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HYDRODYNAMIC DESIGN REPORT
INTERMEDIATE SIZE INDUCER PUMP
FOR SODIUM OPERATION

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Prepared for
ATOMICS INTERNATIONAL DIVISION
ROCKWELL INTERNATIONAL CORPORATION

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INTRODUCTION

The Intermediate-Size Inducer Pump (I.S.I.P.) was designed to operate in the existing FFTF pump housing and to achieve the required pump head rise at the same speed and flow as the existing pump. The existing pump consists of four basic hydrodynamic elements:

1. Inlet Elbow
2. Centrifugal Impeller
3. Vaned Diffuser
4. Discharge Housing

All of these elements except the centrifugal impeller were to be retained in an unmodified form for the I.S.I.P. design. The centrifugal impeller was to be replaced with a new design consisting of both an inducer and centrifugal impeller. The objective is to demonstrate the capability of designing an inducer pump for long life in sodium operation so that the advantages of the inducer pump can be realized in future sodium pump applications. These advantages consist primarily in the smaller envelope size and lower weight realized as a result of the better suction performance capability of the inducer. These advantages result in significant cost savings and ease of fabrication and handling for the very large pumps required in many of the reactor coolant loop systems.

The vaned diffuser in the existing design attempts to diffuse the flow while turning it which typically limits the stable range of operation. Tests of the existing pump have demonstrated instability at a flow of approximately seventy percent of the design flow. The I.S.I.P. design must use this same diffuser, therefore to attempt to extend the stable operating range, Rocketdyne has chosen to design the centrifugal impeller to a smaller outer diameter to leave sufficient space to add an intermediate vaned diffuser upstream of the existing diffuser. This new diffuser would be designed to operate over a wider flow range and should provide a wider range of stable operation for the overall pump.

Thus, the I.S.I.P. design consists of three new hydrodynamic elements:

1. Inducer
2. Centrifugal Impeller
3. Intermediate Vaned Diffuser

This report describes the hydrodynamic design features of each of these. The report also discusses the design rationale and methods of analysis. Following this, the hydrodynamic performance of the design is presented and compared with the design specification.

The specified sodium characteristics used in the design are:

Fluid Inlet Temperature = 1050° F

Fluid Specific Weight = 50.97 lb/ft^3

SUMMARY

An inducer pump has been designed to fit within the existing housing of the FFTF facility. The new design consists of three primary hydrodynamic components: An inducer, centrifugal impeller, and intermediate vaned diffuser. The hydrodynamic design of each component has been based on the design procedures and analytical techniques developed over a twenty-five year period and proven to be accurate by detailed comparison with test data.

The design has been shown to meet the specified requirements in every area where an analytical prediction of the performance can be made. The head-rise and efficiency of the pump have design margins so that the calculated values actually exceed requirements over the full range of operation. The suction performance of the design provides a very large margin at the operating NPSH value which is the major advantage of using the inducer pump. The suction performance margin at off-design is estimated to be adequate for suction performance, but may not be as large as desired for life considerations. Testing of a model inducer is required to verify the suction performance at off-design and to evaluate the life potential. Operation of an impeller pump with similar margins would also require test verification of the life potential.

The design has incorporated features that provide for long life at the design point. These features have been established based on proven performance of commercial waterjet pumps designed by Rocketdyne. Each of the major design parameters contributing to long life fall within the range of previously verified designs except for a somewhat larger tip clearance. Analysis has indicated that the tip clearance is acceptable.

DESIGN DESCRIPTION

Inducer

Figure 1 shows a layout of the pump configuration indicating the various hydrodynamic components to be discussed. The inducer design was based on the same design practices established and demonstrated by the successful waterjet inducer designs at Rocketdyne.

The inducer was designed with four blades to minimize the potential of tip vortex collapse on the blade surface and thereby eliminate collapse damage. The four-bladed design also provides minimum potential for radial loads for any NPSH margin provided to the inducer. The envelope of the pump was sufficient to handle the required length of the inducer using only four full blades without any partial blades. Elimination of the partials improves confidence in achieving the long life required.

The inducer tip diameter was designed to be 18.53 inches, an unshrouded tip configuration being used to achieve cavitation collapse in the fluid passages and not on the blade material. The tip diameter was dictated by three factors:

1. The existing housing outer diameter.
2. The required operating clearance of 0.050 in. radial.
3. The need to provide a smooth wall contour over the inducer tip.

The longer length of the four-bladed inducer requires it to extend beyond the point where the leading edge of the existing impeller was located. As a result, if the existing housing were used without a liner over the tip of the inducer, a circumferential slot would occur over the inducer and could create cavitating vortices which could cause hardware damage. Thus, a liner was added to eliminate any slot over the inducer itself.

FIGURE 1.

The required tip clearance is set by the rotor dynamics of the pump in the facility. This clearance is somewhat larger than values commonly achievable in waterjet practice. However, the effect of the larger clearance was included in both the performance and life analyses. To further avoid any potential for rubbing at the tip, the blade was designed for zero cant angle so that any blade movement would not decrease the operating clearance.

The hub contour was selected to optimize the performance and match the required impeller inlet design. The hub diameter at the inlet is 6.626 in. giving a hub/tip diameter ratio of 0.358. This inlet diameter is selected to achieve the desired suction performance and to maintain the flow coefficient in the region where confidence in long life is high. The discharge hub diameter is 11.358 in. and is selected for optimum efficiency of the inducer-impeller combination. The contour shape of the hub from inlet-to-discharge is based on established procedures optimized for achieving the proper head-rise distribution along the stream surfaces through the inducer.

The blade design includes such factors as leading edge blade angles, blade camber distribution, blade thickness distribution, leading-edge and trailing-edge sweep and blade solidities. These values are selected based on previous commercial experience, and in each case the I.S.I.P. values are consistent with waterjet design experience at Rocketdyne. The blade design is of primary importance in achieving

1. The optimum suction performance capability
2. The long-life with no detrimental blade damage
3. The required head and maximum efficiency

One of the results desired in the design is a head-rise distribution through the inducer that is monotonically increasing at a somewhat uniform rate. Figure 2 shows the result achieved in the I.S.I.P. inducer design

ISIP INDUCER
PREDICTED HEAD RISE THROUGH INDUCER
(BASED ON R3DP)
Q = 14600 GPM
N = 1110 RPM

FIGURE 2.

indicating its agreement with this design criterion. It is also desirable to achieve a relatively uniform head distribution at the inducer discharge and impeller inlet. Figure 3 indicates that this was accomplished.

Five Rocketdyne computer programs were used in the hydrodynamic design and analysis of the inducer. The first was the nonisentropic radial equilibrium program (NISRE) which calculates the head-rise and meridional velocity leaving the inducer at all radii for a fixed (radial) blade. This was used in the preliminary design portion of the analysis to obtain the desired moderate blade loading.

After the blade meanline was calculated, an initial blade thickness was assigned on the basis of past experience. The entire inducer geometry was then input into the three-dimensional analysis program. This program (unlike the NISRE) takes into account the curvature of the walls before, within, and following the inducer as well as the blade contours. It calculates the local static pressures on the suction and pressure sides of the blades at all stations and at all radii. These pressures are used to determine where the blade will cavitate and where the cavity will collapse. They are also used by Stress to determine the blade loadings from which the final blade thickness is determined. The program also calculates the head-rise at each streamline and the relative velocities entering the centrifugal impeller. These are used to determine the desired blade angles at this station.

The third program used was the blade leading edge loading program which calculates the loading on the leading edge wedge. This loading was combined with that of the three-dimensional analysis program and given to Stress. The combined loads at the design point are shown on Figure 4.

The fourth program predicts the jet stopping distance of a cavity progressing across the passage from the suction side of the blade to the pressure side. If the cavity travels as far as the pressure side of the next blade before collapsing, it might cause damage on that blade.

ISIP INDUCER
PREDICTED HEAD RISE AT
INDUCER DISCHARGE

$Q = 14600$ GPM

$N = 1110$ RPM

FIGURE 3.

ISIP FOUR-BLADED INDUCER
BLADE LOADING
Q = 14600 GPM
N = 1110 RPM

FIGURE 4.

The fifth program is the "INDANA" program which is used to calculate the inducer head and efficiency over the full operating range. The calculated performance for I.S.I.P. is presented later in the report in the performance section.

Impeller

The impeller is a shrouded radial-discharge centrifugal-type impeller designed to generate the required head-rise. The impeller has five full blades and five additional partial blades at the discharge. The location of the leading edge of the partials is shown schematically on Figure 1. The number of blades is chosen to optimize the efficiency and generate the required head at the design point. The transition from four inducer blades to five inlet impeller blades is not uncommon in optimum design practice and creates no detrimental effects.

The inlet diameters and blade angles were set based on flow conditions at the inducer discharge. The diameters are essentially a continuation of the inducer discharge diameters. The flow angles at the impeller blade leading edge are a result of an extrapolation from the inducer discharge using continuity of flow and constant angular momentum. The eye diameter is 19.0 inches and inlet hub diameter is 11.36 inches. The eye diameter is slightly larger than the inducer diameter to accommodate the additional wear ring leakage flow.

The inlet section of the blades are designed for no cavitation, and therefore no cavitation damage and long life are assured. This is made possible by the inclusion of the upstream inducer which provides sufficient head-rise to eliminate impeller cavitation either on the blade or in the vortices generated in the wear ring return flow.

The discharge width of 2.75 inches was set equal to the first diffuser width which in turn was set equal to that of the existing diffuser.

This width is wider than required for this operating condition and probably results in a somewhat lower efficiency. However, the I.S.I.P. design must match the existing diffuser in the existing housing, and a mismatch on the width could be a larger efficiency penalty than using the existing width. The discharge diameter of 35.40 inches, number of blades, and blade angle were balanced in such a way to allow room for another diffuser and still generate the required head. This was based on previous Rocketdyne pump experience developed in the design and test of numerous centrifugal impellers used in the turbopumps of the engines for the national space program.

The shroud and hub contour shapes and the blade camber distributions are selected to achieve a uniform loading and head generation distribution. Figure 5 presents the average head-rise at the design point generated through the impeller as a function of meridional length. Similar pressure loading curves at lower flows were generated to assure structural integrity at off design. Figure 6 presents the blade load distribution generated for the Stress department to determine blade stresses.

Two computer programs are used in the impeller design and analysis effort. The quasi-three-dimensional analysis program "VELDIS" was used to analyze the local velocity and pressure field throughout the impeller bladed section. This program provides the design verification that no detrimental regions of back flow or cavitation are occurring in the impeller. The output from the program also provides the head distribution and blade loads previously presented in Figures 5 and 6. The second program is an overall pump performance prediction program that performs a detailed calculation of the various losses that can occur in the pumping elements as well as in the diffusers. This program was used to calculate the head and efficiency over the full operating range, and the results are presented in a later section.

Intermediate Vaned Diffuser

A short, ring diffuser was added between the impeller and the existing

INTERMEDIATE-SIZE INDUCER PUMP
IMPELLER HEAD RISE ONLY

Q = 15816 GPM

N = 1110 RPM

FIGURE 5.

INTERMEDIATE-SIZE INDUCER PUMP

"VELDIS" SOLUTION

Q = 15816 GPM

N = 1110 RPM

FIGURE 6.

diffuser in an attempt to increase the stable flow range of operation. This diffuser will provide a flow into the Westinghouse diffuser with a smaller angular variation over the flow range than would be available directly from the impeller.

The discharge width and number of vanes were set to provide a match of the inlet flow angles to the vanes of the Westinghouse diffuser. The eleven vanes of the intermediate diffuser are staggered between the eleven vanes of the existing diffuser to minimize boundary layer build up and separation with its attendant loss in efficiency. The diameters of the intermediate diffuser are 37.2 in. at the inlet and 42.0 in. at the discharge. The discharge diameter was set by the housing geometry and the inlet diameter was set by the impeller diameter, allowing sufficient clearance to avoid cavitation or fatigue damage to the impeller or diffuser.

The vane profile is a double-circular-arc design with a 7.4 percent thickness. The profile and thickness were selected consistent with good hydrodynamic practice and to control the hydrodynamic loading on the vanes to meet the required structural integrity.

The inlet vane angle was set to match the flow from the impeller at the design flow and the discharge angle was set to provide a flow matching the inlet vane angle of the Westinghouse diffuser. The required vane angles and diameters combine to give a solidity that provides the required diffusion.

The diffuser was designed using the same methods developed for the space program centrifugal pump designs. These design techniques have been proven based on tests over an extensive flow range. The losses associated with the flow through the diffuser were calculated using the same centrifugal pump loss program discussed in the impeller section.

HYDRODYNAMIC PERFORMANCE

Design Point

The specification design point required an overall head-rise of 500 feet flange-to-flange while pumping a delivered flow of 14500 gpm of sodium at a rotational speed of 1110 rpm. The I.S.I.P. calculated head-rise at these conditions is 546 feet providing a nine percent margin in meeting the required 500 feet. The projected efficiency of the pump at the design point is 74 percent giving a required power of 2037 Hp. This quoted efficiency also has a margin, based on calculations, of four percent.

The program used for the analysis has been quite successful in the prediction of performance on other pumps. This program was developed over the past twenty-five years and checked out by consistently comparing predicted results with pump internal data collected in various test facilities. The program calculates the Euler head-rise for each of the rotating elements and the losses that affect both head-rise and power in each of the components. Table 1 presents the respective head and loss values for each of the components at the design point indicating the margin in the design head-rise. The head can easily be reduced to exactly 500 feet, if required, by trimming the impeller. The calculations for the Westinghouse diffuser system were performed with the available information for the design, but they contain a degree of uncertainty due to the lack of more specific design information.

Range of Operation

The pump is required to operate over a relatively large flow range. The maximum flow at the design speed is 18,000 gpm at which point the head-rise must exceed 375 feet. The minimum flow is not specified, but it was desired to achieve a stable operation for flows lower than that achieved with the Westinghouse design.

TABLE 1
DESIGN POINT PERFORMANCE CALCULATIONS

DELIVERED FLOW = 14,500 GPM
SPEED = 1110 RPM

<u>COMPONENT</u>	<u>EULER HEAD</u>	<u>LOSSES</u>	<u>CUMULATIVE DEVELOPED HEAD</u>
INDUCER	79.2	7.2	72 FT.
IMPELLER	630.0	63.0	639
INTERMEDIATE DIFFUSER	-	17.0	622
WESTINGHOUSE DIFFUSER	-	43.0	579
DISCHARGE SYSTEM	-	33.0	546 FT.
DESIGN MARGIN = 46 FT. (9%)			

The predicted overall head and efficiency are shown in Figure 7 assuming that the margin shown by the calculations is not realized in actual operation. The head-rise at the 18,000 gpm flow is projected to be 470 feet well in excess of the 375-foot minimum value. The negative slope of the head-flow curve is predicted to extend down to 8000 gpm. The existing pump was reported to have a stability problem between 10,000 and 12,000 gpm, and the vaned diffuser is expected to be the primary cause of the instability. The additional diffuser designed by Rocketdyne should help to extend the stable flow range, but the stable range cannot be guaranteed because the same Westinghouse diffuser is being used.

Figure 8 presents the head and efficiency of the inducer portion of the pump. This head must be sufficient to keep the impeller out of the cavitation region which causes head fall-off.

Pony Motor Operation

The specification required that at the pony motor speed of 94 rpm, the head-rise was not to exceed five feet of head. Figure 9 presents the scaled head-rise over the required flow range. The maximum head is four feet which is within the specification requirement.

Life Requirements

The specification requires a 20 year life at the design point and a worst-case, off-design life of 250 hours at 18,000 gpm and design speed of 1110 rpm. The inducer design incorporates the long-life design features established and verified through Rocketdyne's waterjet pump experience. The significant parameters that affect the life of the machine have all been chosen to fall within the range of those parameters actually used in the waterjets. This includes such parameters as the leading edge sweep, flow coefficient, incidence angle, inlet blade angle distribution, cant angle, and tip thickness. The one parameter which was slightly outside

INTERMEDIATE-SIZE INDUCER PUMP
PREDICTED PERFORMANCE
1110 RPM

FIGURE 7.

INTERMEDIATE - SIZE INDUCER PUMP
PREDICTED PERFORMANCE
INDUCER ONLY

1110 RPM
TIP DIAMETER = 18.53 IN.

FIGURE 8.

INTERMEDIATE-SIZE INDUCER PUMP
PREDICTED PERFORMANCE
PONY MOTOR OPERATION
94 RPM

FIGURE 9.

the waterjet experience range was the tip-clearance-to-blade-height ratio. However, Rocketdyne does have an analytical program for predicting the influence of this clearance on the potential damage due to the tip vortex cavity. This analytical program has been correlated with waterjet life test results. The analysis indicates that the selected four-bladed design will be able to operate with long life at the required operating clearance.

Other features incorporated to maximize the life include designing for zero cant angle of the blade to prevent a detrimental minimum tip clearance in case of blade bending, elimination of the slot over the inducer to prevent generation of slot-edge vortex cavitation, and increasing the blade thickness at the tip to the maximum value consistent with good hydrodynamic suction performance. This latter feature was included to provide more structural integrity to the blade to minimize blade movement in the high temperature sodium environment.

The maximum flow of 18,000 gpm represents a 24 percent increase in flow over the design point. At such a large off-design condition any rotating component, inducer or impeller, would experience some cavitation damage if the NPSH margin is low. The NPSH margin required to prevent such damage for the I.S.I.P. design must be determined from water tests of model hardware. Such testing does provide a very economical approach to the problem and yields the required technological data for assessing the design in the very early stages of the overall program. This provides the opportunity for hardware modifications, if necessary, before fabricating any full-size hardware.

Suction Performance

The available NPSH provided to the inducer at the design flow is 47 feet. At 18,000 gpm this value drops to 40 feet. Rocketdyne's I.S.I.P design is capable of achieving a predicted required NPSH of 12.8 feet at the design flow. This capability is based on the assumption that the required

suction performance is the same in sodium as in water. Since both fluids have a very low vapor pressure, the thermodynamic suppression head effects should be negligible so that the above assumption is valid.

Rocketdyne's ability to meet this predicted performance is based on an extensive background of designing and testing high suction performance inducers. More recently, the capability of the subscale sodium inducer demonstrated the ability to achieve the predicted performance. The design features which are required to achieve these results have been designed into the I.S.I.P. consistent with previous Rocketdyne experience. No design compromises were required due to the sodium operation or other pump features with one exception. The outer diameter of the inducer was reduced, raising the required NPSH, to provide a smooth liner over the inducer and eliminate the slot over the inducer present in the existing housing.

The suction performance at off-design conditions is more difficult to predict. Based on the subscale sodium inducer the ratio of breakdown NPSH at 1.24 times design flow (18,000 gpm) compared to that at the design point (14,500 gpm) was 2.26. Using this ratio, the I.S.I.P. required NPSH at the maximum flow would be 29 feet giving a margin of 38 percent. This is an adequate margin for suction performance but is relatively low for achieving life. An inducer or impeller operating with such small margin at this off-design value would be expected to experience damage, and the life then depends on the damage rate compared to the off-design life requirement. At 250 hours of life over a 20 year period, the inducer could be expected to maintain its performance capability even though some damage occurred.

The presence of the inducer provides sufficient head-rise at the impeller inlet to keep the impeller from experiencing cavitation performance loss. The maximum required suction specific speed of the impeller is only approximately 5200 at the off-design flow of 18,000 gpm. At the design flow, it will be significantly less than this.

Radial Loads

The specification requires that the design incorporate polar symmetry to minimize radial loads. Such polar symmetry was incorporated in the design. The radial loads are also minimized by the incorporation of a four-bladed inducer, use of a lower impeller diameter than used in the existing pump, use of a more stable vaned diffuser system, and the complete polar symmetry of the existing diffuser discharge. The maximum radial loads to be experienced would occur at the lower flows. The radial loads should be less for the Rocketdyne design than the existing design, but for purposes of analysis a not-to-exceed maximum load was estimated at a low flow of 8000 gpm. These not-to-exceed estimates were:

Impeller: 800 pounds

Inducer: 13,300 in-lb moment

The impeller load is based on an estimate of six percent of the axial thrust acting as a radial thrust. The inducer moment is based on tests of three-bladed inducers which are more likely to experience radial loads. However, the inducer radial load would not be expected to occur at the NPSH margins expected for the I.S.I.P. design.

Axial Thrust Control

The specification required that the axial thrust not exceed 70,000 pounds in the upward direction and 40,000 pounds downward. It also required that these thrust values be met with an assumed variation of the impeller discharge static pressure of ten percent from front shroud to rear shroud. The axial thrust control was to be achieved without use of a balance piston.

Figure 10 presents the nominal axial thrust for the I.S.I.P. design. The nominal thrust is less than 13,000 pounds at all operating conditions.

INTERMEDIATE-SIZE INDUCER PUMP
N = 1110 RPM
INDUCER INLET PRESSURE = 16.64 PSIA TOTAL
12 BLEED HOLES
1.25 IN. BLEED HOLE DIAMETER

FIGURE 10.

With the additional assumption of the ten percent variation in static pressure the following maximum conditions were obtained:

1. Maximum thrust equals 24,800 pounds downward at 8000 gpm.
2. Maximum thrust equals 7,200 pounds upward at 18000 gpm.

Case one assumes the rear shroud pressure is higher by ten percent and case two assumes the front shroud pressure is higher. This ten percent variation in static pressure is believed to be higher than will be experienced in the pump, but the thrust values are all well within the limits specified, and no balance piston system is required.

Recirculating Flows

The specification originally required that the hydrostatic bearing discharge pressure was not to exceed 10 psi above the inducer inlet pressure at the design point. To achieve this goal would require that the bearing discharge flow be ducted back to the inducer inlet. However, because the pump must be fitted into the existing housing, the impeller rear wear ring flow would also have to be dumped into the inducer inlet, or at least a portion of it if another rotating seal were added. Rocketdyne did not like this concept because dumping any significant amount of fluid into the inducer inlet could be detrimental to the suction performance capability. A simpler approach was analyzed and is shown in Figure 11. The bearing discharge flow and rear wear ring flow are collected and dumped into the impeller inlet rather than the inducer inlet. The impeller has the tolerance required to accept the flow without any degradation in overall suction performance. Using this approach gives a bearing discharge pressure that is 13 psi above the inducer inlet pressure rather than the 10 psi originally specified. It was agreed at Atomics International to change the specification to approve the Rocketdyne design.

RECIRCULATION FLOWS
INTERMEDIATE-SIZE INDUCER PUMP
1110 RPM
DESIGN FLOW

FIGURE 11.

Figure 12 shows the differential pressure of the bearing discharge pressure minus inducer inlet pressure as a function of flowrate. The curve results are based on usage of twelve bleed holes at a diameter of 1.25 inches. The total flowrate through the impeller includes the bearing flow, rear wear ring flow, and front wear ring flow. The wear rings are located to achieve the axial thrust results previously presented. The flow through the wear rings is minimized by using properly designed stepped-labyrinth seals.

INTERMEDIATE-SIZE INDUCER PUMP
N = 1110 RPM
INDUCER INLET PRESSURE = 16.64 PSIA TOTAL
12 BLEED HOLES
1.25 IN. BLEED HOLE DIAMETER

ΔP = BRG. DISCH. STATIC PRESS. - INDUCER INLET STATIC PR.

FIGURE 12.

CONCLUSIONS

The hydrodynamic design has been based on techniques developed and proven over the past twenty-five year period. The analytical tools used are the most advanced available and have been checked out by correlation with available test data. The confidence in achieving the design performance predictions is, therefore, very high.

The design has been shown to meet the specification requirements in all cases where analytical predictions can be made. In fact, design margins have been included in all of these cases to provide additional confidence in meeting the predicted values. Two areas need verification through testing, regardless of whether an inducer pump or an impeller-only design is incorporated. The first is the suction performance margin at the highest flow condition. The quoted margin of 38 percent is expected to be representative but requires test support. The second area is the life characteristics of the design. All of the design features are based on proven designs for long life waterjet pumps at the design point. Life at the off-design flow requires test evaluation.