPS=7

CALL FLUID(TEMP,PM6,D,PROPS,NPROPS,ENTRY,VAPOR,ERROR)

IF VAPOR GO TO 690

260 CALL CB1(3,D,RH08)

CALL CB1(4,PROPS(7),SM8)

GO TO 695

690 CALL VAPR(1,XVAL6,S8,RH08,VOID6)

NP7=1

CALL FLUIDL(PROPS,NP,ERROR)

695 CONTINUE

CALL CB1(6,PROPS(7),XM08)

H08M=H08

CALL CM1(5,H8,H8M)

PROPS(3)=H8M

ENTRY=5

NPROPS=7

CALL FLUID(TEMP,PM6,D,PROPS,NPROPS,ENTRY,VAPOR,ERROR)

IF VAPOR GO TO 230

695 IF COUNT=ICOUNT GO TO 240

695 IF(XM0D-MO0T11)242,242,243

240 WRITE(IPRT,72) ICOUNT

C

END ITERATION MOOT11
C *** INITIALIZE FOR ITERATION ON T7

C

235 CONTINUE
   TOLER=0.05
   H6=1.0
   H4=0.01
   JCOUNT=1

C 250 CONTINUE
   H7=(H4+H6)*2.0
   CALL CM(2,P7*P7M)
   CALL CM(3,H4+H7M)
   PROPS(3)=H7M
   NPROPS=7
   ENTRY=5
   CALL FLUID(T7M,P7M,PROPS,NPROPS,ENTRY,VP7,ERROR)
   IF ERROR,NE,0 ) WRITE(IPRT,555)ERROR
   CALL CB1(1,T7M,T7)
   IF(EP7) GO TO 8000
   CALL CB1(3,D,RH07)
   CALL CB1(4,PROPS(2),S7)
   X7=PROPS(7)

C 360 CONTINUE
   CALL CB(6,X7,XMU7)
   MU0.5=(XMU6+XMU7)
   CALL MAGQ(QSIN,WIN61,0)
   MDOT7=(RH06+RH07)*QSIN/2.0
   H7NEW2=(H6+H7)*MDOT7+WIN67/178.16)
   IF (ABS(H7NEW-H7)=TOLER) 400,400,370

C 370 JCOUNT=JCOUNT+1
   IF (JCOUNT.GE.100) GO TO 700

C 320 CONTINUE
   H7M=H7NEW
   GO TO 310

C 330 CONTINUE
   HLM=H7M
   GO TO 250

C 8000 CALL VPAP(1,XVAL7,S7,B7,RH07,VOID7)
   NP7
   CALL FLUIDL(PROPS,NPERROR)
   X7=PROPS(7)
   GO TO 360

C 700 WRITE(IPRT,72) JCOUNT

C 400 H7M=H7NEW
   CALL FLUIDITEMP,PSM,PROPS,NPROPS,ENTRY,VAPOR,ERROR
   IF ERROR,NE,0 ) WRITE(IPRT,555)ERROR
   ALL CB1,TEMP,T5)
   IF VAPOR) GO TO 2000
   CALL VPAP(1,XVAL55H5,RH05,VOID5)

C 390 CONTINUE

C 2000 CALL VPAP(1,XVAL55H5,RH05,VOID5)

C 390 CONTINUE

C 2000 CALL VPAP(1,XVAL55H5,RH05,VOID5)

C 390 CONTINUE

C 2000 CALL VPAP(1,XVAL55H5,RH05,VOID5)

C 390 CONTINUE

C 2000 CALL VPAP(1,XVAL55H5,RH05,VOID5)

C 390 CONTINUE

C 2000 CALL VPAP(1,XVAL55H5,RH05,VOID5)

C 390 CONTINUE

C 2000 CALL VPAP(1,XVAL55H5,RH05,VOID5)

C 390 CONTINUE

C 2000 CALL VPAP(1,XVAL55H5,RH05,VOID5)

C 390 CONTINUE

C 2000 CALL VPAP(1,XVAL55H5,RH05,VOID5)

C 390 CONTINUE

C 2000 CALL VPAP(1,XVAL55H5,RH05,VOID5)

C 390 CONTINUE

C 2000 CALL VPAP(1,XVAL55H5,RH05,VOID5)

C 390 CONTINUE

C 2000 CALL VPAP(1,XVAL55H5,RH05,VOID5)

C 390 CONTINUE

C 2000 CALL VPAP(1,XVAL55H5,RH05,VOID5)

C 390 CONTINUE

C 2000 CALL VPAP(1,XVAL55H5,RH05,VOID5)

C 390 CONTINUE

C 2000 CALL VPAP(1,XVAL55H5,RH05,VOID5)

C 390 CONTINUE

C 2000 CALL VPAP(1,XVAL55H5,RH05,VOID5)

C 390 CONTINUE

C 2000 CALL VPAP(1,XVAL55H5,RH05,VOID5)

C 390 CONTINUE

C 2000 CALL VPAP(1,XVAL55H5,RH05,VOID5)

C 390 CONTINUE

C 2000 CALL VPAP(1,XVAL55H5,RH05,VOID5)

C 390 CONTINUE

C 2000 CALL VPAP(1,XVAL55H5,RH05,VOID5)

C 390 CONTINUE

C 2000 CALL VPAP(1,XVAL55H5,RH05,VOID5)

C 390 CONTINUE

C 2000 CALL VPAP(1,XVAL55H5,RH05,VOID5)

C 390 CONTINUE

C 2000 CALL VPAP(1,XVAL55H5,RH05,VOID5)

C 390 CONTINUE

C 2000 CALL VPAP(1,XVAL55H5,RH05,VOID5)

C 390 CONTINUE

C 2000 CALL VPAP(1,XVAL55H5,RH05,VOID5)

C 390 CONTINUE

C 2000 CALL VPAP(1,XVAL55H5,RH05,VOID5)

C 390 CONTINUE

C 2000 CALL VPAP(1,XVAL55H5,RH05,VOID5)
CALL MAGQIQSIN,WIN61,fl
GPM1 = IHOOT80448,831/R~04
~GPM1(MAGQIQSIN+W1N67,1)~
GPM8(MAGQIQSIN+W1N67,1)~
GPM5(MAGQIQSIN+W1N67,1)~
GPM7(MAGQIQSIN+W1N67,1)~
GPM0(MAGQIQSIN+W1N67,1)~
GPM2(MAGQIQSIN+W1N67,1)~
GPM3(MAGQIQSIN+W1N67,1)~
GPM4(MAGQIQSIN+W1N67,1)~
WRITE (IPRT,6200) MDOT5,GPM5,MDO~3,MDO~7,GPM7,
WRITE (IPRT,6500) MDOT8,GPM8,MDO~5,MDO~1,GPM1,
TEST=144*MDOT4*(P6-P4Y/(RHO6*WIN67))
WRITE (IPRT,777) TEST,ETA-OVP
WRITE (IPRT,666) P6,T6,RHO6,H6,S6
WRITE (IPRT,666) P7,T7,RHO7,H7,S7
CALL CM2,P9,P9M,PROPS,NPROPS,ENTRY,VAPOR,ERROR
IF (ERROR NE 0) WRITE (IPRT,555) ERROR
CALL CB(TEMP,T9) IF (VAPOR) GO TO 2050
CALL CB(1,PROPS(2),S9)
GO TO 2030
WRITE (IPRT,666) P8,T8,RHO8,H8,S8
IF (TEMP NE T9) WRITE (IPRT,777) TEST,ETA-OVP
CALCULATION OF PROPERTIES AT 9
CALL CM2,P9,P9M,PROPS,NPROPS,ENTRY,VAPOR,ERROR
IF (ERROR NE 0) WRITE (IPRT,555) ERROR
CALL CB(TEMP,T9)
IF (VAPOR) GO TO 2050
CALL CB(ENTRY=2,PROPS=3,NPROPS=3)
WRITE (IPRT,666) P9,T9,RHO9,H9,S9
WRITE (IPRT,666) P10,T10,RHO10,H10,S10
IF (TEMP NE T9) WRITE (IPRT,777) TEST,ETA-OVP
CALCULATION OF PROPERTIES AT 10
CALL FLUID(TEMP,P9M,CM2,PROPS,NPROPS,ENTRY,VAPOR,ERROR)
IF (ERROR NE 0) WRITE (IPRT,555) ERROR
CALL CB(ENTRY=2,PROPS=3,NPROPS=3)
WRITE (IPRT,666) P9,T9,RHO9,H9,S9
CALCULATION OF PROPERTIES AT 11
CALL CM2,P9,P9M,PROPS,NPROPS,ENTRY,VAPOR,ERROR
IF (ERROR NE 0) WRITE (IPRT,555) ERROR
CALL CB(ENTRY=2,PROPS=3,NPROPS=3)
WRITE (IPRT,666) P9,T9,RHO9,H9,S9
CALCULATION OF PROPERTIES AT 12
CALL CM2,P9,P9M,PROPS,NPROPS,ENTRY,VAPOR,ERROR
IF (ERROR NE 0) WRITE (IPRT,555) ERROR
CALL CB(ENTRY=2,PROPS=3,NPROPS=3)
WRITE (IPRT,666) P9,T9,RHO9,H9,S9
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SUBROUTINE VAPR(XVAL,S,H,RHO,VOID)

FOR TWO PHASE FLOWS CALCULATES MASS FRACTION XVAL AND VOLUME FRACTION VOID

C

DIMENSION VAL(10)

1 100 CONTINUE

SUBROUTINE CM(N,BIN,OBT)

CONVERT TO METRIC UNITS

N3 FOR TEMPERATURE N5 FOR DENSITY

N4 FOR ENTROPY N6 FOR ENTHALPY

DIMENSION VAL(10)

1 100 CONTINUE

SUBROUTINE CB(N,CIN,OUT)

CONVERT TO BRITISH ENG. UNITS

N3 FOR TEMPERATURE N5 FOR DENSITY

N4 FOR ENTROPY N6 FOR ENTHALPY

DIMENSION VAL(10)

1 100 CONTINUE

SUBROUTINE VAPR(N,XVAL,S,H,RHO,VOID)

CALCULATES MASS FRACTION XVAL AND VOLUME FRACTION VOID

C

DIMENSION VAL(10)

1 100 CONTINUE

SUBROUTINE FOR TWO PHASE FLUID

C

DIMENSION VAL(10)

1 100 CONTINUE

SUBROUTINE CM(N,BIN,OBT)

CONVERT TO METRIC UNITS

N3 FOR TEMPERATURE N5 FOR DENSITY

N4 FOR ENTROPY N6 FOR ENTHALPY

DIMENSION VAL(10)

1 100 CONTINUE

SUBROUTINE CB(N,CIN,OUT)

CONVERT TO BRITISH ENG. UNITS

N3 FOR TEMPERATURE N5 FOR DENSITY

N4 FOR ENTROPY N6 FOR ENTHALPY

DIMENSION VAL(10)

1 100 CONTINUE

SUBROUTINE VAPR(N,XVAL,S,H,RHO,VOID)

CALCULATES MASS FRACTION XVAL AND VOLUME FRACTION VOID

C

DIMENSION VAL(10)

1 100 CONTINUE

SUBROUTINE FOR TWO PHASE FLUID

C

DIMENSION VAL(10)

1 100 CONTINUE

SUBROUTINE CM(N,BIN,OBT)

CONVERT TO METRIC UNITS

N3 FOR TEMPERATURE N5 FOR DENSITY

N4 FOR ENTROPY N6 FOR ENTHALPY

DIMENSION VAL(10)

1 100 CONTINUE

SUBROUTINE CB(N,CIN,OUT)

CONVERT TO BRITISH ENG. UNITS

N3 FOR TEMPERATURE N5 FOR DENSITY

N4 FOR ENTROPY N6 FOR ENTHALPY

DIMENSION VAL(10)

1 100 CONTINUE

SUBROUTINE VAPR(N,XVAL,S,H,RHO,VOID)

CALCULATES MASS FRACTION XVAL AND VOLUME FRACTION VOID

C
SUBROUTINE MAGQ (QSIN, WIN67, IFLAG)

REAL MU, MQDOT, MALPHA
DIMENSION WDOT(200), MALPHA(200)
COMMON (FLUOC, GAMMA, WL, WG, DENS, ENTL, ENTG, ENTHL, ENTHG)

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CALCULATES DENSITY OF THE MIXTURE

100 FORMAT (5X, 10E5, 2X, 5E6)
110 FORMAT (5X, 10E5, 2X, 5E6)
120 FORMAT (5X, 10E5, 2X, 5E6)
130 FORMAT (5X, 10E5, 2X, 5E6)

C c

FORMAT (1X, "REAL MU, MQDOT, MALPHA")
100 FORMAT (5X, 10E5, 2X, 5E6)
110 FORMAT (5X, 10E5, 2X, 5E6)
120 FORMAT (5X, 10E5, 2X, 5E6)
130 FORMAT (5X, 10E5, 2X, 5E6)

C c

FORMAT (1X, "DIMENSION WDOT(200), MALPHA(200)")
100 FORMAT (5X, 10E5, 2X, 5E6)
110 FORMAT (5X, 10E5, 2X, 5E6)
120 FORMAT (5X, 10E5, 2X, 5E6)
130 FORMAT (5X, 10E5, 2X, 5E6)

C c

FORMAT (1X, "COMMON (FLUOC, GAMMA, WL, WG, DENS, ENTL, ENTG, ENTHL, ENTHG")
100 FORMAT (5X, 10E5, 2X, 5E6)
110 FORMAT (5X, 10E5, 2X, 5E6)
120 FORMAT (5X, 10E5, 2X, 5E6)
130 FORMAT (5X, 10E5, 2X, 5E6)

C c

FORMAT (1X, "SUNSTRAND CORPORATION VER 10.0 09/06/79 PAGE 9")
100 FORMAT (5X, 10E5, 2X, 5E6)
110 FORMAT (5X, 10E5, 2X, 5E6)
120 FORMAT (5X, 10E5, 2X, 5E6)
130 FORMAT (5X, 10E5, 2X, 5E6)
Fig. 5.3-FLUID PROPERTIES OF LIQUID HYDROGEN ESTIMATED IN EACH FLOW PATH
**Fig. 5.4**  FLUID PROPERTIES OF LIQUID OXYGEN ESTIMATED IN EACH FLOW PATH
6. VANE STAGE DESIGN
6.1 GENERAL MECHANICAL FEATURES

A vane pump generally consists of a rotor, shaft, vanes, a liner, 2 port plates, bearings and housing. Pumping is accomplished both by the undervane and the overvane swept volume pumping action.

The rotor can either be keyed to the shaft or driven by splines.

The vanes are rounded at both ends with a tip radius less than the minimum radius of curvature of the liner cam profile.

A cam profile is machined on to the liner or cam ring.

Flow paths for undervane kidneys, main inlet and discharge ports are on the port plates.

Bearings are placed as close to the rotor as possible in order to minimize the amplitude of shaft whirl at critical speeds. Tighter clearances can be held if the displacement amplitude is small.

Housings have a thermal expansion coefficient similar to that of the liner in order to reduce the sealing problem. Same thing holds for the shaft and rotor for limiting the leakage and stress.

6.2 SIZING

The total displacement of the vane pump is calculated by the following equation;

$$\text{Displacement} = 2 \pi L (R^2 - r^2)$$

where $L = \text{width of the vane}$

$R = \text{major radius of the cam ring}$

$r = \text{minor radius of the cam ring}$

The value of $R$ is limited by the maximum allowable surface rubbing speed of the vane tips. In the case of LH₂, this is 14.3 m/sec, LOX, it is 6.1 m/sec. The other dimensions of the pump are governed by such aspect ratios as rotor length/diameter, vane height/width, max. vane throw/height. The allowable ranges of these ratios are based on the previous experience in vane fuel pump design (Fig. 6.2.2). The basic information of the vane pump designed is shown in Fig. 6.2.3.

The cam contour is generated by two computer programs - 7th degree Polynomial and Trapezoidal. The acceleration curve of the Trapezoidal design consists of a constant and a sinusoidal acceleration curve. Both the 7th degree polynomial and the trapezoidal can be matched with a dwell depending on the circumstances. A trade-off between the two profiles was made and the 7th degree polynomial was chosen. The cam profile is selected on the basis of low "jerk*", minimum and maximum radial accelerations of the cam.

*jerk - rate of change of acceleration (m./sec³)
Generally we try to hold the jerk below $10^7 \text{m/s}^3$, maximum acceleration, no higher than $16764 \text{m/s}^2$ and minimum acceleration no lower than $6705 \text{m/s}^2$ (at 15000 rpm). A smooth contour will give a low jerk value. The minimum acceleration determines the lowest limit that a vane can remain in contact with the liner. The maximum acceleration dictates the boundary where wear and stress failure may occur.

The original design goals for the LH₂ pump set the operating speed at 12,000 rpm, and LO₂ pump at 4,000 rpm. As a result of the restrictions listed above, the optimum speed for LH₂ has to be 8000 rpm. If the same cam profile is to be used for LO₂, the width of the pump has to be reduced to obtain the target displacement per minute, but this will place the design outside the conservative design limit set by the aspect ratios, and at 4000 rpm the rubbing speed will be at the boundary value of 6.1 m/sec.

The operating speed of LO₂ Pump is then chosen to be at 3000 rpm to avoid these problems. Plots for the displacement, velocity, acceleration, jerk and the cam contour are shown in Figures 6.2.4 to 6.2.8. The computer print out is shown in Fig. 6.2.9.

The next step in evaluating the cam design is to study the vane dynamics. A computer program is available for this purpose. The program takes into consideration:

- the pressure forces on the vane,
- centrifugal force on vane,
- centrifugal force on the undervane fluid,
- friction force in the vane guide slot due to the Coriolis component of acceleration,
- the friction force in the vane guide slot due to the tip friction load on the vane,
- the friction force in the vane guide slot due to pressure side load,

and calculates the vane/liner reaction force. This vane/liner reaction force is used to evaluate the cam design. A zero or very small value dictates that the vane may leave the cam surface at that location. The maximum value of the force is used to calculate the Hertz contact stress between the vane and the liner. For fuel pumps, good experience falls below $8.27 \times 10^8 \text{N/m}^2$. Our design is only $1.79 \times 10^8 \text{N/m}^2$ for the LH₂ Pump.

A plot showing the magnitude of forces listed above through a complete rise & fall cycle is shown in Fig. 6.2.10.

6.3 LEAKAGE AND CARRYOVER VOLUME

The performance of a vane pump is greatly dependent on its volumetric efficiency which in turn is affected by the following factors:

I. Leakages through the clearances at

A. Vane Tips (minimized by hydraulic force balance and centrifugal force on vanes).
B. Vane Sides (controlled by clearance between vane and port plate).
C. Rotor Side (controlled by clearances between rotor and port plates).
D. High pressure undervane kidney slot leakage (controlled by rotor/ port plate clearance).
1. To inlet kidney slot.

2. To the center of the pump in to spline area.

E. Clearance between vane slot and vane
The leakage paths are shown in Figure 6.3.1.

II. Carryover volume

A. At the bottom of vane slot when the vane is at its minimum rise.

B. Between the rotor O.D. and liner minor radius.

Some designs have a circular cross-hole at the bottom of the slot, but in this design, since the fuel is going to be contamination free, the cross-hole is eliminated to provide a minimum carryover volume.

When the fluid is incompressible, carryover volume at the cross-hole is insignificant, but it has a very adverse effect on the volumetric efficiency when the fluid is as compressible as LH₂.

In order to reduce the leakage, 16 vanes are used so there will always be at least 2 vanes sealing the low pressure side from the high pressure side. In other words, at any instant, the leakages across the vane tip have to leak around two obstacles before they can see low pressure. The leakages controlled by the liner/port plate interface are minimized by a liner pressure plate. This approach is to spring load the plate against the pump liner. The critical operating temperature and operating clearances are selected. This allows the pump to be basically designed to have zero clearance between the rotor/port plate interface and the port plate/liner interface at a pre-selected condition. When the rotor width is less than the liner width caused by thermal, manufacturing or design constraints a clearance exists between the rotor and port plate. As the rotor increases in width from operational or thermal considerations, the rotor clearance with the port plate decreases until rubbing contact and zero clearance is achieved. Should the rotor continue to expand during operation, it will lift the pressure plate off of the liner interface at a pre-determined load consistent with the pump speed, differential pressure, material bearing compatibility and other design considerations. This design thus allows a very close fixed clearance operating condition.

6.4 VANE STAGE PERFORMANCE PREDICTION & BOOST STAGE MATCHING

The major leakage of the vane pump is contributed by the undervane discharge kidney to the shaft key area. This leakage decreases as the length of the leakage path increases. This dimension is restricted by the rotor diameter and the height of the vane. All the parameters involved have to weigh against one another with the aid of the cam contour, vane dynamics and thermal leakage program. Using the thermal leakage program, plots of volume flow rate (Q), volumetric efficiency (Nᵥ), amount of subcooling from saturation temperature (ΔT) at the vane pump entry point vs. inlet pressure (P₄) were generated over a range of side clearances for the purpose of optimizing the design. This was done on both fluids (Figure 6.4.1 to 6.4.6). These plots are the tools with which the operating point, performance and inlet pressure requirement are determined. The procedure for selecting the operating condition is as follows:
First, the amount of subcooling desired to enter the vane stage was estimated. From Fig. 6.4.3 (ΔT vs. P in) clearances and inlet pressure were found and plotted on the flow vs. inlet pressure curve Figure 7, and also plotted on the efficiency curve Figure 6.4.1. With the desired flow rate known, the clearance and inlet pressure were fixed. These parameters were used to obtain the volumetric efficiency of the pump.

For liquid hydrogen, an inlet pressure of 75838 to 89627 n/m² is sufficient to enable the vane pump to operate with a low percentage of vapor in the leakage path and with all liquid at the entry point of the vane pump. A two-stage boost pump is required to achieve this pressure rise. At this design point the LH₂ pump will operate with a wide margin of subcooling while maintaining a reasonable volumetric efficiency. For LH₂ fluid, it is possible to tolerate a small percentage of vapor in the system without creating major problems. This avoids using a higher number of boost stages and sacrificing volumetric efficiency. With a single boost stage using the same impeller at 3000 rpm, a pressure rise of 782733 n/m² will be generated for LOX. This will provide 100% liquid throughout the entire pump package. This design feature is important because only oxygen in vapor phase can support combustion. Without O₂ vapor in the system, no fire can occur unless there is a sudden localized temperature rise occurring somewhere along the rubbing surface and the liquid gains enough energy to flash into vapor to support ignition.

6.5 PORT TIMING

The main inlet and discharge ports on the cam ring are situated at the middle of the cam rise and fall. In order to improve sealing, a 45° arc is left between the inlet and discharge main ports. This configuration insures two vanes sealing at any instant. The undervane kidneys on the port plates perform two functions:

1. They provide flow passages for the undervane inlet and discharge.

2. They pressurize the vane from underneath. This reduces leakage across the vane tips.

Where the over-vane pressure is low, the undervane pressure is designed to be low. This keeps the contact stress between the vane tip and the liner to a low value. The extend of the discharge undervane kidney is designed so that the vane sees high pressure underneath 10° before it sees high pressure over-vane, and the high undervane pressure is also maintained until the vane is 10° past the end of the high pressure zone. This assures good sealing between vane tip and liner.
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*Fig. 6.2.2 Vane Stage Design Constraints & Aspect Ratios*
# Vané Stage Parameters

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**FIGURE 6.2.3**
Fig. 6.2.4  Cam Contour Design - Cam Rise vs. angular rotation
Fig. 6.2.5 Cam Contour Design - Cam Radial Velocity vs. Angular Rotation
MODEL = 19B 8000 RPM .05 IN THROW TITANIUM CARBIDE
(1.27 CM)

7TH POLYNOMIAL

Fig. 6.2.6 Cam Contour Design - Cam Radial Acceleration vs. Angular Rotation
Fig. 6.2.7 Cam Contour Design - Cam Jerk vs. Angular Rotation
6.2.8 Intentionally left blank.
Fig. 6.2.10 Vane/Liner Reaction (Force 1)
Fig. 6.3.1 Leakage Paths in Vane Stage
Fig. 6.4.1  VANE STAGE PERFORMANCE (LH₂) -
EFFECT OF INLET PRESSURE ON DISCHARGE FLOW
Fig. 6.4.2  
VANE STAGE PERFORMANCE (LH₂)  
EFFECT OF INLET PRESSURE ON PUMP VOLUMETRIC EFFICIENCY
Fig. 6.4.3

VANE STAGE PERFORMANCE (LH₂)

SUBCOOLING EFFECT OF INLET PRESSURE ON LH₂
Fig. 6.4.4
VANE STAGE PERFORMANCE (LOX)
- EFFECT OF INLET PRESSURE ON DISCHARGE FLOW
VANE STAGE PERFORMANCE (LOX) - SUBCOOLING EFFECT OF INLET PRESSURE ON LOX

Fig. 6.4.6

\( \frac{N}{m^2 \times 10^4} \)
7. BOOST STAGE DESIGN
The fluid dynamic design of the two-stage centrifugal pump for boiling liquid hydrogen is based on the following design specification:

Volume flow rate: 7.5 gpm (28.39 l/min)
Rotational speed: 8000 rpm
Pump pressure rise: 13.0 psi (89.627 N/M²)
Liquid temperature: 37.0 deg R (20.2 °K)
Pump efficiency: 0.35 to .040

In the above specification, the pump pressure rise is equally divided between the two stages, and the hydrogen at 37 deg R is saturated liquid with 10 percent vapor (by volume) upstream from the pump inlet. The pump efficiency includes recirculation and disk friction losses. The density of the liquid hydrogen has been assumed constant at 4.404 lb/ft³ (692 N/m³)

\[ H = \frac{144 \cdot \Delta P}{2g} = 212.5 \text{ ft} = 64.77 \text{ m} \]

For the above design specification values, the stage specific speed \( N_s \) becomes

\[ N_s = \frac{N \cdot Q}{H^{3/4}} = 393.6 \]

\[ N_s = \frac{W \cdot Q^{1/2}}{g \cdot H^{3/4}} = 1.116 \]

This low value for \( N_s \) is responsible for the relatively small specified values of pump efficiency, and has been a major factor determining the final pump geometry, particularly the relatively large impeller diameters and the highly backward curved impeller blades.
September 24, 1979
775-L-794080

NASA, Lewis Research Center
21000 Brookpark Road
Cleveland, Ohio 44135

Attention: Mr. L. E. Light
Head, Space Systems Section
MS500-213

Subject: Final Engineering Report No. CR159648
Liquid Oxygen/Liquid Hydrogen Boost/Vane Pump for the Advanced Orbit Transfer Vehicle Auxiliary Propulsion System

Reference: Contract No. NAS3-20401

Gentlemen:

Please find enclosed a copy of the subject report. Distribution has been made in accordance to the instructions contained in your letter no. 1434(IS) dated 8-16-79.

Sincerely,

SUNDSTRAND AVIATION OPERATIONS
Advanced Technology Group
Sundstrand Corporation

Richard Alms
Data Administrator

For: S. Tamborello
Contract Administrator

ST/RA:aj
Enclosures
7.2.1 Friction Losses

For the above low value of specific speed, friction losses are a dominant factor affecting pump efficiency, both internally and in terms of disk friction. Fortunately, the kinematic viscosity of liquid hydrogen at 20.2 K is about $2.03 \times 10^{-7}$ m$^2$/sec (reference 1), compared with about $9.29 \times 10^{-7}$ m$^2$/sec for water at 23.9°C. Thus, for comparable velocities and sizes of pump, the Reynolds number is five times larger for liquid hydrogen than for water. This difference in Reynolds number ameliorates the adverse effects of friction at low specific speeds and permits, for example, larger impeller tip diameters and larger $l/d$ ratios for the channel between impeller blades than might otherwise be used.

7.2.2 V/L Ingestion

A second problem area having a major impact on the pump geometry is the 10 percent vapor (by volume) upstream from the pump inlet. Normal design procedure at low NPSH calls for a separate axial-flow inducer to increase the impeller inlet head so that the suction specific speed of the impeller is reduced below about 7500. For this liquid hydrogen pump with 10 percent vapor at inlet, sufficient inducer head must be developed to: (1) condense the vapor, and (2) raise the impeller NPSH. Here the head required to condense the vapor was three times larger than that required to raise the NPSH. To achieve the inducer head required under the above conditions, it was necessary to use a mixed-flow inducer geometry.

7.2.3 Dynamic Head at Impeller Discharge

The third problem area having a major impact on the pump geometry is the high absolute velocity (kinetic energy) of the liquid hydrogen at the
impeller discharge. This high absolute velocity \( \mathbf{u} \) occurs in spite of highly backward curved blades \( (\text{large } \beta) \), because the relative velocities \( \mathbf{u} \) in the impeller have been kept low to minimize internal friction losses. Thus,

\[
\begin{align*}
\mathbf{u} & \quad \text{(absolute velocity)} \\
\mathbf{u} & \quad \text{(relative velocity)} \\
\mathbf{u} & \quad \text{(velocity component)}
\end{align*}
\]

It is clear that special care must be taken to convert this kinetic energy to static pressure rise, and so the diffuser design consists in a multiplicity of optimum conical diffusers.

7.3 **Boost Stage Design**

7.3.1 **First stage inducer** - The inducer inlet area was sized conservatively to handle the liquid hydrogen plus twice the 10 percent volume occupied by the vapor. For a hub-tip radius ratio \( \xi_H \) of 0.5, and an inlet relative flow angle \( \beta_{I,T} \) at the leading-edge tip of 86.00 degs, simple continuity gives

\[
\xi_H = 0.5
\]

\[
\beta_{I,T} = 86.00 \text{ degs}
\]

\[
\tau_{I,T} = 0.6307 \text{ ins} = 1.6 \text{ cm}
\]

\[
\tau_{I,H} = 0.3154 \text{ ins} = 0.8 \text{ cm}
\]

The blade angle \( \beta_{I,T}^* \) at the leading edge tip was set at 82.92 degs, thus providing an incidence angle \( \hat{\beta}_{I,T} \) of 3.08 degs to accommodate the vapor cavity, which is attached to the leading edge and lies along the suction surface, and to accommodate to a much lesser degree the blade blockage of the relatively sharp leading edges. Thus,

\[
\beta_{I,T}^* = 82.92 \text{ degs}
\]

\[
\hat{\beta}_{I,T} = \beta_{I,T} - \beta_{I,T}^* = 3.08 \text{ degs}
\]
The leading-edge profile has a 5-deg wedge angle, with all material removed from the suction surface, and a nominal nose radius of .0025 cm ins; otherwise, the inducer blades have a constant thickness of .51 cm along the shroud and .076 cm along the hub.

The inducer head required to condense the 10 percent vapor (by volume) was estimated from the velocity head required to generate 10 percent vapor when generated from saturated liquid hydrogen at static conditions, as given in reference 2. This head $H_{I,1}$ is 4.83 m. The additional inducer head $H_{I,2}$, required to obtain a suction specific speed of 6000 for the first-stage impeller, is 1.71 m. Thus, the required inducer head $H_I$ becomes

$$H_I = H_{I,1} + H_{I,2} = 15.87 + 5.62 = 21.49 \text{ ft.} = 6.55 \text{ m}$$

A design value of 25 feet was used, and it was assumed that the hydraulic efficiency $n_{I,\text{HYD}}$ to produce this head is 50 percent. The resulting work input cannot, as discussed earlier, be achieved in this pump by an axial-flow inducer. However, for a mixed-flow configuration (where the average exit radius $r_{EX}$ from the inducer can be larger), based on continuity and assuming a slip factor $\mu_I$ of 0.75, the required work input is achieved at $r_{EX}$ equal to 2.235 cm with an exit vane height $h_{EX}$ of .51 cm when the product of the exit flow coefficient $C_{FL,EX}$ and the blade blockage factor $S_{EX}$ is 0.871, and when the exit blade angle $\beta_{EX}^*$ is equal to the leading-edge tip angle $\beta_{I,T}^*$. Thus, $H_I = 25.00 \text{ ft.} = 7.62 \text{ m}$

$$n_{I,\text{HYD}} = 0.50$$
$$\mu_I = 0.75$$
$$r_{EX} = 0.880 \text{ ins.} = 2.235 \text{ cm}$$
$$h_{EX} = 0.202 \text{ ins.} = .51 \text{ cm}$$

$$(C_{FL,EX})(S_{EX}) = 0.871$$

$$_{EX}^* = _{I,T}^* = 82.92 \text{ degs}$$
The meridional configuration of the inducer is shown in figure 7.3.1.1 or 7.3.1.2. The hub radius of curvature $r_h$ and the shroud radius of curvature $r_s$ are 2.706 and 2.667 cm respectively. The blade angle $\beta_1$ is a constant 82.92 degs along the shroud, and the mean blade surface is generated by straight line elements lying in meridional planes (constant $\theta$) and extending from shroud to hub at equal percentages of shroud and hub lengths.

For this configuration, the relative velocity ratio $W_{EX}/W_{I,T}$ across the inducer is 0.800, and the dwell time $t_D$ of the liquid hydrogen in the inducer is 0.0253 sec. The values of both parameters are excellent; the first, because deceleration of the flow is moderate so that separation should not occur on the suction surface; and the second, because sufficient time is available to achieve condensation of the hydrogen vapor before leaving the inducer.

A computerized quasi-three-dimensional analysis was made to determine the velocity distributions on the blade surfaces along the hub and shroud lines. The results for the 3-bladed inducer are shown in figure 7.3.1.3 (See NOTE on next page) It is noted that the velocity distribution along the suction surfaces at both hub and shroud are relatively constant or steadily increasing. Their types of velocity distribution are considered to be excellent because flow separation from the suction surface is precluded.

Along the pressure surface at the hub, the velocity becomes negative,

* A 2 bladed inducer will be used.
thus indicating a small reverse flow for a portion of that surface. Although negative velocities on the pressure surface are not desirable, neither are they especially harmful and in the actual pump may not in fact exist.

NOTE: A 3 bladed inducer was proposed at the time when this section was written. But it was later modified to a 2 bladed configuration. The hardware will conform to the 2 bladed structure.

For two blades, the solidity $\sigma_I$ based on the wrap angle of 533 degrees is 2.96. Thus,

\[ \gamma^*_I = \text{number of blades} = 2 \]
\[ \sigma^*_I = \text{blade solidity} = 2.96 \]

In conclusion, it should be noted that the relative velocities will be higher than shown in the figure, at least for the first half of inducer length, due to the presence of the vapor and vapor cavity. This vapor has been partially accounted for by specifying flow coefficients that increase linearly with station number from 0.833 at the leading-edge (station 1) to 0.925 at the inducer exit (station 16). However, the one-dimensional design value for the average inlet relative velocity at the inducer tip is 13.4m/sec. The inducer dimensions are tabulated in Fig. 7.3.1.

7.3.2 First Stage Impeller. The leading-edge of the first-stage impeller is nearly contiguous with the exit from the inducer. Thus, the mean leading-edge radius and annulus height are essentially the same. The impeller has 6 blades and the inlet blade angle $\beta^{*}_{I,LE}$ is 84.00 degrees (vs 82.92 degrees for the inducer exit angle). Thus,

\[ \gamma^*_I = \text{number of blades} = 6 \]
\[ \beta^{*}_{I,LE} = 84.00 \text{ degs} \]
A recirculation flow rate of 4 percent was assumed for both the first and second stage impellers.

To achieve the stage head $H$ of 64.7 m as given in the design specification, assuming a stage hydraulic efficiency $\eta_{\text{HYD}}$ of 52 percent, and with an exit blade angle $\beta^*_T$ of 78.29 degrees, continuity and the required work input give a tip radius $r_T$ of 4.62 cm and a tip blade height $h_T$ of 0.429 cm. The slip factor $\mu_T$, based on the method of Wiesner (ref. 5), is 0.871 and the impeller-tip flow-coefficient $C_{\text{FL}, T}$ is 0.925. Thus,

$$H = 212.5 \text{ ft.} = 64.77 \text{m}$$

$$\eta_{\text{HYD}} = 0.52$$

$$\beta^*_T = 78.29 \text{ degrs.}$$

$$r_T = 1.822 \text{ ins.} = 4.63 \text{cm}$$

$$h_T = 0.169 \text{ ins.} = 0.429 \text{cm}$$

$$\mu_T = 0.871$$

$$C_{\text{FL}, T} = 0.925$$

The blade angle $\beta^*$ varies linearly from 84.00 degrees at the leading edge to 78.29 degrees at the tip. The blade thickness $t_I$ is constant at 0.0508 cm and the meridional profile is shown in figure 7.3.1.1 (supplied by Sundstrand) or 7.3.1.2.

For this design, the stage head coefficient $\psi$ is 0.423 and the relative velocity ratio $W_{\text{ex}}/W_{\text{T1}}$ across the impeller is 0.825. Based on the method of Daily and Nece (ref. 4), the stage disk-friction power is 15 watts. Also, the useful hydraulic output power is 20.88 watts and the hydraulic input power $HP_{\text{IN}}$ is 0.0571, giving an estimated stage efficiency $\eta_p$ of 36.6 percent. Thus,
\[ \psi = 0.423 \]
\[ \frac{W_{EX}}{W_{TI}} = 0.825 \]
\[ HP_D = 0.0205 \]
\[ HP_{HYD} = 0.0284 \]
\[ HP_{IN} = 0.0571 \]
\[ \eta_p = 0.366 \]

Dimensions are tabulated in fig. 7.3.2

### 7.3.3 Second-Stage Impeller

For reasons to be discussed, the second stage impeller is assumed to have the same hydraulic efficiency (0.52) as the first-stage inducer and impeller combined. Thus, for the same tip radius \( r_t \), blade height \( h_t \) and blade exit angle \( \beta_T^* \), the flow conditions out of the second-stage impeller are the same as those of the first stage impeller. (Thus, the vaned diffuser design is the same for both stages.)

The second stage impeller does not require a separate inducer, thus, the impeller can have a radial inlet and a constant blade height \( 0.429 \text{cm} \) between parallel plates, as shown in figure 7.3.1.1 or 7.3.1.2. The leading-edge radius \( r_{2,LE} \) is 1.651 cm. the blade angle \( \beta_{2,LE}^* \) is 81.36 degrees, the inlet flow angle \( \beta_{2,LE} \) (including the effects of blade blockage and a flow coefficient \( C_{2,FL-LE} \) of 0.875) is 83.36 degrees. Thus, the effective incidence angle \( \alpha_{2,LE} \) is 2.00 degrees. This positive incidence is considered desirable, because of the relatively sharp turning upstream from the leading-edge in the meridional plane (see fig. 7.3.1.1) (Sundstrand). Thus,

\[ h_{2,LE} = 0.169 \text{ ins.} = 0.429 \text{ cm} \]
\[ r_{2,LE} = 0.650 \text{ ins.} = 1.651 \text{ cm} \]
\[ \alpha_{2,LE} = 81.36 \text{ degs} \]
\[ \beta_{2,\text{LE}} = 83.36 \text{ degs.} \]
\[ \alpha_{2,\text{LE}} = 2.00 \text{ degs} \]
\[ C_{2,\text{FL-LE}} = 0.875 \]

Dimensions are tabulated in fig. 7.3.3.

The relative velocity ratio \( W_{2,T}/W_{2,\text{LE}} \) across the impeller is 0.519, which value is considered marginally safe, and, for this reason, the hydraulic efficiency \( n_{\text{HYD}} \) is assumed to be equal (0.52) for both stages.

Computerized quasi-three-dimensional analyses to determine the relative velocity distributions on the blade surfaces along the hub and shroud lines were made for both the 1st and 2nd stage impellers. As for the 1st. stage inducer (fig. 7.3.1.3) the velocity distributions were found to be satisfactory.

Finally, it should be noted that the relatively large blade angles \( \beta^* \) used in from the second stage is not designed to create a static pressure rise, but merely to collect the flow at low velocity in order not to lose static pressure.

The performance curves for the boost stage(s) in liquid oxygen and liquid hydrogen are shown in Figures 7.4 and 7.5.
Fig. 7.3.1.1 - Meridional View of the LH Boost Stages
FIRST STAGE INDUCER

Type: Mixed - flow, 2 blades
Inlet flow angle $B_{I,T} = 86.0^\circ$
Inlet blade angle $B_{I,T}^* = 82.92^\circ$
Incidence angle $I_{T} = 3.08^\circ$
Leading-edge profile wedge angle = $5^\circ$
Nose radius = 0.001 in. = 0.00254cm
Blade thickness = 0.02 in. along shroud = 0.0508cm
= 0.03 in. at hub = 0.0762cm
Exit Radius = 0.88 in = 2.2352cm
Exit vane height = 0.202 in = 0.513
Exit blade angle = $B_{I,T}^* = 82.92^\circ$

Fig. 7.3.1 Inducer Dimensions
Fig. 73-13 Velocity distribution on blade surfaces of first-stage inducer at equi-distant stations along hub and shroud (stationary).
FIRST STAGE IMPELLER

Number of blade
Leading edge mean radius
Inlet blade angle \( B_{1,LE} \)
Exit blade angle \( B_T \)
Tip Radius \( T_T \)
Tip Blade height \( h_T \)
Blade Thickness \( t_I \)

\[
\begin{array}{ll}
\text{Number of blade} & 6 \\
\text{Leading edge mean radius} & 0.88 \text{ in} \quad (2.2352 \text{cm}) \\
\text{Inlet blade angle } B_{1,LE} & 84^\circ \\
\text{Exit blade angle } B_T & 78.29^\circ \\
\text{Tip Radius } T_T & 1.822 \text{ in} \quad (4.628 \text{cm}) \\
\text{Tip Blade height } h_T & 0.169 \text{ in} \quad (0.429 \text{cm}) \\
\end{array}
\]

Fig. 7.3.2 1ST STAGE IMPELLER DIMENSION

SECOND STAGE IMPELLER

Number of blade
Head edge mean radius \( T_{2,LE} \)
Inlet blade angle \( B_{2,LE} \)
Inlet flow angle \( B_{2,LE} \)
Effective incidence angle \( i_{2,LE} \)
Exit Blade angle \( B_T^* \)
Tip radius \( T \)
Tip blade height \( h_T \)
Blade thickness \( t \)

\[
\begin{array}{ll}
\text{Number of blade} & 6 \\
\text{Head edge mean radius } T_{2,LE} & 0.65'' \quad (1.651 \text{cm}) \\
\text{Inlet blade angle } B_{2,LE}^* & 81.36^\circ \\
\text{Inlet flow angle } B_{2,LE} & 83.36'' \\
\text{Effective incidence angle } i_{2,LE} & 2^\circ \\
\text{Exit Blade angle } B_T^* & 78.29^\circ \\
\text{Tip radius } T & 1.822 \text{ in} \quad (4.628 \text{cm}) \\
\text{Tip blade height } h_T & 0.169 \text{ in} \quad (0.429 \text{cm}) \\
\text{Blade thickness } t & 0.02 \text{ in} \quad (0.0508 \text{cm}) \\
\end{array}
\]

Fig. 7.3.3 2ND STAGE IMPELLER DIMENSION
Figure 7.4 - LOX Boost Stage Performance Curves
Figure 7.5 - LH$_2$ Boost Stages Performance Curves
8. BOOST/VANE PUMP

PERFORMANCE PREDICTION
HYDRAULIC SCHEMATIC
The Performance curves of the individual vane stage (LH₂) and boost stage (LH₂) are shown in figures 6.4.1, 6.4.2, 7.5. In order to estimate the performance of the integrated unit, \( \dot{Q} \) discharge vs. \( P_4 \) of the vane stage at various discharge pressures \( P_6 \) were generated and plotted on the same graph with the performance curve (\( \Delta P \) vs. \( \dot{Q} \)) of the boost stage (Figure 8.1). The performance curve (\( \Delta P \) vs. \( \dot{Q} \)) of the integrated boost/vane pump was then read off from the intersections of these two families of curves. The resulting LH₂ pump performance curve is shown in Figure 8.2 together with the \( H_p \) and \( n_p \) curves. The LOX pumps is shown in Figure 8.4.

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<table>
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<tr>
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<th>( n_{\text{vane}} )</th>
<th>( P_{\text{vane}} ) (PSIG)</th>
<th>( P_{\text{boost/vane}} ) (PSIG)</th>
<th>( n'_{\text{vane}} )</th>
<th>( n'_{\text{(boost \&amp; vane)}} )</th>
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<tr>
<td>3.17</td>
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<td>420.05</td>
<td>435.3</td>
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<td>.538</td>
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</tbody>
</table>
Fig. 8.2 - LH₂ Pump Performance
Fig. 8.3 - LOX Vane/Boost Stages Matching
Fig. 8.4 - LOX Pump Performance
9. MECHANICAL DESIGN

9.1 Stress on Vane Stage
9.2 Thrust Load on Impellers
9.3 Bearing Life & Seal Selection
9.4 Liner Pressure Plate
DESIGN PARAMETER LIMITATIONS

1. Adequate material properties at 20.20 K
2. Surface speeds less than 6.096 m/s for LOX pump
3. Zero NPSH operation
4. Eliminate vortex action at boost pump inlet
5. Gas inside the pump is to be avoided
6. Isolation of seal package from cryogenic temperatures
7. Material compatibility with liquid O₂; design for safety.
8. Material compatibility with Liquid H₂, design for wear.
9. Reprime capability
10. Same vane pump design to be used on both hydrogen and oxygen, only materials change.
11. Pressure plate on vane pump to eliminate catastrophic failures due to rotor or vane growth.
9.1 STRESSES
9.1 STRESSES ON VANE STAGE

BOOSTED VANE PUMPS

ROTOR KEY SIZE

HP @ 40% , HP = 2.25 \( (LH_2) \) = 2.28
= 1.47 \( (LOX) \) = 1.49

\[
T = \frac{63025 \text{ HP}}{N} \quad N = 8000 \text{ rpm} \quad \text{(LH}_2) \quad = 3000 \text{ rpm} \quad \text{(LOX)}
\]

\[
T = \frac{63025 \times 2.25}{3000} = 17.73 \text{ in. lb.} \quad \text{(LH}_2) = 20.4 \text{ cm. kg.}
\]

\[
= \frac{(63025 \times 1.47)}{3000} = 30.88 \text{ in. lb.} \quad \text{(LOX)} = 35.6 \text{ cm. kg.}
\]

Shaft Dia. \( D = .325 \text{ in.} = .7937 \text{ cm} \)

For LOX:

Key shear,

\[
= \frac{T}{.5b1D} = \frac{T}{bL} \quad b = \text{ key width} \quad L = \text{ Key length}
\]

Material,

for \( A1S1 \text{ 4130} \) (40-50 HRC)

\[
\sigma_{un} = 180,000 \text{ psi} \quad \sigma_{fy} = 93,000 \text{ psi}
\]

\[
= 1.24 \times 10^9 \text{ N/m}^2 \quad = 6.41 \times 10^8 \text{ N/m}^2
\]

\[
\sigma_{fy} = 163,000 \text{ psi} \quad \sigma_{ey} = 173,000 \text{ psi}
\]

\[
= 1.12 \times 10^9 \text{ N/m}^2 \quad = 1.19 \times 10^9 \text{ N/m}^2
\]

Let \( b = .125 \text{ in.} = .3175 \text{ cm} \)

\[
M.S. = .5
\]

\[ \pi = \text{ factor of safety} = M.S. + 1 \]

\[ = 1.5 \]

\[
L = \frac{T \times L}{1.5 \times .5}
\]

\[
= \frac{30.88 \times 1.5}{.5 \times (.125 \times .3125 \times 93000)} = .026 \text{ in.} = .066 \text{ cm}
\]

Use 1/8 in min. available key. (.3175 cm key)
Compression,

\[ \sigma_c = \frac{F}{.25tLD} \]

Let \( L = 1/4" \)

\[ \sigma_c = \frac{30.88}{.25 (.125) (.25) (.3125)} = 12,650 \text{ psi} = 8.72 \times 10^7 \text{ N/m}^2 \]

M.S. \( \rightarrow \) large

Shaft

\[ \tau = \frac{T r}{J} = \frac{T}{\frac{PD}{32}} = \frac{16 T}{\pi D^3} \]

\[ = \frac{16 (30.88)}{\pi (.3125)^3} = 5153 \text{ psi} = 3.55 \times 10^7 \text{ N/m}^2 \]

Stress concentration due to keyslot,

\( t = 3.0 \) (Ref. Peterson, 1974, Fig. 183)

Max. stress,

\[ \tau' = 5153 (3) = 15,460 \text{ psi} = 1.06 \times 10^8 \text{ N/m}^2 \]

\[ \sigma' = 27,122 \text{ psi} \quad \text{M.S.} \rightarrow \text{LARGE} \]

\[ = 1.87 \times 10^8 \text{ N/m}^2 \]

Vane

Material properties: \( \sigma_u = 280 \text{ ksi} = 1.93 \times 10^9 \text{ N/m}^2 \)

Material - FERRO-TIC

Vane dimensions,

\( t = .040 \text{ in.} = .102 \text{ cm} \)

\( a = .05 \text{ in (stroke)} = .128 \text{ cm} \)

\( w = .771 \text{ in} = 1.98 \text{ cm} \)

\( h = .2 \text{ in} = .514 \text{ cm} \)
for a beam of relatively
great width \( \left( \frac{W}{a} > 4 \right) \)
bending stress, (reflecting the load due to
friction between vane tip & liner)
\[
\sigma_b = \kappa_m \left( \frac{6P}{t^2} \right)
\]
for \( \frac{z}{a} = 0 = \frac{c}{a} = .5 \)

\[
\kappa_m = .370 \quad \text{(see pg. 135 of Ref, Book 7)}
\]
\[
P = (\Delta \text{ PSI}) \times (W) \times (a)
\]
\[
P = .317 \times (.771) \times (.05) = 12.2
\]
\[
\sigma_b = .37 \times \frac{6(12.2)}{(0.04)^2} = 16927 \text{ psi.} = 1.17 \times 10^8 \text{ N/m}^2
\]

M.S. is LARGE

Compressive stress at edge of vane slot;
The vane/slot combination is similar to a pin in a solid body. The vane is
assumed to be infinitely stiff and the rotor material linearly elastic.
The maximum pressure at the edge of the slot is,

\[
p_{\text{max}} = \left[ \frac{P}{h-a} \left( 4 + \frac{6a}{h-a} \right) \right] / W
\]
\[
= \frac{13}{.15(.771)} \left[ 4 + \frac{6}{.05} \right]
\]
\[
= 674 \text{ psi} = 4.65 \times 10^6 \text{ N/m}^2
\]

M.S. \rightarrow \text{LARGE}

Bending stress from vane load on
the rotor
\[
\sigma_b = \kappa_m \left( \frac{6P}{t^2} \right) \quad \kappa_m = .51 \quad \text{for } \frac{c}{a}
\]
\[
= 1
\]
\[ P = R_A = 15.46 \]

\[ \sigma_d = \frac{0.51(b)(15.46)}{(0.1066)^2} = 4163 \text{ psi} = 2.87 \times 10^7 \text{ N/m}^2 \]

Stress concentration factor for bending

\[ K_t = 1.55 \text{ for } \frac{r}{d} = \frac{r}{t} = 0.020 = 0.188 \]

\[ \frac{D}{d} = \frac{3t}{t} = 3 \]

(ref. Peterson, '74, fig. 73)

\[ \sigma' = 1.55 \times 4163 = 6453 \text{ psi} = 4.45 \times 10^7 \text{ N/m}^2 \]

Ring section stress (neglecting vane load)

at bore, \( \sigma_{eff} = 189 \text{ psi} \) (ref. E4 output)

\[ = 1.3 \times 10^6 \text{ N/m}^2 \]

M.S. \( \rightarrow \) large.

Input and output are shown in Fig. 9.1.1 and 9.1.2
Rotor & Vane Material - Ferrotic

2 Pump Designs with same dimensions
Hydrogen - HT6
Oxygen - eNS

Rotor O.D. = 1.0468" = 2.66 cm
Rotor width = .771" = 1.96 cm
No. of Vanes = 16
Dual Lobe
Inlet and discharge ports 45°
Vane Height = .200" (0.508 cm)
Vane Thickness = .040" (.1016 cm)
Shaft Diam. = .3125 (.7937 cm)

<table>
<thead>
<tr>
<th></th>
<th>LH₂</th>
<th>LOX</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Inlet</strong></td>
<td>28 PSIAY</td>
<td>21 PSIAY</td>
</tr>
<tr>
<td></td>
<td>1.93 x 10⁵ N/m²</td>
<td>1.45 x 10⁵ N/m²</td>
</tr>
<tr>
<td><strong>Discharge</strong></td>
<td>231 PSIAY</td>
<td>338 PSIAY</td>
</tr>
<tr>
<td></td>
<td>1.59 x 10⁶ N/m²</td>
<td>2.33 x 10⁶ N/m²</td>
</tr>
<tr>
<td><strong>Flow</strong></td>
<td>7.0 GPM</td>
<td>3.0 GPM</td>
</tr>
<tr>
<td></td>
<td>26.49 l/min</td>
<td>11.35 l/min</td>
</tr>
<tr>
<td><strong>Speed, RPM</strong></td>
<td>8000</td>
<td>3000</td>
</tr>
<tr>
<td><strong>Hp @ 40% N</strong></td>
<td>2.25</td>
<td>1.47</td>
</tr>
<tr>
<td><strong>Operating Temp.</strong></td>
<td>37 °R 20°K</td>
<td>163 °R 90.2°K</td>
</tr>
</tbody>
</table>
9.1 Vane Port Sizing Based on Vane Contact Stress and Area Porting

Entrance Porting Area = \(0.45 \times 0.15 = 0.0675 \text{ in}^2/\text{port} = 0.454 \text{ cm}^2/\text{port}\)

For LH2 Pump 2 Ports For 7.5 GPM (28.39 L/min)

Velocity = \(\frac{7.5 \times 2.31}{1.0575 \times 120} = 17.8 \text{ ft/sec} = 5.425 \text{ m/sec}\)

Porting Arc of 45°

Mean Radius of Porting Arc = \(\frac{1}{2} (r_{\text{max}} + r_{\text{min}}) = \frac{1}{2} (1.054 + 1.156)\)
\(= 0.553'' = 1.404 \text{ cm}\)

Vane Thickness = 0.040'' = 0.1016 cm
Vane Radius Tip = 0.040'' = 0.1016 cm
Vane Height = 0.200'' = 0.508 cm
Vane Length = 0.771'' = 1.958 cm

CNS (LOX) \(\rho = 11.8 \text{ gr/cc} = 0.4263 \text{ lb/in}^3\)
HT6 (LH2) \(\rho = 6.60 \text{ gr/cc} = 0.2384 \text{ lb/in}^3\)

Vane Mass (LOX) = 0.04 \times 0.771 \times 0.2 \times 0.4263 = 2.629 \times 10^{-3} \text{ lb}
Vane Mass (LH2) = 0.04 \times 0.771 \times 0.2 \times 0.2384 = 1.478 \times 10^{-3} \text{ lb}

Radius of Vane Rotation = 0.553'' \times 0.5 = 0.2765'' = 0.702 cm

Centrifugal Force \(F_c = Mrw^2\)

Pressure Force \(F_p = \frac{1}{2} \Delta P \times \text{Area} = \frac{0.04 \times 0.771 \times \Delta P}{2}\)
\(= 0.0304 \Delta P/2 = 0.0152 \Delta P\)
**LOX**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>LOX</th>
<th>LHe</th>
</tr>
</thead>
<tbody>
<tr>
<td>Speed</td>
<td>3000</td>
<td>8000</td>
</tr>
<tr>
<td>( \omega ), rad/sec</td>
<td>314.2</td>
<td>837.2</td>
</tr>
<tr>
<td>( N ), lb-ft/ft</td>
<td>8.171 x 10^-5 (1.235 g)</td>
<td>4.569 x 10^-5 (0.794 m)</td>
</tr>
<tr>
<td>( \Gamma ), ft</td>
<td>0.3775 (0.96 cm)</td>
<td>0.3775 (0.96 cm)</td>
</tr>
<tr>
<td>Center force, ( F_c ), LB</td>
<td>30.45 (1.35 N)</td>
<td>1.2107 (5.385)</td>
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<tr>
<td>( \Delta P ), psi</td>
<td>32.5 (2.25 x 10^6 N/m²)</td>
<td>2.16 (1.48 x 10^6 N/m²)</td>
</tr>
<tr>
<td>Pressure force, ( F_p ), LB</td>
<td>4.981 (22.1 N)</td>
<td>3.331 (14.82 N)</td>
</tr>
<tr>
<td>( F_{tot} ), LB</td>
<td>5.286 (23.45 N)</td>
<td>4.542 (20.21 N)</td>
</tr>
<tr>
<td>( F_{tot} + 20% )</td>
<td>6.343 (28.14 N)</td>
<td>5.450 (24.25 N)</td>
</tr>
</tbody>
</table>

Use max load of LOX pump for sizing.

\[
P_0 = 6.343 \text{ lb} = 28.21 \text{ N}
\]
\[
L = 1.771 \text{ in} = 1.958 \text{ cm}
\]
\[
r_a = 0.040 \text{ in} = 1.016 \text{ cm}, D_a = 0.020 = 2.032 \text{ cm}
\]
\[
r_b = 1.553 \text{ in}, D_b = 1.06 = 2.6 \text{ cm}
\]
\[
E = 33 \times 10^6 \text{ psi} = 2.27 \times 10^6 \text{ N/m}^2
\]
\[
\delta = 0.3
\]
\[
b = \text{width of rectangular contact area}
\]
\[
P = 6.343/1.771 = 3.527 \text{ lb/in} = 14.25 \text{ N/cm}
\]
\[
b = 1.6 \sqrt{\frac{P (D_a D_a) \left[ 1/\delta^2 + 1/\delta^4 \right]}{(D_b - D_a) \left[ E_1 / E_2 \right]}}
\]
\[
b = 1.6 \sqrt{\frac{3.527 (0.040)(1.016)}{1.06 - 0.020} \left[ 1/33 \times 10^6 \right]}
\]
\[
b = 3.165 \times 10^{-4} \text{ in}
\]
\[
= 8.13 \times 10^{-4} \text{ cm}
\]
MAX CONTACT STRESS

\[ \sigma_c = 0.798 \sqrt{\frac{P \left[ \frac{D_b - D_a}{D_b + D_a} \right]}{2 \left[ 1 - \frac{h}{2} \right]}} \]

\[ \sigma_c = 0.798 \sqrt{\frac{8.227 \left( 1.106 - 0.070 \right)}{1.106 \left( 0.070 \right)} \left[ 1 - \frac{3 \times 10^{-2}}{3 \times 10^{-2}} \right]} \]

\[ \sigma_{c,\text{max}} = 33,139 \text{ psi} \quad \text{Load} = (33139)(1.771)(3.145 \times 10^{-2}) \]

\[ = 2.29 \times 10^6 \text{ N/m}^2 \quad \sigma = 8.099 \text{ psi} = 56.02 \text{ N} \]

PORT ARC LENGTH \( \Theta = 45^\circ \)

\[ \Theta = 0.553 \times \frac{\pi}{4} = 0.434'' = 1.102 \text{ cm} \]

ENTRANCE AREA = 0.0675 \( \text{ in}^2 = 0.435 \text{ cm}^2 \)

PORT WIDTH = \( \frac{0.0675}{0.434} = 0.1555'' = 0.3949 \text{ cm} \)

USE PORT WIDTHS OF 0.086'' FOR 2 PORTS

\[ L = 0.771 - 2(0.086) = 0.599'' = 1.521 \text{ cm} \]

\[ bL = 3.145 \times 10^{-6} \times 0.599 = 189.6 \times 10^{-6} \text{ in}^2 = 1.22 \times 10^{-3} \text{ cm}^2 \]

\[ S_{c,\text{max}} = \frac{8.099}{149.6 \times 10^{-6}} = 427.15 \text{ psi} = 2.94 \times 10^6 \text{ N/m}^2 \]
9.2 Thrust Load
9.2 THRUST LOADS FROM BOOST PUMP

LH2 - 7.5 %PM AT 8000 RPM, 6.5 psi/STAGE

1ST STAGE

FROM PAUL HERMANN MEMO RE 75/536

\[ F = \frac{\pi}{3} (p_1 - p_2) \left( r_1^2 - r_1 r_i + r_i^2 \right) + \pi \frac{p_1}{144} u_i^2 \left( \frac{r_2 - r_i^2}{r_2} \right) + \frac{\pi}{6} \frac{p_2}{144} u_i^2 \left( \frac{r_2 - r_i^2}{r_2} \right) \]

\[ \frac{m v_{in}}{m} \]

\[ r_1 = 0.625'' = 1.58 \text{ cm} \]
\[ r_2 = 1.822'' = 4.627 \text{ cm} \]
\[ r_3 = 2.75'' = 6.98 \text{ cm} \]
\[ P_1 = 14.7 \text{ psi} = 101 \times 10^5 \text{ N/m}^2 \]
\[ P_2 = 21.2 \text{ psi} = 146 \times 10^5 \text{ N/m}^2 \]
\[ g = 32.174 \text{ ft/sec}^2 \]
\[ \frac{\pi}{720} = 12.72 \text{ ft/sec} \]
\[ \frac{\pi}{720} = 38.77 \text{ m/sec} \]

INLET AREA = \( \pi (r_i^2 - r_i^3) = 0.944 \text{ in}^2 = 6.09 \text{ cm}^2 \)

\[ v_{in} = \frac{7.5 \times (231)}{60 \times 12 \times 0.944} = 2.55 \text{ ft/sec} = 1.77 \text{ m/sec} \]

\[ m = \frac{7.5 \times 231 \times 4.375}{32.174 \times 60 \times 12^3} = 2.27 \times 10^{-3} \text{ lb/sec}^2/\text{ft} \]
\[ = 33.12 \text{ m/sec} \]
\[ F_1 = \frac{\pi}{3} \left( 14.7 - 21.2 \right) \left( 1.822^2 + 1.822 \times 1.125 + 1.125^2 \right) \\
+ \pi \left( 21.2 \right) \left( 1.275 \right)^2 = \pi \left( 21.2 \right) \left( 1.275 \right)^2 \left( \frac{1.822^2 + 1.275^2}{1.822} \right) \\
+ 2.27 \times 10^{-3} \times 2.55 \\
F_1 = -45.16 + 5.04 + 9.52 + 0.01 \\
= -30.59 \text{ LBS.} = -136.06 \text{ N} \\

2nd STAGE

\[ \eta = 0.705'' = 1.79 \text{ cm} \]
\[ \rho_1 = 2.12 \text{ psi} = 14.6 \times 10^5 \text{ N/m}^2 \]
\[ \rho_2 = 1.822'' = 4.63 \text{ cm} \]
\[ \rho_2 = 27.7 \text{ psi} = 191 \times 10^5 \text{ N/m}^2 \]
\[ \rho_3 = 0.825'' = 2.095 \text{ cm} \]
\[ \rho_3 = 14.7 \text{ psi} = 101 \times 10^5 \text{ N/m}^2 \]
\[ u_4 = 127.2 \text{ FT/sec} \text{ or } 8 \text{ m/sec} \]
\[ g = 32.174 \text{ FT/sec}^2 \]
\[ m = 2.27 \times 10^{-3} \text{ LB-sec/FT} \]
\[ = 9.8 \text{ m/sec}^2 \]
\[ = 33 \text{ kg/m}^2 \]
\[ \text{INLET AREA} = \pi \left( 0.575^2 - 0.275^2 \right) = 0.801 \text{ m}^2 \]
\[ \text{CONV. FACTOR} = 5.17 \text{ cm}^2 \]
\[ \text{CONV. FACTOR} = 3.00 \text{ FT/sec} \]
\[ = 0.774 \text{ m/sec} \]
\[ F_2 = \frac{\pi}{3} \left( 21.2 - 27.7 \right) \left( 1.822^2 + 1.822 \times 0.705 + 0.705^2 \right) \]
\[ + \frac{\pi}{16} \times \frac{4.375 - 127.2^2}{144 \times 22.174} \left[ \frac{1.822^2 - 0.225^2}{1.822} \right] + \pi \left( 27.7 \right) \left( 0.225^2 \right) \]
\[ + 2.27 \times 10^{-3} \times 3.00 - \pi \times 14.7 \left( 0.18 \right) - \pi \left( 21.2 \right) \left( 0.225^2 \right) \]
\[ F_2 = -34.72 + 6.29 + 59.23 + 0.01 - 28.54 - 5.04 \]
\[ = -3.77 \]
\[ = -12.32 \text{ N} \]

\[ F_{\text{total}} = F_1 + F_2 \]
\[ = -30.59 + (-2.77) \]
\[ = 33.36 \text{ LBS} \]
\[ = 148.4 \text{ N} \]
THRUST LOADS FROM BOOST PUMP

LOX = 2.8 GPM AT 3000 RPM, 12 psi

\[ r_1' = 1.625'' = 1.58 \text{ cm} \]
\[ r_2' = 1.322'' = 4.63 \text{ cm} \]
\[ r_3' = 1.275'' = 6.98 \text{ cm} \]
\[ P_1 = 14.7 \text{ psi} = 101.325 \times 10^5 \text{ N/m}^2 \]
\[ P_2 = 26.7 \text{ psi} = 18.4 \times 10^5 \text{ N/m}^2 \]

\[ u_m = \frac{2.8 \times (23.1)}{720 \times 0.944} = 0.952 \text{ ft/sec} \]
\[ u_m = \frac{0.952 \times 12}{71.0} = 0.138 \text{ m/sec} \]
\[ n = \frac{2.31 \times 71.0}{32.174 \times 60 \times 12^3} = 0.018 \text{ Kgs} \]

\[ F = \frac{r_1^2}{3} (14.7 - 26.7)(1.322^2 + 1.322 \times 1.275 + 1.275^2) \]
\[ + \pi (26.7)(1.275)^2 + \pi \times 71.0 \times 71.7 \left[ \frac{1.322^2 - 1.275^2}{1.52} \right] \]
\[ + 0.952 \times 0.138 \]

\[ F = -83.38 + 6.34 + 21.70 + 0.1 \]
\[ F = 55.33 \text{ lbs} \]
\[ = 246.1 \text{ N} \]
9.3 Bearings and Seals
9.3 BEARING CALCULATIONS

Bearing Selection:

Angular contact ball bearings were chosen having 440C stainless steel balls and races with runon cages. Angular contact bearings were chosen over radials in order that one piece cages could be used. The radial contact bearings have rivetted cages. These 440C bearings have been run successfully at cryogenic temperatures and material compatibility with hydrogen and oxygen is adequate.

Only axial loads are assumed on the bearings. From the differential pressure balance on the boost pump and the two spring loads. MRC bearings were chosen.

B₁ is an R₆ bearing on the hot end (outside the seal package). It is a deep groove radial ball bearing with a 0.9525 cm bore. This bearing is clamped axially to prevent seal damage, and will operate dry.

B₂ is an R₆ bearing on the cold side that will take all of the thrust loading. It will operate in the flooded state at cryogenic temperatures. It is an angular contact ball bearing with a 1.27 cm bore.

B₃ is a 38 bearing on the cold side closest to the boost pump. This is an angular contact ball bearing with a 0.8001 cm bore. It will operate in the flooded state at cryogenic temperatures. The only axial load on this bearing will be the thrust load from the wavy spring.
## Bearing Life Calculations (MRC)

Wavy Spring Behind B3 Bearing - 5# Axial
Coil Spring At Spline - 2.85# Axial

### 3000 RPM LOX Pump

<table>
<thead>
<tr>
<th>Parameter</th>
<th>B₁</th>
<th>B₂</th>
<th>B₃</th>
</tr>
</thead>
<tbody>
<tr>
<td>Axial Load (T)</td>
<td>2.85</td>
<td></td>
<td>5.0</td>
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<tr>
<td>Thrust Load Index (K)</td>
<td>.171</td>
<td>.281</td>
<td>.171</td>
</tr>
<tr>
<td>T/K</td>
<td>16.7</td>
<td>225</td>
<td>29.2</td>
</tr>
<tr>
<td>Thrust Factor (Y)</td>
<td>3.25</td>
<td>2.1</td>
<td>2.98</td>
</tr>
<tr>
<td>Equivalent Load (B)</td>
<td>9.26</td>
<td>132.7</td>
<td>14.9</td>
</tr>
<tr>
<td>Speed Rating</td>
<td>115</td>
<td>170</td>
<td>113</td>
</tr>
<tr>
<td>Speed Factor (.6934) (A)</td>
<td>79.7</td>
<td>117.9</td>
<td>78.4</td>
</tr>
<tr>
<td>Service Factor (A)/(B)</td>
<td>8.6</td>
<td>.888</td>
<td>5.3</td>
</tr>
<tr>
<td>Life Hrs.</td>
<td>9.5 \times 10^5</td>
<td>1050</td>
<td>2.2 \times 10^5</td>
</tr>
<tr>
<td>Derate to 20%, Hrs.</td>
<td>190,000</td>
<td>210</td>
<td>44,000</td>
</tr>
</tbody>
</table>

### 8000 RPM LH₂ Pump

<table>
<thead>
<tr>
<th>Parameter</th>
<th>B₁</th>
<th>B₂</th>
<th>B₃</th>
</tr>
</thead>
<tbody>
<tr>
<td>Axial Load (T)</td>
<td>2.85</td>
<td></td>
<td>5.0</td>
</tr>
<tr>
<td>Index (K)</td>
<td>.171</td>
<td>.281</td>
<td>.171</td>
</tr>
<tr>
<td>T/K</td>
<td>16.7</td>
<td>146.7</td>
<td>29.2</td>
</tr>
<tr>
<td>Thrust Factor (Y)</td>
<td>3.25</td>
<td>2.28</td>
<td>2.98</td>
</tr>
<tr>
<td>Equivalent Load (B)</td>
<td>9.26</td>
<td>94.0</td>
<td>14.9</td>
</tr>
<tr>
<td>Speed Rating</td>
<td>115</td>
<td>170</td>
<td>113</td>
</tr>
<tr>
<td>Speed Factor (.50) (A)</td>
<td>57.5</td>
<td>85</td>
<td>56.5</td>
</tr>
<tr>
<td>Service Factor (A/B)</td>
<td>6.21</td>
<td>.904</td>
<td>3.79</td>
</tr>
<tr>
<td>Life Hrs.</td>
<td>3.59 \times 10^5</td>
<td>1108</td>
<td>.816 \times 10^5</td>
</tr>
<tr>
<td>Derate to 20%</td>
<td>71,800</td>
<td>222</td>
<td>16,300</td>
</tr>
</tbody>
</table>
### THERMAL CONTRACTION (LH₂) 37°F

SHAFT IS MONEL 400 (K-MONEL) $\alpha = 5.07 \times 10^{-6}$ in/in/°F
BEARING IS 440C STAINLESS $\alpha = 3.76 \times 10^{-6}$ in/in/°F
HOUSING IS MONEL 400 (K-MONEL)

<table>
<thead>
<tr>
<th>T.E.C. -5</th>
<th>$R_u$</th>
<th>$R_y$</th>
<th>$R_2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>BEARING BORE</td>
<td>.3750 ± .0002</td>
<td>.5000 ± .0002</td>
<td>.3150 ± .0002</td>
</tr>
<tr>
<td>SHAFT AT (37°F)</td>
<td>.4991</td>
<td>.4992 ± .0002</td>
<td>.3146 ± .0002</td>
</tr>
<tr>
<td>FIT (TIGHT)</td>
<td>0-4</td>
<td>0-4</td>
<td>0-4</td>
</tr>
<tr>
<td>SHAFT AT (62°F)</td>
<td>.3752 ± .0002</td>
<td>.5005 ± .0002</td>
<td>.3154 ± .0002</td>
</tr>
<tr>
<td>FIT (TIGHT)</td>
<td>0-4</td>
<td>3-7</td>
<td>2-6</td>
</tr>
<tr>
<td>INTERNAL CLEARANCE</td>
<td>.0003</td>
<td>.0006</td>
<td>.0005</td>
</tr>
<tr>
<td>BEARING I.D. (62°F)</td>
<td>.8750 ± .0002</td>
<td>1.1250 ± .0002</td>
<td>.7661 ± .0002</td>
</tr>
<tr>
<td>HT (37°F)</td>
<td>.1129</td>
<td>.1129</td>
<td></td>
</tr>
<tr>
<td>FIT (LOOSE)</td>
<td>0-5</td>
<td>0-5</td>
<td>2-7</td>
</tr>
<tr>
<td>HOUSING (37°F)</td>
<td>1.1229 ± .0003</td>
<td>.8647 ± .0003</td>
<td></td>
</tr>
<tr>
<td>HOUSING (62°F)</td>
<td>.8750 ± .0003</td>
<td>1.1257 ± .0003</td>
<td>.8669 ± .0003</td>
</tr>
<tr>
<td>FIT (LOOSE)</td>
<td>0-5</td>
<td>7-12</td>
<td>8-13</td>
</tr>
</tbody>
</table>

### INNER RACE

| HOOP STRESS | 26,700 | 33,500 | 42,000 psi |
| RADIAL STRESS | 6,800 | 7,400 | 13,000 |
| MAX. EQUIL. YIELD | 29,700 | 37,700 | 49,000 |

### INTERNAL CLEARANCE REQUIRED, IN.

(MIN. VALUE) | .0003 | .0006 | .0005 |
THERMAL CONTRACTION (LOX) 163°K

SHAFT AND HOUSING MONEL 400 \( a = 6.24 \times 10^{-6} \text{ in/}^{\circ}\text{F} \)
BEARINGS 440C STAINLESS \( a = 4.71 \times 10^{-6} \text{ in/}^{\circ}\text{F} \)

<table>
<thead>
<tr>
<th>ABEC-5</th>
<th>Bearing Bore</th>
<th>( R_b )</th>
<th>( R_p )</th>
<th>(-32)</th>
</tr>
</thead>
<tbody>
<tr>
<td>FIT (TIGHT)</td>
<td>at 163°K</td>
<td>0-4</td>
<td>0-4</td>
<td>0-4</td>
</tr>
<tr>
<td>SHAFT (163°K)</td>
<td>AT (68°F)</td>
<td>0.3750 ±.0002</td>
<td>0.5000 ±.0002</td>
<td>0.350 ±.0002</td>
</tr>
<tr>
<td>FIT (TIGHT)</td>
<td>0-4</td>
<td>2-6</td>
<td>2-6</td>
<td></td>
</tr>
<tr>
<td>BEARING O.D. (68°F)</td>
<td>0.8750 ±.0002</td>
<td>1.1250 ±.0002</td>
<td>0.866 ±.0002</td>
<td></td>
</tr>
<tr>
<td>FIT (LOOSE)</td>
<td>0-5</td>
<td>0-5</td>
<td>2-7</td>
<td></td>
</tr>
<tr>
<td>HOUSING (163°K)</td>
<td>AT (68°F)</td>
<td>1.1231 ±.0002</td>
<td>1.1287 ±.0002</td>
<td>1.1268 ±.0002</td>
</tr>
<tr>
<td>FIT (LOOSE)</td>
<td>0-5</td>
<td>7-12</td>
<td>7-12</td>
<td></td>
</tr>
</tbody>
</table>

USE LH₂ SHAFT AND HOUSING DIMENSIONS FOR LOX PUMP
9.3 SEAL ARRANGEMENT

Seal Selection:
For the oxygen pump a gas trap seal will be used since the pump will operate in the vertical position. The hydrogen pump will use a floating Gits radial seal acting as a labyrinth (controlled leakage) riding on the shaft. Both pumps will utilize back to back Sealol face seals operating at the warm end. Sundstrand experience has been to operate seals at room temperature isolated from the pump by a standoff pipe and insulation. Since Sundstrand has exhibited successful operation with warm seals this program will also use this concept.

Face Seals:
Sealol 800 Class

1.376" O.D. ± .008" (3.495 cm O.D. ± .008 cm)
.660" I.D. (1.676 cm I.D.)
.515" ± .020" operating length (1.308 cm ± .051 cm)
Face O.D. 1.014" ± .002" (2.57 cm ± .005)
    I.D. .874" ± .002" (2.22 cm ± .005)
Metal Bellows - Inconel
Cup and Backplate - 347 Stainless
Carbon Face Clamped By Carpenter 42
Mating Ring 347 Stainless with chrome carbide spray
9.4 Liner Pressure Plate
9.4 LINER PRESSURE PLATE

ROTATOR SIDE

Low Pressure  63.5° = 35.3%
High Pressure  116.5° = 64.7%

Pressure Plate O.D. = 1.308" = 3.32 cm
Pressure Plate I.D. = 0.375" = 0.963 cm
Area = (1.308² - 0.375²) \(\pi/4\) = 1.233 in² = 7.95 cm²
Seal O.D. = 0.824" = 2.09 in
High Pres. Area = (1.308² - 0.824²) \(\pi/4\) = 0.104 in² = 5.23 cm²
Low Pres. Area = (0.824² - 0.375²) \(\pi/4\) = 0.422 in² = 2.73 cm²
### NASA-Lewis LOX/LH₂ Vane Pump

**EWR #09449**

**Page** 7.3.1  
**Date** 11-11-77  
**Rev.**

<table>
<thead>
<tr>
<th>Component</th>
<th>LOX</th>
<th>LH₂</th>
</tr>
</thead>
<tbody>
<tr>
<td>High Pres. Pressure</td>
<td>230.7 psi (1597 × 10⁶ N/m²)</td>
<td>337.7 psi (2.33 × 10⁶ N/m²)</td>
</tr>
<tr>
<td>Low Pres. Pressure</td>
<td>27.7 psi (191 × 10⁵ N/m²)</td>
<td>27.7 psi (1.91 × 10⁶ N/m²)</td>
</tr>
<tr>
<td>Loading Rotor Side</td>
<td>196.1 lbf (872.25 N)</td>
<td>221.5 lbf (1252 N)</td>
</tr>
<tr>
<td>Loading Seal Side</td>
<td>198.7 lbf (883.8 N)</td>
<td>225.2 lbf (1266 N)</td>
</tr>
<tr>
<td>Spring Load</td>
<td>33.0 lbf (146.78 N)</td>
<td>33.0 lbf (146.7 N)</td>
</tr>
<tr>
<td>Seal Press Load*</td>
<td>29.1 lbf (129.44 N)</td>
<td>42.6 lbf (199.48 N)</td>
</tr>
<tr>
<td>Total Hydraulic Clamping</td>
<td>31.7 lbf (141 N)</td>
<td>46.3 lbf (205.9 N)</td>
</tr>
</tbody>
</table>

*Load = (4/12² × 320)π² P = 12P*

---

**Fluorocarbon Omniseal**

AR 214 C 00.664 A1Q
References


APPENDIX
**Material**

- $\Delta H_f$: Heat of oxidation, Kg. cal/100gms
- $\lambda$: Thermal diffusivity, cm$^2$/sec.
- $k$: Thermal Conductivity, cal/sec. cm $^0$K
- $\rho$: Density, gm/cm$^3$
- $c$: Specific Heat, cal/gm $^0$K
- $\frac{\Delta H_f}{\lambda}$: Burn factor
- V/O: Volume percentage
- WC: Tungsten Carbide
A5.0  Thermal

\( \dot{m} \) Mass Flow rate, kg/ per sec.

\( \dot{w} \) Work input rate, kcal/sec.

\( p \) Pressure, N/M²

\( A \) Area, M²

\( F_{sx} \) Shaft force exerted upon the fluid, Newtons

\( U \) Fluid velocity in/sec

\( J \) 778.10 n. m / Kcal

\( h \) Specific enthalpy, Kcal

\( S \) Specific entropy, kcal/kg ⁰K

\( \dot{E} \) Entropy production rate

Kcal/sec

\( \dot{Q} \) volume flow rate in²/sec

\( V_0 \) Velocity of the moving plate, m/sec.

\( b \) Clearance between plates, m

\( t \) "thickness" of flow path (at right angles to flow), m

\( l \) Length of flow path, m

\( \mu \) Fluid viscosity, average value Kg/m sec

\( r_j \) "radius" or "location" of the moving element j from shaft center line, m

\( N \) rpm

\( V_c \) Clearance volume/revolution; m³

\( b_{ks} \) kidney-side clearance, m

\( N_{ovp} \) Overall efficiency

\( N_t \) Torque efficiency

continued.........
A5.0 (continued)

Nv  Volumetric efficiency
Rmaj  Cam ring major radius, m
A 6.0 Vane Pump

L Width of the ran, cm.
R Major radius of the cam ring, cm.
\( \rho \) Minor radius of the cam ring, cm.
\( \Delta T \) Degree of subcooling of the liquid in °C
### Boost Pump

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\Delta p$</td>
<td>Pressure rise, (psi or N/M$^2$)</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Liquid Density, (lb/ft$^3$ or N/M$^3$)</td>
</tr>
<tr>
<td>$H$</td>
<td>Pressure head, (ft or M)</td>
</tr>
<tr>
<td>$S H$</td>
<td>Distance from center line to the hub</td>
</tr>
<tr>
<td>$S H$</td>
<td>Distance from center line to the tip of the leading blade edge</td>
</tr>
<tr>
<td>$\beta_{I, T}$</td>
<td>Inducer inlet relative flow angle</td>
</tr>
<tr>
<td>$r_{I, T}$</td>
<td>Blade radius at inlet</td>
</tr>
<tr>
<td>$r_{I, H}$</td>
<td>Hub radius at inlet</td>
</tr>
<tr>
<td>$\alpha_{I, T}$</td>
<td>Blade angle at the leading edge tip</td>
</tr>
<tr>
<td>$i_{T}$</td>
<td>Incidence angle</td>
</tr>
<tr>
<td>$H_I$</td>
<td>Inducer pressure head, (ft or M)</td>
</tr>
<tr>
<td>$\eta_{I, hyd}$</td>
<td>Inducer hydraulic efficiency</td>
</tr>
<tr>
<td>$r_{ex}$</td>
<td>Inducer average exit radius, (in or cm)</td>
</tr>
<tr>
<td>$\mu_I$</td>
<td>Slip factor</td>
</tr>
<tr>
<td>$C_{F I, ex}$</td>
<td>Exit flow coefficient</td>
</tr>
<tr>
<td>$\sigma_{ex}$</td>
<td>Blade blockage factor</td>
</tr>
<tr>
<td>$\beta_{ex}$</td>
<td>Exit blade angle</td>
</tr>
<tr>
<td>$\beta_{s, T}$</td>
<td>Blade angle along the shroud</td>
</tr>
<tr>
<td>$W_{EX}$</td>
<td>Fluid relative velocity at exit, (ft/sec. or m/sec)</td>
</tr>
<tr>
<td>$W_{I, T}$</td>
<td>Fluid relative velocity at inlet, (ft/sec. or M/sec)</td>
</tr>
<tr>
<td>$t_D$</td>
<td>Fluid dwell time in the inducer, (sec.)</td>
</tr>
<tr>
<td>$\sigma_1$</td>
<td>Blade solidity of 1st boost stage</td>
</tr>
<tr>
<td>$\beta_{1, LE}$</td>
<td>1st impeller leading edge blade angle</td>
</tr>
<tr>
<td>$r_{T}$</td>
<td>Impeller tip radius, (in. or cm)</td>
</tr>
<tr>
<td>$h_T$</td>
<td>Tip blade height, (in. or cm)</td>
</tr>
<tr>
<td>$\mu_T$</td>
<td>Slip factor at blade tip</td>
</tr>
</tbody>
</table>

*continued............*
Flow coefficient at impeller tip

Head Coefficient

Leading edge radius of 2nd impeller, cm.

Pressure head at 2nd stage leading edge, cm.

2nd stage leading edge radius, cm.

2nd stage leading edge blade angle

Inlet flow angle

2nd stage incidence angle

2nd stage impeller flow coefficient

Relative velocity at 2nd stage impeller discharge, (ft/sec. or M/sec.)

Relative velocity at 2nd stage impeller inlet., (ft/sec. or M/sec.)
A 8.0 Performance Prediction

\[ Q \quad \text{Discharge flow rate, GPM or Liter/Min.} \]
\[ \Delta P \quad \text{Pressure Rise, ft. or M.} \]
\[ \eta_{\text{Boost}} \quad \text{Efficiency of the centrifugal stage alone} \]
\[ \eta'_{\text{Boost}} \quad \text{Efficiency of the centrifugal stage in the entire pump package} \]
\[ \eta_{\text{Vane}} \quad \text{Efficiency of the vane stage} \]
\[ \eta'_{\text{Vane}} \quad \text{Efficiency of the vane stage in the whole pump package} \]
\[ \eta'_{\text{(Boost & Vane)}} \quad \text{Efficiency of the entire pump package} \]
\[ P_x \quad \text{Pressure at the fictitious station X, psi or N/M}^2 \]
HP  Horsepower
T  Torque, in. lb. or cm.kg.
N  Speed in RPM
σu  Ultimate Tensile Strength, psi or N/M²
σy  Tensile yield strength, psi or N/M²
τγ  Shear yield strength, psi or N/M²
σcγ  Compressive yield strength, psi or N/M²
M.S.  Margin of safety
N  Safety factor
b  Key width, in or cm.
l  Key length, in. or cm.
σb  Bending stress, psi or N/M²
t  Vanes thickness, in. or cm.
a  Stroke, max. vane travel, in. or cm.
w  Axial vane width, in. or cm.
h  Vane height, in. or cm.
Fc  Centrifugal force, lb. or N
Fp  Pressure force, lb. or N
σ  Poision ratio
b  Width of rectangular contact area, in. or cm.
Sc  Contact stress, psi or N/M²
A 9.2

\( r_1 \) Distance from center line to the labyrinth on the inlet side of the impeller, in. or cm

\( r'_1 \) Impeller inlet radius, in. or cm.

\( r_2 \) Distance from center line to the labyrinth on the back side of the impeller, in. or cm.

\( r_3 \) Impeller discharge radius, in. or cm.

\( \bar{v}_m \) Average fluid inlet velocity, ft./sec. or M/sec.

\( \dot{m} \) Mass flow rate, lb.-sec.²/ft. or gm/sec.

\( U_t \) Impeller tip velocity, ft./sec. or M/sec.

\( F_1 \) Axial thrust on first stage impeller, lb. or N.

\( F_2 \) Axial thrust on second stage impeller, lb. or N.