Centrifugal Pumps for Rocket Engines

W. E. Campbell and J. Farquhar

NERVA Rocket Operations[1]
Aerojet Nuclear Systems Company

The use of centrifugal pumps for rocket engines is described in terms of general requirements of operational and planned systems. Hydrodynamic and mechanical design considerations and techniques and test procedures are summarized. Some of the pump development experiences, in terms of both problems and solutions, are highlighted.

The centrifugal pump has been an important element of the history of pump-fed liquid propellant rocket engines. The use of this type of pump is the resultant of its relative simplicity and reliability, wide operating flow range, and adequate performance. Additionally, at the outset of the rocket program, centrifugal pumping system experience was available from the commercial pumping industry.

Successful development of centrifugal pumping systems, however, has not been without some dramatic problems. These problems have been due mostly to the difficult application parameters, such as

(1) A wide variety of design requirements—flow rate, head rise, and fluid characteristics

(2) The importance of very low suction pressure

(3) The importance of high efficiency while achieving high stage head rise

(4) The effects of unknown or difficult-to-handle cryogenic, corrosive, and low-density pumping fluids

(5) Very rapid start, shutdown, and flow change transients

---

[1] The Nuclear Engine for Rocket Vehicle Application Program (NERVA) is administered by the Space Nuclear Propulsion Office, a joint office of the U.S. Atomic Energy Commission and the National Aeronautics and Space Administration. Aerojet-General Corporation, as prime contractor for the engine system, and Westinghouse Electric Corporation, as subcontractor for the nuclear subsystem, are developing a nuclear propulsion system for deep space travel.

The solutions of these problems resulted in improved analytical techniques for hydrodynamic and mechanical design, a partial understanding of such hydrodynamic/mechanical interactions as axial thrust and blade stresses, new and interesting fabrication and materials technology, and effective development test techniques.

The intent of this paper is to orient the reader to the general requirements for, and experience gained from, rocket engine pumps already operational and currently being designed; to summarize the hydrodynamic and mechanical design considerations and techniques and test procedures employed; and to highlight the development experience in terms of both problems and solutions.

Primary emphasis centers on the pumps designed and developed by Aerojet-General Corporation, but the general discussions of requirements, sizes, materials, and fabrication methods reflect the experience of other companies as known by the authors.

GENERAL REQUIREMENTS

Flow Rate and Variations

Rocket engine centrifugal pumps have been developed for flow rates as low as 10 gpm and as high as 25 000 gpm, with the preponderance of systems operating in the 1000–10 000 gpm range. Earlier ballistic missiles, being essentially fixed thrust systems, required a flow-coefficient range up to only ±10 percent; some space vehicle systems need a thrust variation (throttling) of 10:1, while others have to develop a "pump-out" capability (the ability to continue to deliver full engine flow for full thrust engine operation after a failure of one of the two parallel turbopumps). These requirements result in a flow-coefficient[2] shift of 65 percent for the throttling case and 2:1 for the pump-out case. Table I summarizes the important parameters of the pumps utilized in some well-known rocket engines.

Head or Pressure Rise and Variations

Head or pressure requirements likewise vary considerably; very large head rises are required for low-density liquid hydrogen. Rocket pumps have been built for pressure rises ranging from 700 psi to about 5000–6000 psi; corresponding head-rise values of 1500 to 12 000 ft for conventional density fluids and 20 000 to nearly 200 000 ft for hydrogen are seen in these applications. The earlier ballistic missiles required small variations

[2] Defined at end of paper.

of pressure—only enough to cover engine tolerance effects, say ±10 percent; the more recent throttleable systems require a delivered pressure variation of 10:1.

Specific Speed

Specific speeds for the pumps in operating systems have ranged from 500 to 2000 (rpm, gpm, ft units); the majority lie between 600 and 1500. These unfavorably low values are dictated by high head rise, low suction pressure speed limits, and mechanical speed limits of the drive system.

NPSH/NPSP

A tank NPSH (net positive suction head) of zero (i.e., the ability to pump saturated fluid at the inlet to the pump suction line) is becoming a popular criterion for liquid hydrogen systems and has been demonstrated with inducers adaptable to centrifugal pumps. For other fluids such as liquid oxygen, kerosene, and the storable propellants, NPSH values have ranged from 20 ft to 130 ft for current systems. Conventional suction specific speeds being achieved with acceptably low cavitating head losses (2 percent or less) are about 35 000 (rpm, gpm, ft units).

Fluids

A fairly large variety of conventional and unconventional propellants and testing fluids have been handled, as summarized in table II.

Duration/Number of Starts

The duration or life required of rocket pumps is low: ballistic-missile pumps operate for 2 to 5 minutes; space vehicular pumps for 1 to 10 minutes. Newer applications, such as the nuclear and chemical space shuttles, may require multimission usage cycles up to 20 minutes in duration for a usable life of 10 hours. The number of operational cycles has varied from 1 to current requirements of perhaps 100. Testing has typically been conducted on several multiples of these operating times, but the accrual of more than a few hours operating time on a single unit has been rare.

DESIGN CONSIDERATIONS AND TECHNIQUES

Types of Systems

Various pump and drive arrangements have been used and are being portrayed in conceptual designs of new systems. The drive systems for the main propellant pumps have been limited exclusively to gas turbines.

TABLE I.—*Summary of Characteristics—Typical Rocket Engine Pumps*

Pump	Flow rate (gpm)	Head rise (ft)	Shaft speed (rpm)	SHP	Impeller OD (in.)	Pump fluid	Impeller description	Housing description
Titan I fuel first stage	1 642	2 758	8 814	1 734	10.06	RP–1	Aluminum forging—machined	Aluminum casting—volute
Titan I oxidizer fist stage	2 629	1 494	7 979	1 770	10.06	LO_2	Aluminum forging—machined	Aluminum casting—volute
Titan I fuel second stage	659	2 985	24 888	697	4.695	RP–1	Aluminum forging—machined	Aluminum casting—volute
Titan I oxidizer second stage	1 108	1 548	8 831	756	9.00	LO_2	Aluminum forging—machined	Aluminum casting—volute
Titan II fuel first stage	2 234	3 273	8 800	2 295	10.86	A–50	Aluminum forging—machined	Aluminum casting—volute
Titan II oxidizer first stage	2 695	1 640	8 008	2 146	9.635	N_2O_4	Aluminum forging—machined	Aluminum casting—volute
Titan II fuel second stage	896	2 988	22 830	856	5.035	A–50	Aluminum forging—machined	Aluminum casting—volute
Titan II oxidizer second stage	1 006	1 717	8 100	899	8.825	N_2O_4	Aluminum forging—machined	Aluminum casting—volute

J–2 oxidizer	3 000	2 400	8 000	3 000	10	LO_2	Aluminum casting—shrouded	Aluminum casting—volute
J–2S fuel	10 000	50 000	28 000	9 500	11.5	LH_2	Titanium forging—machined, shrouded	Aluminum casting—rolled over volute
M–1 oxidizer	19 000	3 400	3 650	27 000	29	LO_2	Aluminum forging—machined	304L–SS casting—rolled over volute and diffuser
F–1 fuel	15 000	6 000	6 000	27 000	23.3	RP–1	Aluminum casting—machined, shrouded	Aluminum casting—volute, double discharge
F–1 oxidizer	24 000	3 600	6 000	37 000	19.5	LO_2	Aluminum casting—machined, shrouded	Aluminum casting—volute, double discharge
CENTAUR fuel	195.7	29 000	28 600	568	----------	LH_2	Aluminum castings—mounted back-to-back—back shrouded	Aluminum casting—volute, and diffusers
CENTAUR oxidizer	185.3	840	11 400	90.4	----------	LO_2	Stainless forging—shrouded	Aluminum casting—volute, and diffuser
NERVA (I)	7 850	33 100	22 800	7 180	12.25	LH_2	Aluminum forging—machined	Aluminum casting—volute, diffuser

TABLE II.—*Summary of Typical Pumping Fluids*

Fluid	Symbol	Specific gravity	Temperature, °F	Character
Liquid oxygen	LO_2	1.15	−297	Corrosive, explosive
Liquid hydrogen	LH_2	0.07	−423	Cryogenic, low density
Kerosene	RP–1	0.80	70	Easy to handle
Nitrogen tetroxide	N_2O_4	1.45	70	Toxic, corrosive
Aerozyne-50 (½ UDMH/½ hydrazine)	A–50	0.90	70	Toxic
Liquid nitrogen	LN_2	0.80	−320	Cryogenic, easy to handle (substitute test fluid)
Water	H_2O	1.0	70	Substitute test fluid

The pump rotors are supported on shafts carried on rolling-element bearings cooled by propellant or lubricated by specially supplied oil. Figures 1a through 1e illustrate the typical pump and drive arrangements.

Early configurations were typified by single-stage pumps running at suction-pressure-limited speeds, with one or both propellant pumps driven in parallel through a speed reducing gearbox by a common turbine (Titan I, Atlas, Thor, Titan II). The first operational hydrogen/oxygen engine, RL–10 (used on Centaur), employed a two-stage direct-driven hydrogen pump and an LO_2 pump gear driven by the same turbine. This system also employed remote (tank mounted) electric-motor-driven boost pumps. The F–1 engines for Saturn IC utilize in-line low-speed (6000 rpm) single-stage, suction-pressure-limited kerosene and LO_2 pumps direct driven by a common turbine on a single shaft. J–2 and M–1 (200 000-lb and 1 500 000-lb thrust hydrogen/oxygen engines) introduced the separate direct-drive pump concept for each propellant. In these engines, the high-speed hydrogen pump and the lower-speed oxygen pump were each driven at moderate speed. An updated version of the J–2 engine utilizes a hydrogen pump with a very high-work axial stage in series with a single centrifugal stage to deliver a pressure rise of 2000 psi.

Current conceptual designs for high thrust chamber pressure space shuttle engines utilize boost pumps (low-speed inducers) to allow main pump speeds in the range of 30 000 to 40 000 rpm at low NPSH, while utilizing two or three stages to deliver a total pump pressure of 6000 psi. Current NERVA conceptual designs employ a two-stage centrifugal pump of approximately 30 000 rpm, operating at ultralow NPSP (0–2 psi). This pump will require a flow rate change of nearly 2:1 at constant pressure rise and high efficiency to provide pump-out capability to the engine, which incorporates two pumping systems in parallel.

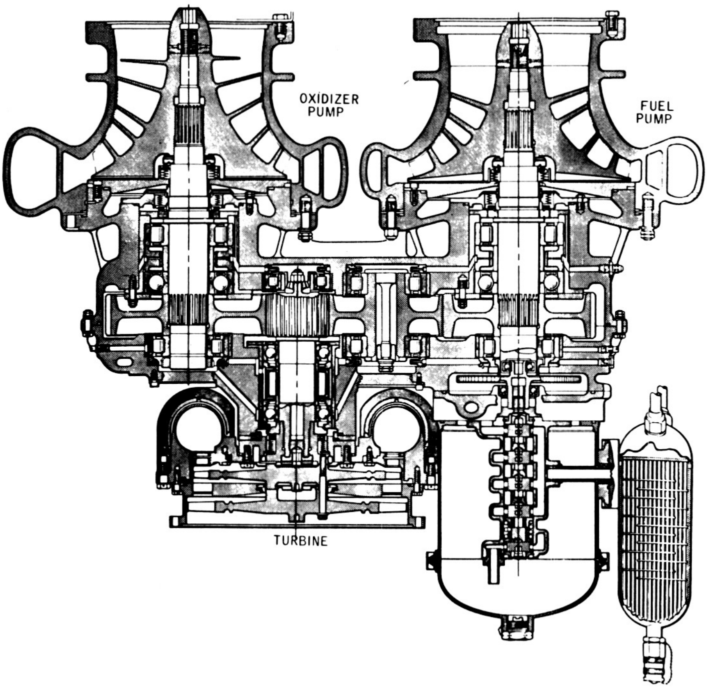

FIGURE 1a.—*Gear-box-driven system: both pumps gear-driven (Titan II).*

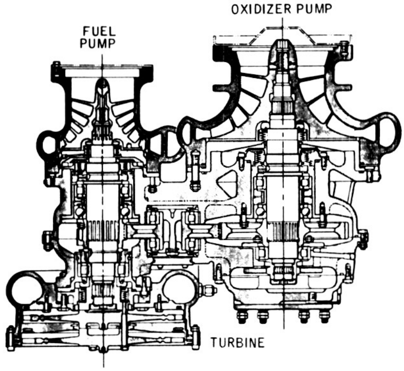

FIGURE 1b.—*Gear-box-driven system: one pump direct-driven (Titan II).*

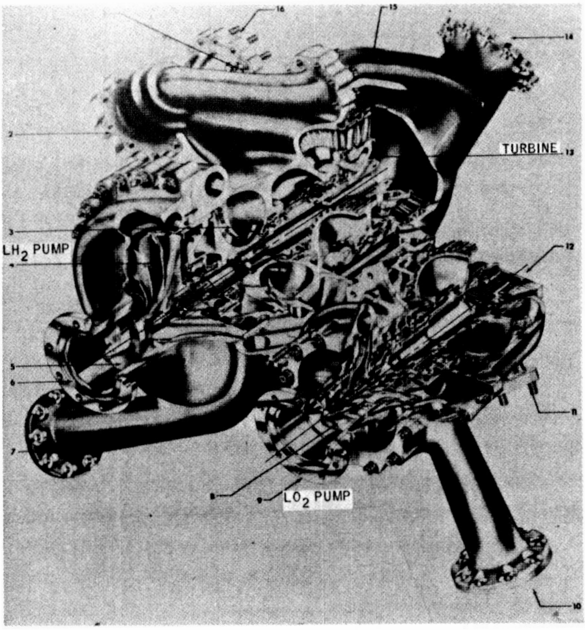

FIGURE 1c.—*Two-stage, direct-driven LH_2 pump; gear-driven LO_2 pump.*

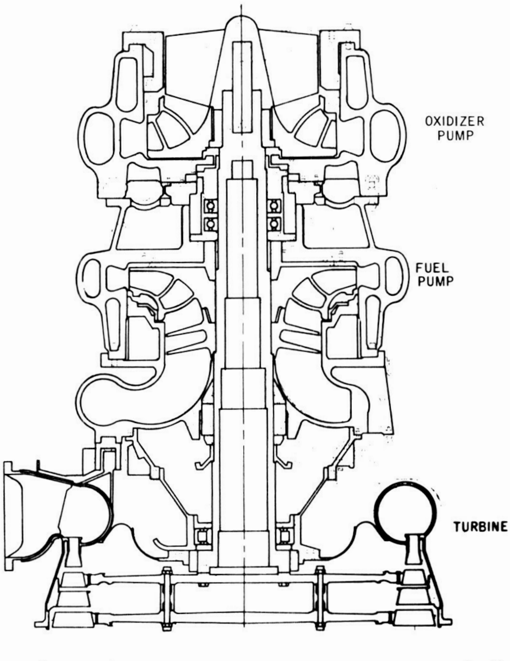

FIGURE 1d.—*In-line system: both pumps on common shaft (F-1).*

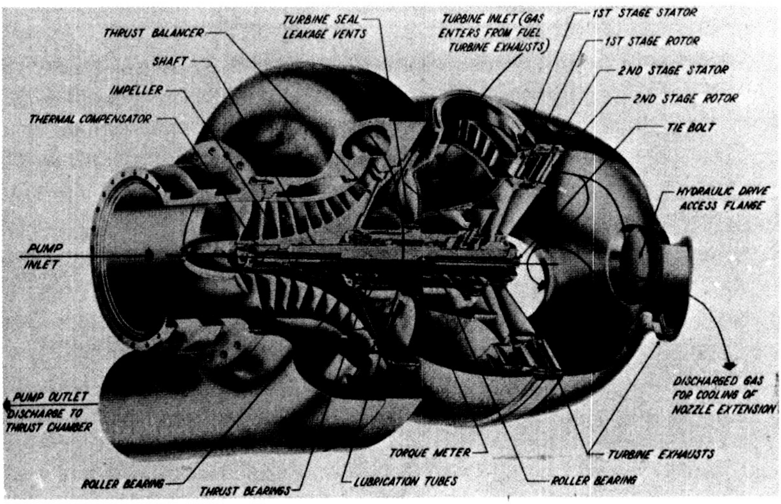

FIGURE 1e.—*Direct-driven LO_2 pump with its own turbine (M-1).*

Shaft Speed

As in most aerospace turbomachinery applications, the most important variables are speed and efficiency. Because speed influences the efficiency as well as turbopump weight, this parameter dominates the pump design. Widely varying shaft speeds have been utilized—3500 rpm to about 40 000 rpm—with the very high speeds used for the low-density hydrogen pumps. The more recent typical consideration of low-speed preinducers (separate stages preceding the main pump inlet which were previously used only on Centaur) has removed the NPSH limit on speed for current conceptual designs, resulting in the speed limit being determined by allowable stresses, gear and seal velocities, or optimum specific speed.

The $N(Q)^{1/2}$ term in the specific speed expression, $N_s = N(Q)^{1/2}/H^{3/4}$, can be used as a measure of relative speed for pumps of differing flow rates. Table I contains such data for selected rocket pumps.

The conventional suction specific speed parameter,

$$S = N(Q)^{1/2}/(\mathrm{NPSH})^{3/4},$$

is used as the index for cavitation performance of pumps. The values achieved with rocket pumps of the past 20 years have evolved from about 20 000 to 40 000. A typical empirical curve for achievable suction specific speed as a function of design inlet flow coefficient $\phi_{1t} = C_{m1}/U_{1t}$ is shown in figure 2, which also depicts a theoretical curve as derived by Brumfield (ref. 1).

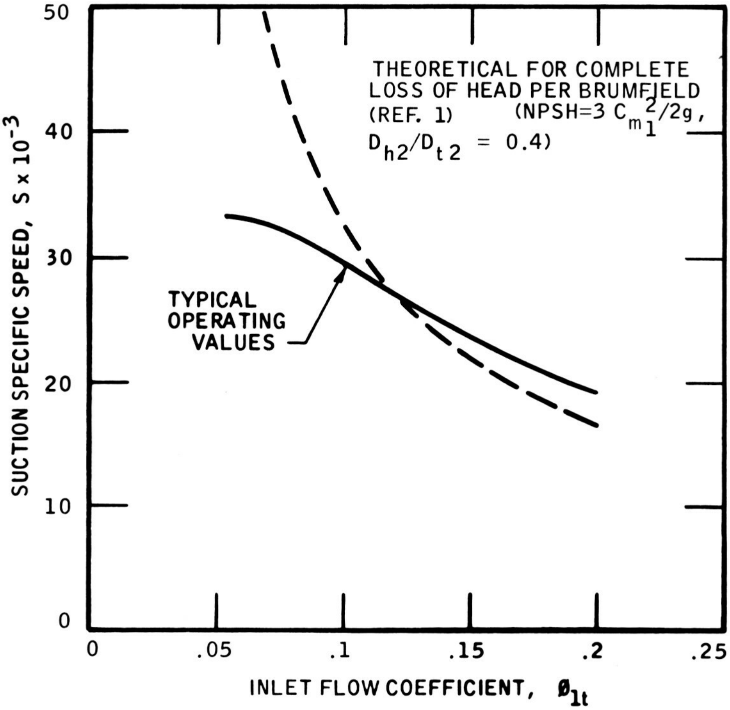

FIGURE 2.—*Specific speed as a function of inlet flow coefficient.*

The cryogenic fluids, particularly hydrogen, exhibit a "sub-cooling" effect, with the fluid having a lower effective vapor pressure because of cooling as the fluid is accelerated and vaporized in the blade passages. With liquid hydrogen, it is possible to operate without significant performance loss while ingesting two-phase flow with up to 15 percent vapor volume at the pump inlet. The extent of the subcooling effect appears to be limited by heat transfer from the bulk liquid to the vapor cavity on the blade surface. The pump speeds that are obtainable with hydrogen are apparently limited by sonic or "choking" flow considerations. Recent test results (ref. 2) indicate that with boiling hydrogen (zero quality) in a tank at 41°R, pump $N(Q)^{1/2}$ values of 2.1×10^6 are possible, provided a close-coupled, low-loss inlet system is used. At higher temperatures, higher speeds can be achieved because of additional fluid subcooling effect; conversely, lower temperatures require lower speeds. Liquid oxygen and the multi-component, mineral-base fuels (RP–1 and similar fuels), while not cryogenic, exhibit the same characteristics to a much lesser extent.

Pump Efficiency

Figure 3 shows typical design efficiency curves for rocket pumps as a function of stage specific speed and impeller size. Impeller size has been used rather than volumetric flow rate (as in the Worthington "experience" curves, ref. 3) because efficiency appears to be dependent on size for rocket pumps and because the higher $N(Q)^{1/2}$ values of rocket pumps result in smaller sizes than for a typical fixed-station pump of comparable flow rate. The inferior rocket pump efficiencies, particularly at the higher specific speeds, are attributed to the higher suction specific speeds and relatively large inlet diameters (poor geometry) of rocket-pump impellers.

Configuration Selection and Special Design Considerations

No basically new pump configurations have been used for rocket-pump design. Differences from fixed-station pumps have generally been more apparent in the values of design parameters used, due to the special considerations of the application, such as drive system and envelope limits, and ducting arrangements.

The major types of impeller and housing concepts used for commercial systems have all found use in rocket-pump applications, including the

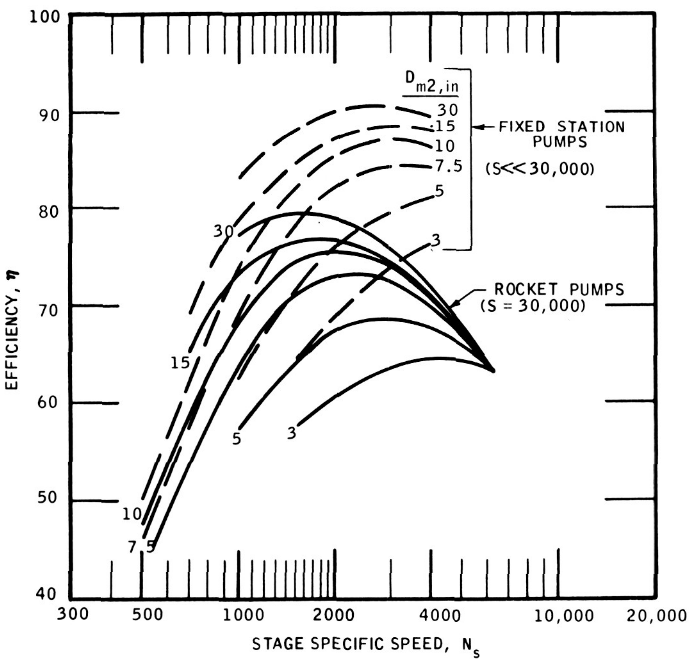

FIGURE 3.—*Efficiency as a function of stage specific speed.*

double shrouded impeller, the open impeller, the diffusion housing and the single volute, the double volute, and volute housings with various internal guide vane and splitter arrangements. Radial- and axial-flow inlet housings have been used, as well as the elbow and divided flow (or "baby pants") types of radial-flow inlet housings. The only multistage concepts have been for hydrogen applications. The Centaur (RL–10) pump is an example, the two stages being mounted with the impellers back-to-back with volute housings and an external, interstage crossover pipe.

The type of impeller and housing have often been "designer's choice" because the relative merits of each from a hydrodynamic or mechanical standpoint have not been firmly established and probably will not be in the near future. Based on one-dimensional analysis, it would appear that the open impeller is more efficient at the lower specific speeds, while the shrouded impeller is more efficient at the higher specific speeds. The same can be said of diffusion housings or vaneless volute housings (especially the double-discharge type) when compared to volute housings with internal guide vanes and impeller backvanes compared to wear rings or labyrinths.

Mechanical-design choices also are not clearly defined. A major problem area with the high-head-rise hydrogen stage is centrifugal stress in the high-tip-speed impeller. The open impeller has advantages over the shrouded impeller from a fabrication and stress standpoint, but the specific value of stage head rise at which the shrouded type can no longer be used is not easily defined. A choice the designer faces in the use of vaned housings is whether to let the vanes carry a significant portion of the separating force or to have a stiffer outer wall support the loads. The design with load-carrying vanes is lighter; however, the stress concentrations in the region of the vanes and the housing tongue are not easily defined for such a complex structure, and consequences of a miscalculation can be serious.

Hydrodynamic Design

Impeller/inducer. Conventional pump design practice has been used as the primary design technique. The one major exception is the use of empirically derived criteria such as that of figure 2 to select inlet geometry. Inducer concepts in which the blading is continuous with the main impeller, as well as concepts with separate blading incorporating a gap between the inducer and impeller, have been used. The separate-blading type exhibits better performance at higher suction specific speeds.

One-dimensional techniques are used for the flow analysis, except for the use of axisymmetric stream-filament solutions for the flow in the impeller to position the shrouds or design the through-flow profile. The results of such calculations are also used to estimate the blade surface velocity distributions, making simple assumptions for the blade-to-blade velocity distribution.

The selection of the impeller exit geometry is also based primarily on empirical data such as figure 4, which shows representative values of impeller head coefficient, $\psi = \Delta H / U_2^2 / g$, as a function of specific speed at the best efficiency point.

The "area-moment" method or cascade solutions (as originally proposed by Stodola, (ref. 4) are used to estimate impeller exit-flow deviation effects. The "rule-of-thumb" that the number of impeller vanes is one-third of the discharge blade angle has been applied with fair success for rocket-pump impellers, at least at the impeller discharge. The relatively low inlet blade angles dictated by suction performance requirements result in a small number of blades at the inlet. The use of partial or splitter blades, starting one-third to three-fourths through the impeller, provides proper solidity to the high-head generation region. To simplify

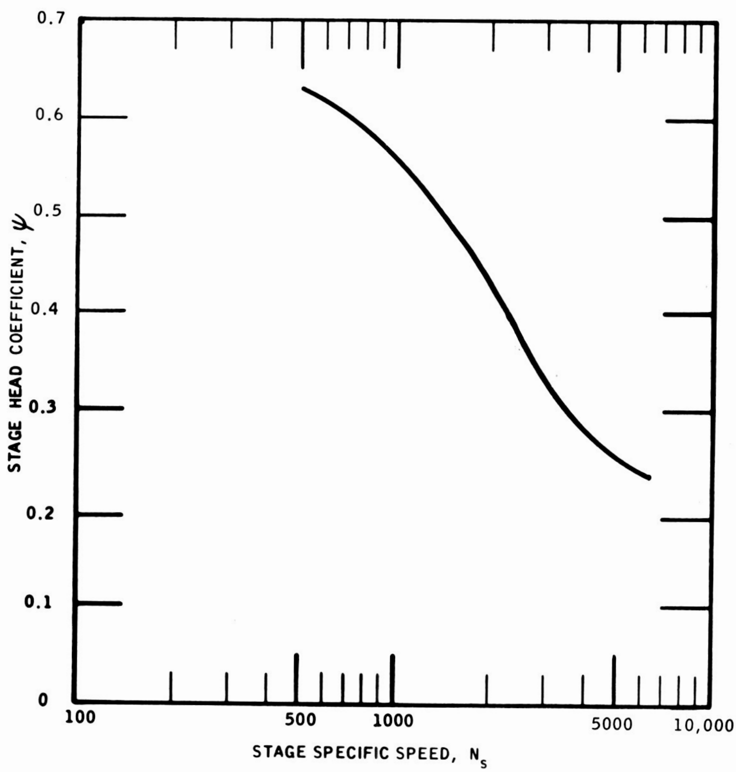

FIGURE 4.—*Stage head coefficient as a function of stage specific speed.*

machining of impellers, "straight-element" type blades are primarily used so that side-milling may be employed.

To calculate leakage through wear rings and labyrinths, methods developed for steam and gas turbines are used. For open impellers, empirical test data (such as that of fig. 5) are used to estimate the effects of clearance on pump head rise and efficiency. Figure 6 indicates the types of open impeller clearances used for rocket pumps as a function of specific speed.

Housing. One-dimensional analyses are also employed for housing design. Figure 7 shows some of the experience in terms of housing-base-circle/impeller-discharge-diameter ratio as a function of specific speed, with values of housing and diffuser-throat-velocity/base-circle-velocity ratio shown in figure 8.

For some very low specific-speed applications, very small volute and diffuser angles have been used, with the absolute flow angle of the base

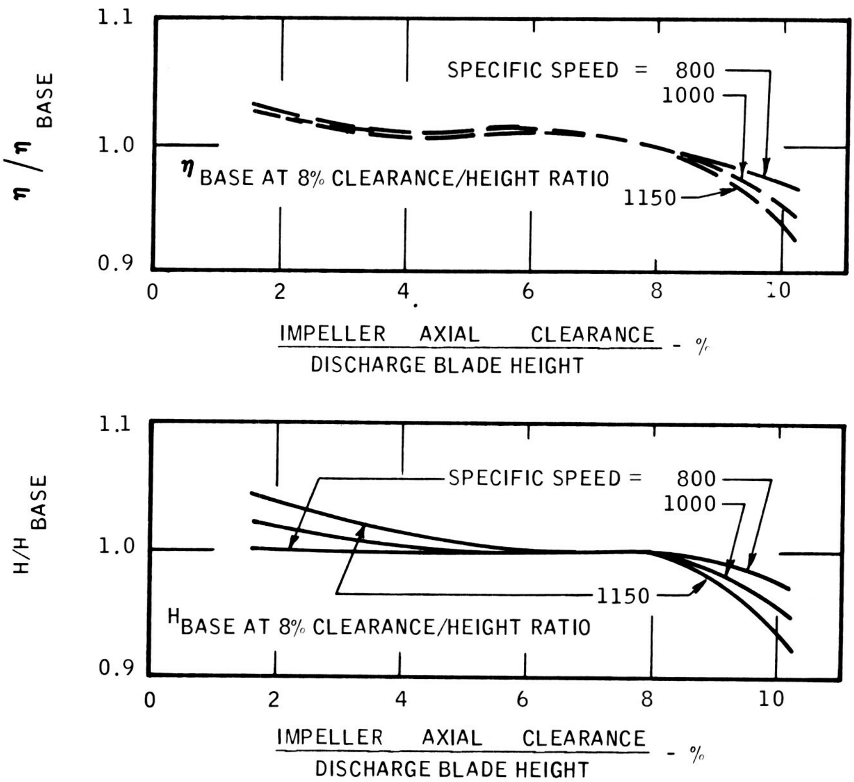

FIGURE 5.—*Head rise and efficiency as functions of clearance to blade-height ratio.*

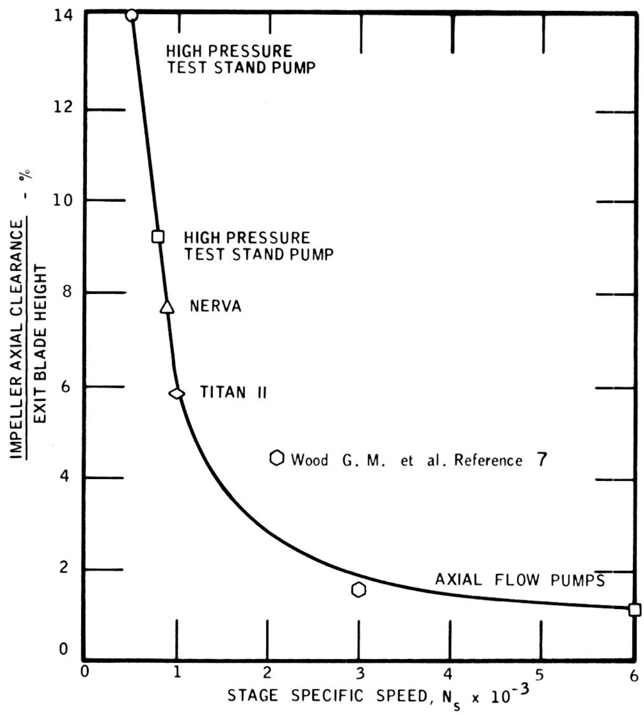

FIGURE 6.—*Clearance to blade-height ratio as a function of stage specific speed.*

circle as low as 4 or 5 degrees. Diffuser vane systems with inlet angles as low as 6 degrees have been used.

Both the cascade and channel flow approaches have been used for vane system design. For the very-low-angle vane systems, the inlet portions of the vane have a mean camber line which is close to a log spiral. This is the counterpart of the constant lead helical blade in an axial-flow system, and similar fluid-incidence/vane-angle relationships seem to apply at the minimum loss, with the minimum loss incidence angle about 40 percent of the vane angle.

Vane proportions have usually followed values given by Stepanoff (ref. 3) for chord length to spacing. With the increasing use of fabricated housings and higher strength materials, and since chord length has also been governed in many cases by structural considerations, it can be expected that somewhat shorter chord lengths and larger members of vanes may be used in the future.

Prediction of performance. One-dimensional methods supplemented with empirical data are usually used for off-design performance estimates. Simple approximations can be made of the off-design losses through

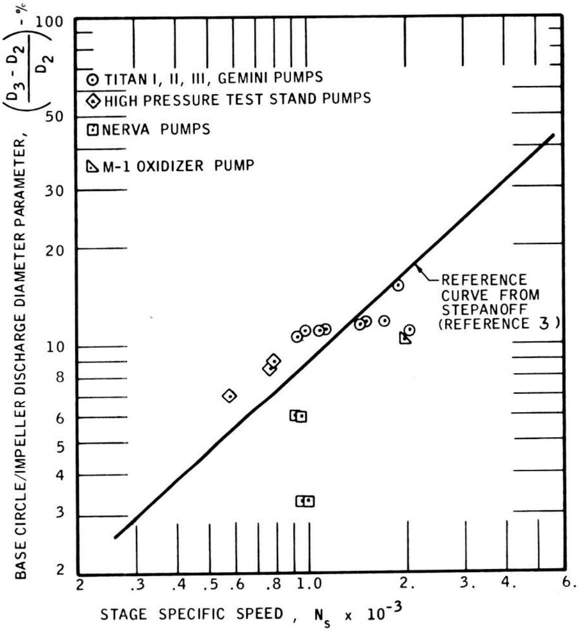

FIGURE 7.—*Housing/impeller diameter ratios versus stage specific speed.*

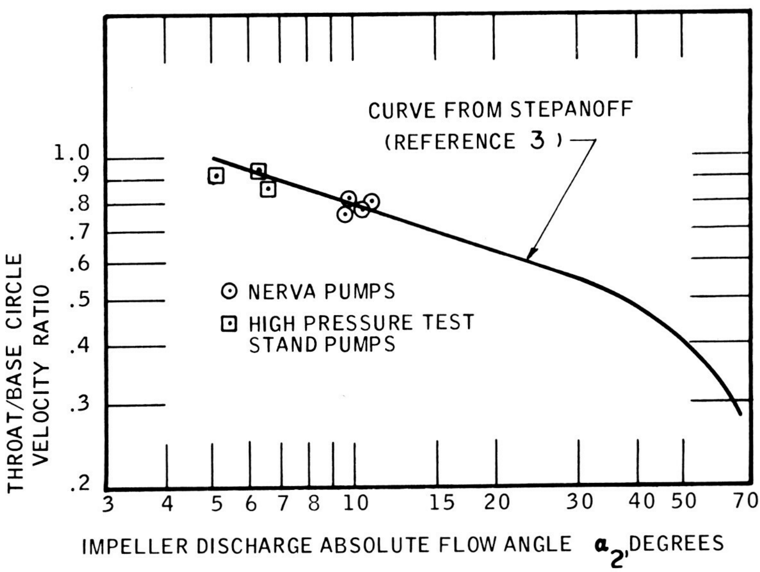

FIGURE 8.—*Housing velocity ratio versus impeller flow angle.*

blade rows, such as the assumption that losses are equal to the velocity-head difference between the off-design and the design inlet tangential velocity components.

With the advent of throttleable engine applications and redundant (two-pump) feed systems, more accurate predictions of the slope of the head-flow curve and the location of the gross flow separation point are required to ensure fluid-dynamic stability. The cascade-analysis approach appears to be more accurate than the channel-flow technique. For low-angle diffuser vanes, it has been found that the gross separation point occurs at fluid-incidence/vane-angle ratios similar to the corresponding point for the constant-lead-helix axial-flow blade; i.e., a fluid-incidence angle approximately 60 percent of the blade angle.

Axial and radial thrust. Procedures for estimating axial and radial thrust forces follow those used for fixed-station pumps. Single-discharge, double volutes with long internal splitters have been used to minimize radial thrust forces.

The most accurate method of estimating axial thrust has been to use local head-coefficient distributions as obtained for similar designs. For the higher-speed hydrogen-pump designs, it has been necessary to use single or double acting axial-clearance-compensating thrust pistons or discs because of the relatively low axial load capacity of the high-speed liquid-hydrogen cooled bearings. As pump speeds increase, it may be necessary to use such balancers for the lower-speed dense-propellant pumps.

Mechanical Design

Structural factors of safety. Since minimum weight is a prime design objective, the mechanical design of rocket pumps is based upon small structural factors of safety. The factors currently being used with the available analytical techniques are in the range of 1.10 to 1.20 for tensile yield strength and 1.2 to 1.5 for ultimate tensile strength, applied at a speed and associated pressure about 10 percent above the maximum speed to be expected in engine operation. This allows transient excursions and some adjustment in engine requirements.

Methods of analysis. The burst speed margin of the impeller disc, deflections of the disc and blade as they affect fits and clearances, blade stresses (including pressure, vibratory, and thermal loads), and housing stresses are the principal structural limits. Disc and housing stresses are calculated using finite element techniques on the digital computer. The analysis includes the magnitude and distribution (nonuniformity) of material strength and ductility and accounts for pressure, thermal, centrifugal or rotational (impeller), and flange (housing) loads, as well as stress concentrations. The criteria are acceptable tangential impeller stresses, acceptable deflections in the impeller and housing, and adequate safety margins on yield or ultimate strengths in the housing.

Blade stresses are calculated based upon centrifugal and steady-state pressure loads, cyclic pressure loads, and the effect of operation close to blade natural frequencies. Structural adequacy is determined by a comparison of calculated stresses with the allowable envelope of the modified Goodman diagram, which indicates combinations of mean and alternating stress for a given cyclic life and material.

Materials and fabrication techniques. Low-tip-speed impellers (<1000 ft/sec) are usually cast or machined (cam-generated or pattern-duplicated) from aluminum (356, Tens 50, 7075, 7079) or corrosion-resistant alloys (Monel, K-Monel).

Impellers for high-tip-speed applications (hydrogen pumps) are machined from high-strength forged material, usually alpha titanium (5 Al–2.5 Sn, ELI). Figures 9 and 10 illustrate typical unshrouded, machined impellers.

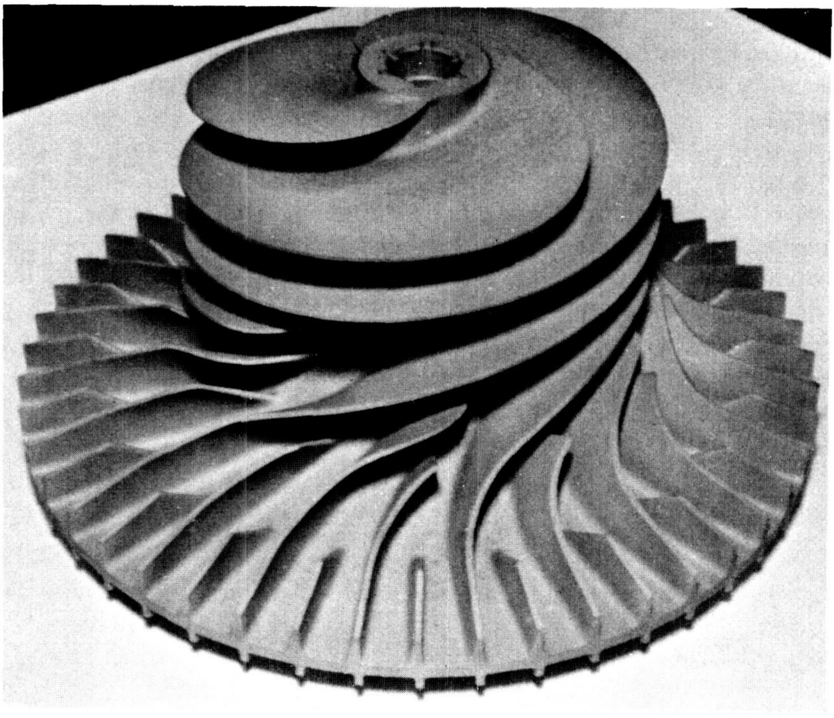

FIGURE 9.—*Typical unshrouded LH_2 pump impeller (NERVA I).*

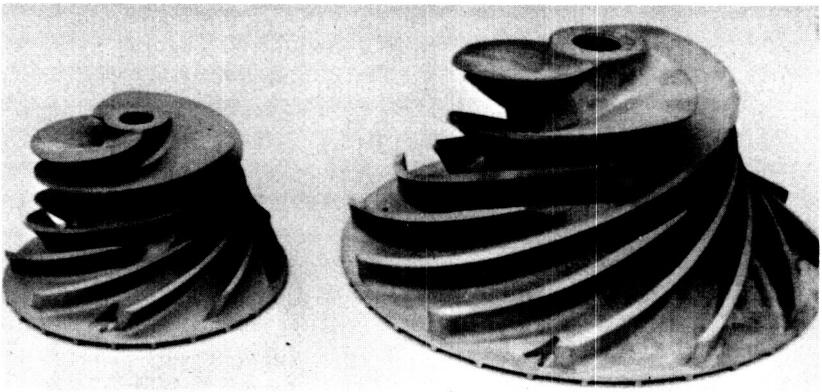

FIGURE 10.—*Typical conventional density propellant impellers* (*Titan* II).

Shrouds have been used on both the cast and machined configurations; in the latter, the shrouds are integral with the blades and are machined from both the inlet and exit ends by use of milling cutters or the EDM (electrical discharge machining) process.

Operating and burst tip speeds of at least 2000 ft/sec have been achieved with high-strength forged alloys. The ratio of strength to density favors titanium over the higher-strength Inconel and the lighter-density aluminum.

Housings are fabricated in two basic forms, either as castings or as weldments of formed elements. Both of these methods have been used for housings with and without diffuser vanes. The common materials include aluminum (Alloys 356 or Tens 50) and 300-series stainless steel for castings; and 300-series stainless, Inconel 718, and alpha titanium (5 Al–2.5 Sn) for the welded/wrought construction. The need to retain ductility for stress redistribution is an important factor, particularly in the cryogenic applications. The high-pressure systems are currently utilizing the high-strength alloys to reduce weight. Figures 11 and 12 illustrate typical housings.

Weight. The weight of rocket pumps is difficult to correlate because of a variety of design approaches and applications; since pump weight alone is not particularly meaningful, only turbopump weights are readily available. The effects of pump-shaft torque requirements on turbopump weight are shown in figure 13. The data tend to fall on a single line, regardless of whether the turbine is driving one or two pumps. The exceptions are the hydrogen pumps, which have significantly less weight for a given shaft torque.

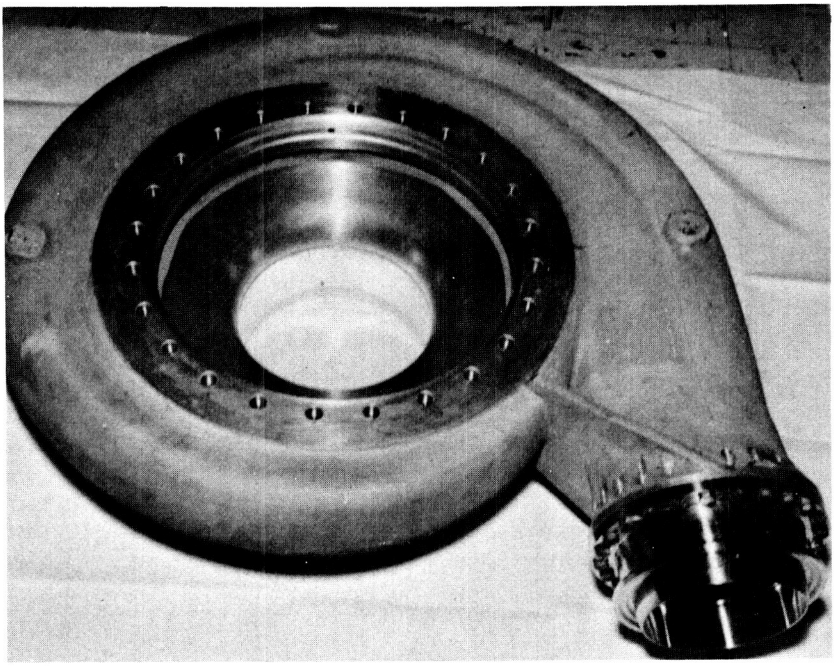

FIGURE 11.—*LH_2 pump housing with diffusers (NERVA I).*

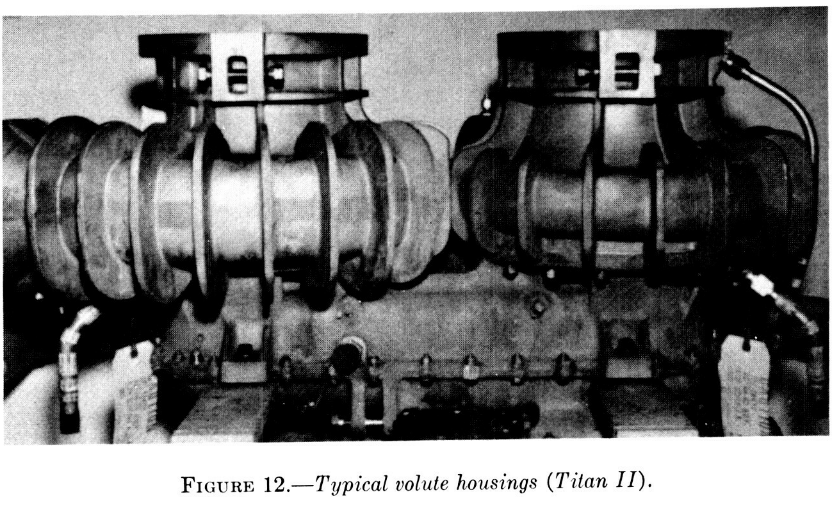

FIGURE 12.—*Typical volute housings (Titan II).*

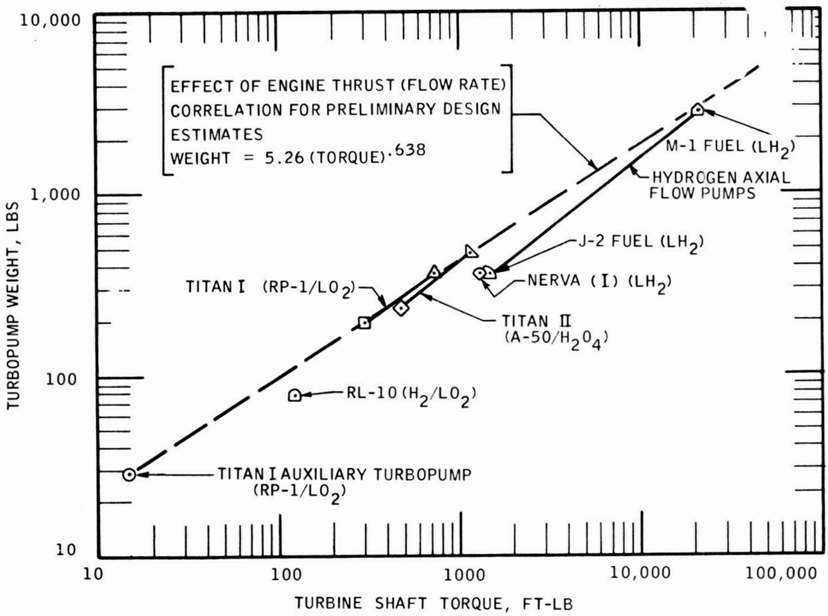

FIGURE 13.—*Turbopump weight as a function of turbine torque.*

TESTING AND DEVELOPMENT PROCEDURES

The development of rocket engine pumps has been somewhat equally concentrated between improvements of performance and mechanical integrity. Earlier systems favored achievement of safe operation over refined performance. However, the desire for better performance achieved equal status in later systems, once mechanical-design solutions were evolved for the earlier mechanical problems such as explosions, pump rubs, excessive axial and radial thrust, and impeller and housing failures. A new cycle of mechanical integrity emphasis may now be imminent with the advent of the long-duration, high-pressure cryogenic systems with stringent reliability requirements.

In contrast to the commercial pump application, rocket engine pumps are rarely an adaptation of an existing design due to significant changes in requirements from system to system. Because of this, all such pumps have required a development program. This data accrual/design adjustment cycle utilizes data from three sources: (1) component (pump only) tests, usually electric-motor driven at full or partial speed, with actual or substitute fluids (e.g., water or air); (2) turbopump (turbine-driven pump) tests at full speed, using actual or substitute fluids (water, liquid nitrogen); and (3) engine tests, at full speed with actual fluids.

The pump tests are usually of long duration and are used for mapping and wide off-design excursions without speed transient effects. Head rise and efficiency are derived as functions of flow rate and NPSH for a variety of speeds from parameters recorded at 1 to 1500 samples per second by an analog-to-digital measuring-recording system. Measurements are also made of axial thrust and shaft axial and radial displacement, as well as

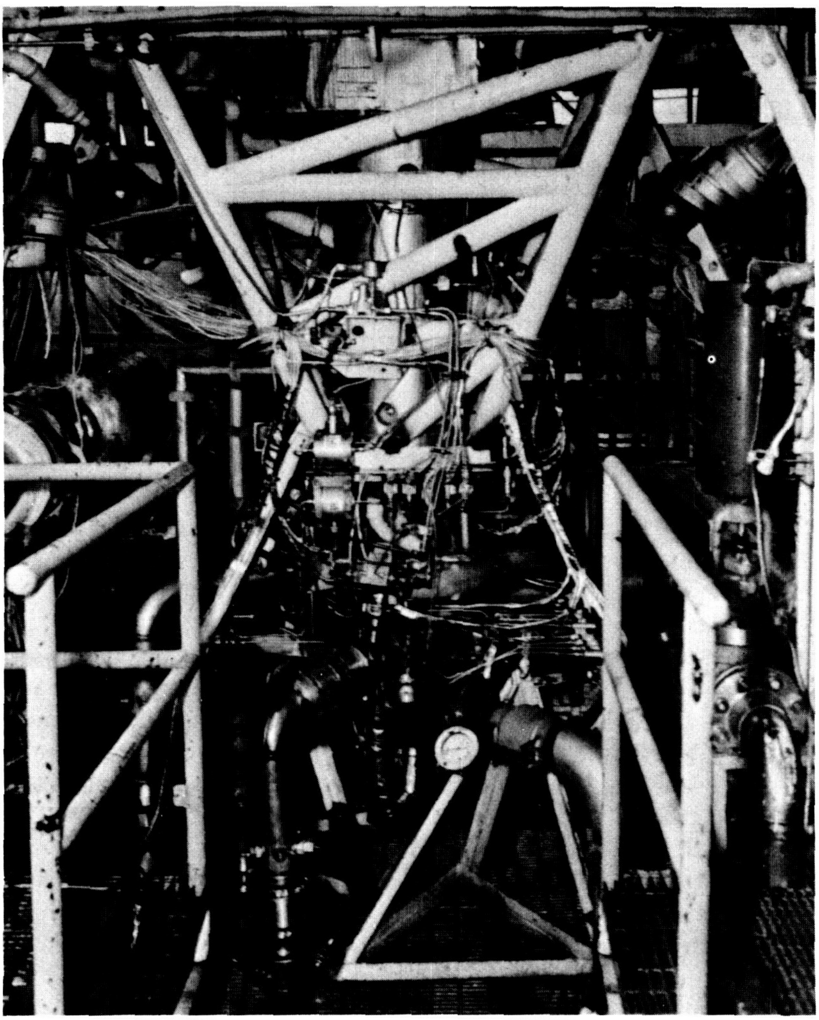

FIGURE 14.—*Typical turbopump test installation.*

velocity surveys, high-frequency pressure-oscillation surveys, and vibration monitoring. The output of this testing, during which approximately 50 to 100 parameters are typically measured, is a set of pump performance characteristic maps and a description of operating behavior.

Turbopump tests, utilizing 100 to 250 measurements, confirm performance over a more limited range of operation. This type of testing concentrates on transient behavior, verifies structural integrity, and acquires operating environment data under the more severe conditions imposed by the turbopump to support additional design analysis. Figure 14 shows a typical turbopump test installation.

Engine ground and flight tests further confirm performance, but primarily confirm or adjust start and shutdown transient effects and environmental data, such as stage or gimbaling induced pump inlet velocity distortion, stage-coupled structural oscillation (POGO) effects, and malfunction shutdown flow and speed excursions.

Most of the testing, other than that done at the pump level, is considered behavioral/interaction testing, and serves primarily to identify those areas requiring additional analysis, design refinement, or additional testing. These behavioral/interaction effects have significant impact on pump reliability, affecting both performance and mechanical integrity. Since the effects are often subtle and difficult to attack analytically, a large portion of the total testing effort is devoted to understanding these effects.

Structural testing is used to a lesser degree, but serves to further the understanding of part behavior under a simulation of imposed operating conditions. The more important structural tests are (1) housing (casing) pressure-integrity hydro-tests to confirm that burst pressure is adequately high and pressure-induced distortions are acceptable; (2) impeller-blade vibration survey tests to verify that resonant frequencies which would produce vibratory stress failures lie outside operating regions; and (3) impeller or disc spin tests to verify the structural adequacy of the impeller under the required operating tip speeds.

DEVELOPMENT EXPERIENCE HIGHLIGHTS

Problems

Most rocket pump programs have suffered their share of problems, including such vivid early experiences as LO_2 pump explosions and fires. Causes were usually found to be either contamination or severe rubbing resulting in sufficient heat to actually ignite the material, as illustrated by figure 15, an example of the results. Light rubs of the impeller in the housing, caused by shaft deflection due to combined radial and axial

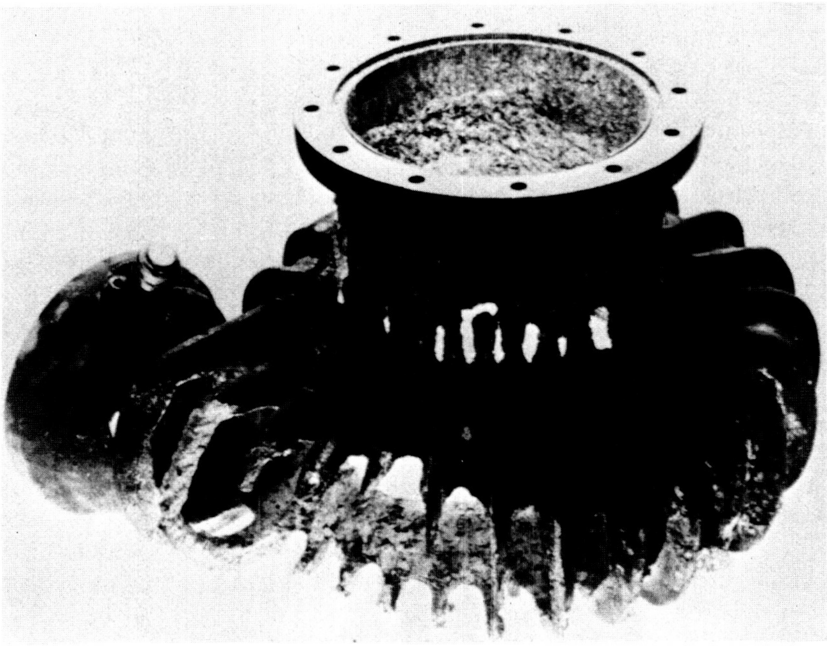

FIGURE 15.—*Results of LO_2 pump fire.*

thrust, were noncatastrophic but frustrating because of the explosion potential.

Structural failures of housings and impellers have occurred. These have most often been impeller-blade fatigue failures, usually of blade corners or tips (fig. 16 shows failure and secondary damage), and housing vane or tongue cracks at the leading or trailing edge. Housing failures were usually due to high local stress in the load-carrying tongue or vanes, together with low ductility and low strength of the cast material (fig. 17). The impeller failures have been attributed to operation at or near blade resonance and to the low stress redistribution capability of the comparatively low ductility aluminum material. The already low ductility was exaggerated by exposure to cryogenic temperatures in some of the systems. "Contamination" has also caused failures, as shown in figure 18.

A very impressive problem in a mechanical sense has been axial (end) thrust, particularly on large or high-pressure pumps. The problem rises from an inability to predict the magnitude of the small difference of the two very large pressure forces of the front and back faces of the impeller. Even with pressure survey data, errors of a few percent result in intolerable variations in net forces. Unbalanced forces of several thousand pounds have been observed on initial tests.

FIGURE 16.—*Typical impeller blade failure.*

Performance deficiencies occurred in head rise, efficiency, and low NPSH capability. The first two were the results of poor estimates and data scaling, together with improper accounting for the effects of generous clearances used to preclude rubs. Additionally, losses in the high-rate-of-diffusion impeller passages have been a reason for performance disappointments. Suction performance deficiencies occurred when pumps were operated at design (and more often at off-design) inlet flow coefficients. Mismatches of the engine operating conditions and pump performance (due to changes in both compared to initial design point assumptions) caused operation at deviations from the best efficiency point or best inlet flow coefficient.

The rapid start and shutdown transients (varying from 0.5 to 3 seconds from application of power to full speed) have caused transient axial and

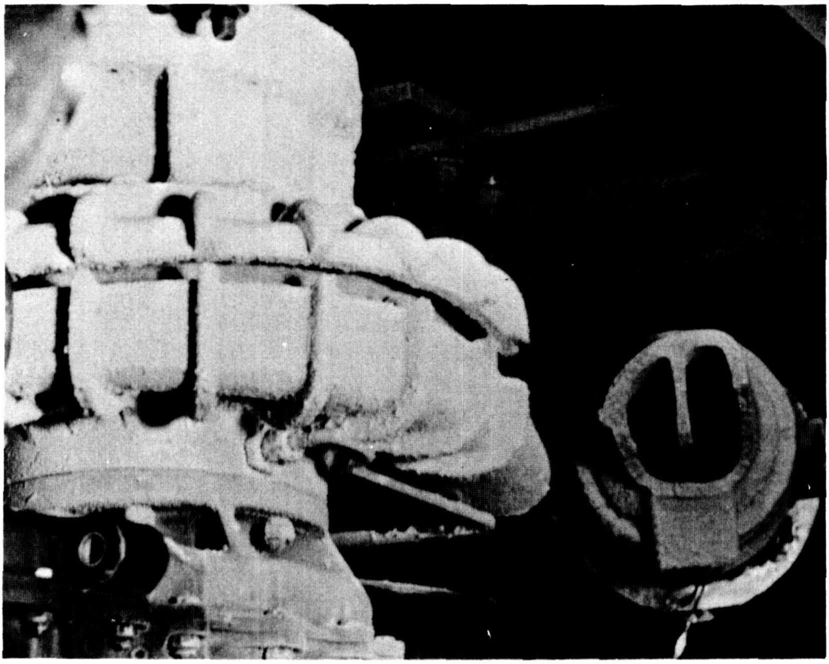

FIGURE 17.—*Pump housing rupture during cryogenic test.*

radial hydrodynamic loads and off-design fluid blade loadings, resulting in overstress and shaft deflections and rubs.

Difficulties have been experienced with cryogenic pumps in maintaining proper fits and clearances due to the large change in absolute and relative dimensions between ambient fabrication/assembly temperature and an operating temperature perhaps 500° R lower. These changes of several thousandths of an inch require detailed analysis when critical fits or clearances are selected, particularly for the transient period between ambient equilibrium and uniform operating temperature. Even the data scatter in the coefficient of thermal contraction has caused the variation of pilot diameter fits to be unacceptable.

Cavitation damage has not been a problem on operational pumps due to the short operating cycles. Some damage has been observed on both aluminum impeller blades and housing vanes; this damage has become excessive on test pumps which have been run longer than flight duration, but damage of this sort has not been observed in hydrogen pumps. Prevention of cavitation damage will be a serious consideration in the design of the currently envisioned 10-hour duty cycle systems.

During flight tests of several missiles and vehicles, conditions of longitudinal oscillation of the entire vehicles were observed. This phenomenon,

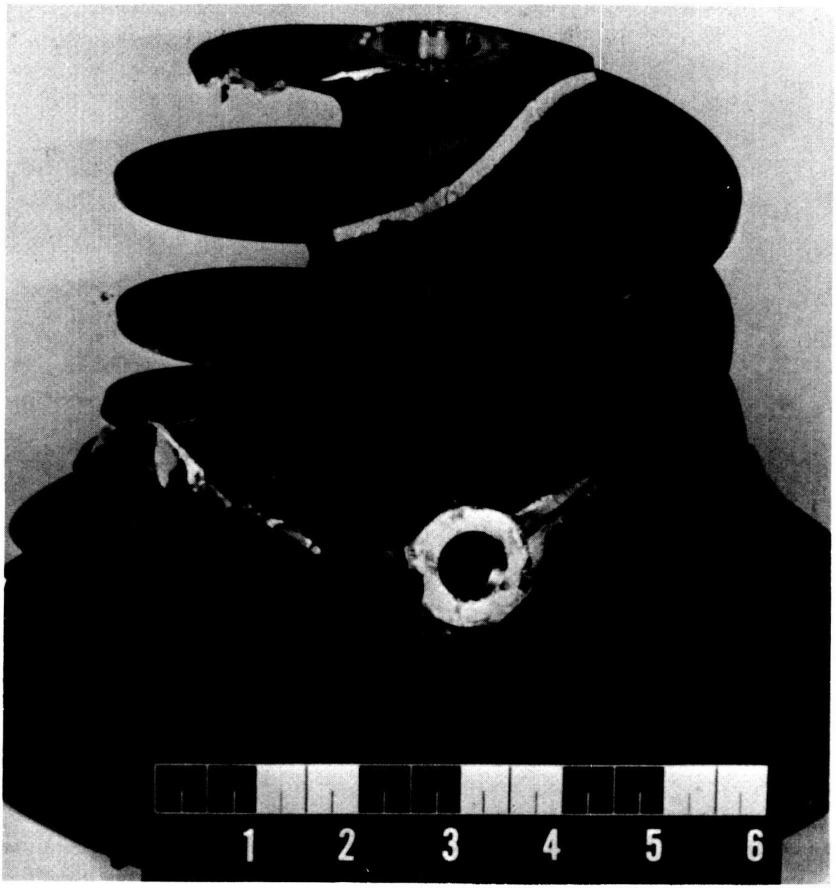

FIGURE 18.—*Results of test stand contamination.*

called POGO, is felt to be caused by a coupling of the fluid/thrust system with the structural frequency of the engine and vehicle. One of the parameters in this loop is pump "gain." Significant efforts were devoted to trying to improve suction performance (in terms of slope of the head rise versus NPSH characteristic of the pump).

Embrittlement (loss of ductility) of some materials (titanium 5 Al–2.5 Sn ELI and Inconel 718) has been recently observed in the presence of high-pressure hydrogen. This problem is not thoroughly understood but is causing concern for the highly stressed applications.

Improvements

In spite of these disappointments, successful pumps were evolved through a series of test-fail-fix steps and improvements in analysis and design techniques. When the change in difficulty of new applications did not outdistance the improvement in state-of-the-art or design capability, progress was evident.

Explosions and fires are essentially unknown in current operational LO_2 pumps. This advance has largely been the result of tremendous attention to cleanliness control; elimination of severe rubs with more rugged designs and the use of compatible and non-heat-generating materials for close clearance elements, such as wear rings, have also led to successful solutions.

Structural failures still persist. An adequate understanding of both disc and housing structural-analysis methods has been achieved largely because of the use of analytical (including finite element) techniques correlated with experimental data using strain gage and burst test information. However, blades are still a problem due to both hydrodynamic and vibratory loading. Impeller blades are virtually impossible to analyze for vibratory stresses; hydrodynamic forces are difficult to predict, and the complex blade geometry precludes translating this into a meaningful stress value. Rules of thumb based upon "this one failed—that one didn't" have been used for sizing, followed by blade shake tests and pumping tests for confirmation. Changes in environment, such as engine or turbopump testing versus pump testing, still cause surprise failures on seemingly successful configurations. However, a recently completed program (ref. 5) may furnish a tangible solution.

The lesson of low ductility has been well learned, both for housings and impellers; however, the attraction of high yield and ultimate strengths still presents a tradeoff for the use of marginal ductility (elongation between 3 percent and 10 percent). Fracture-mechanics stress-analysis techniques which treat brittle or flaw-containing material are beginning to be employed. Improvements in material fabrication methods have been noteworthy. Forgings have been produced with virtually consistent strength in all directions (no directional grain effects) or with controlled flow lines. Heat treatment intermediate through the machining process has been employed to increase strength and ductility. Aluminum castings have been much improved in quality; strength and ductility have been bettered due to careful impurity control and elaborate chilling approaches, together with the use of nondestructive and tensile tests for process development and control. Surface finish has been dramatically improved (finishes of 63 rms are now available) by the use of shell and plastic coring techniques. Welding methods (particularly electron beam) have allowed the use of both high-strength alloys (Inconel and titanium) and high stress levels, together with weight optimized shapes, in fabricated housings.

Axial and radial thrust are still problems, particularly in high-pressure systems. The axial thrust problem has become more understandable due to the accrual of test data on the pressure-area forces (measurements of pressure across front and back faces) and the net shaft thrust (force measuring sleeves in the shaft support system). The technique of wear ring diameter adjustment has been well known; back-vane trimming based upon backside vane performance (including cavitation) is reasonably well understood. Most new hydrogen pumps have abandoned thrust bearings in favor of thrust balancers.

Radial thrust remains unpredictable, particularly at wide off-design flow rates. The use of diffuser or dual exit housings has reduced the effect. For very close-clearance systems, pressure surveys from which symmetry and radial force can be inferred are useful as a development technique.

Performance improvements have been realized primarily in the ability to pump at low NPSH values. This has been most obvious in hydrogen pumps and is due not so much to particular design improvements as to the realization of the beneficial thermodynamic character of hydrogen and its somewhat homogeneous nature when boiling. Ratios of NPSH to inlet axial velocity head of 1.0 for hydrogen pumps and 3.0 for other fluids are common performance achievements. However, this entire subject is the scope of other papers and is too complex to be properly discussed here. The reduction of parasitic and leakage flows, the reduction of impeller diffusion or secondary flow losses, and the improvement of surface finishes have accounted for small efficiency improvements. Stall margins have been improved by more nearly optimum selection of housing vane geometry. However, it may be seen from the previous discussion that the rocket pump is still not competitive, from the standpoint of efficiency, with the fixed-station system.

Shaft speed and pressure rise limits have been increased, due largely to improved materials and to fabrication and design techniques. Currently proposed systems will operate at shaft speeds up to about 40 000 rpm and achievement of hydrogen pressure rise of about 6000 psi (nearly 200 000 ft) in two or three centrifugal stages is predicted. Without shrouded impellers, tip speeds are acceptable with two stages; with shrouds, three are more attractive. Experimental rotors have been run at tip speeds above 2000 ft/sec at room temperature. The increased strength at cryogenic temperature allows the use of titanium impellers at higher tip speeds.

Problems associated with rapid start/shutdown and flow change transients have been reduced by a better understanding of transient conditions and more attention to design of those features affected by transients. Better understanding has been derived by improved analytical prediction techniques, using analog computers, based largely on data acquired on routine or special-purpose development tests. Especially when correlated with applicable test data, these analytical techniques can accurately predict flow rate, speed, pressure, and suction pressure transients as functions of time during anticipated engine behavior. As an example, a special test program was recently completed to acquire transient behavior data and to refine analytical models for a dual (parallel) pumping system to prove the "pump-out" capability during highly transient operation.

Temperature effects have been adequately controlled, provided a transient heat transfer and stress/deflection/fit-clearance analysis is performed. This rather sophisticated analysis is required to preclude excessive thermal stresses and to prevent loss of critical fits or clearance (impeller-to-housing or thrust balancer-to-housing).

Achieving the elimination of cavitation damage for long duration applications remains to be proven. Initial solutions involve the use of high-strength alloys rather than aluminum, particularly for the higher-density fluid applications.

A tremendous amount of work has been devoted to elimination of POGO. The reader is referred to the literature for an overall view of the problem and its solutions.

Hydrogen embrittlement studies of high-strength alloys are now the subject of vigorous experimental investigation to determine those combinations of material, pressure, temperature, and time that cause embrittlement or reduction of strength, as well as the design values of mechanical properties that can be used. Reference 6 presents an excellent bibliography on this subject.

Testing techniques and test facilities have been improved, particularly those used for turbopump development, where a large portion of cryogenic pump development is conducted. Improvements have been mostly in the form of control systems that allow "pump designer type" data to be acquired, compared to what was formerly almost a single data-point test capability. Automatic speed and Q/N controllers have been used very successfully for pump mapping. Moderate success has been obtained with control of NPSH by means of a computerized "on-line" computation of NPSH from measured values of static pressure, temperature, and flow rate (convertible to velocity head) and control of static pressure to effect the desired NPSH. It must be noted that these are very complex facilities (as shown in figs. 19 and 20), utilizing computers for both control and data accrual-conversion, electronic control systems, and an operational crew of 30 to 60 highly skilled engineers and technicians.

CONCLUDING REMARKS

The most significant influence on the evolution of the rocket engine pump has been the evolution itself. The wide variety of design requirements from one application to another has caused *nearly every system to* *be a new product.* To maximize the performance of new designs (products), each is pushed to or beyond existing experience limits so that *improvements in the state-of-the-art are projected concurrently with the pump development.*

The wide range and combination of flow rate, head rise, and fluid properties virtually prevents accurate scaling of either performance predictions or mechanical designs; in other words almost no adaptation of existing designs has been possible. As a result, no ready set of designs or adaptable hardware has existed, and most new pump programs have gone through the entire sequence of concept selection, design, development, and qualification prior to production. This trend is expected to continue. However, either quicker product development or improved performance and reliability or both can be expected to result from the design techniques developed during the further evolution of rocket pumps and from the continued attention to the development of pump technology supporting the mainstream engine programs.

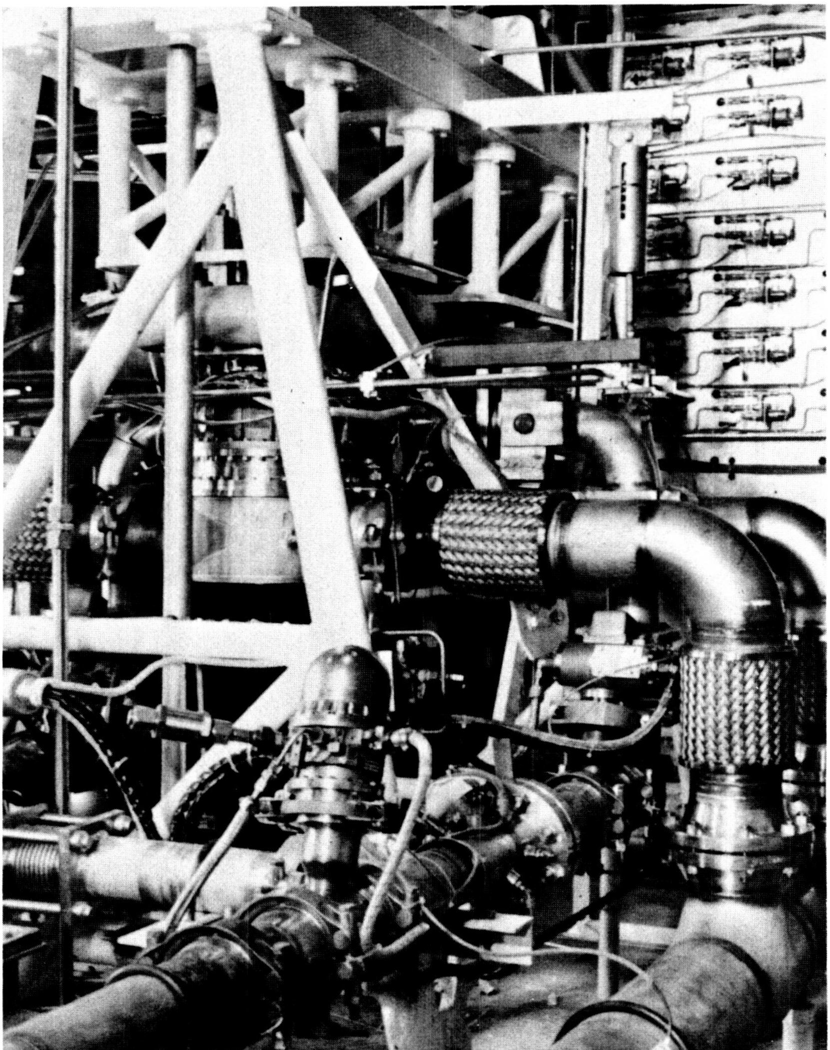

FIGURE 19.—*Complexity of turbopump test installation.*

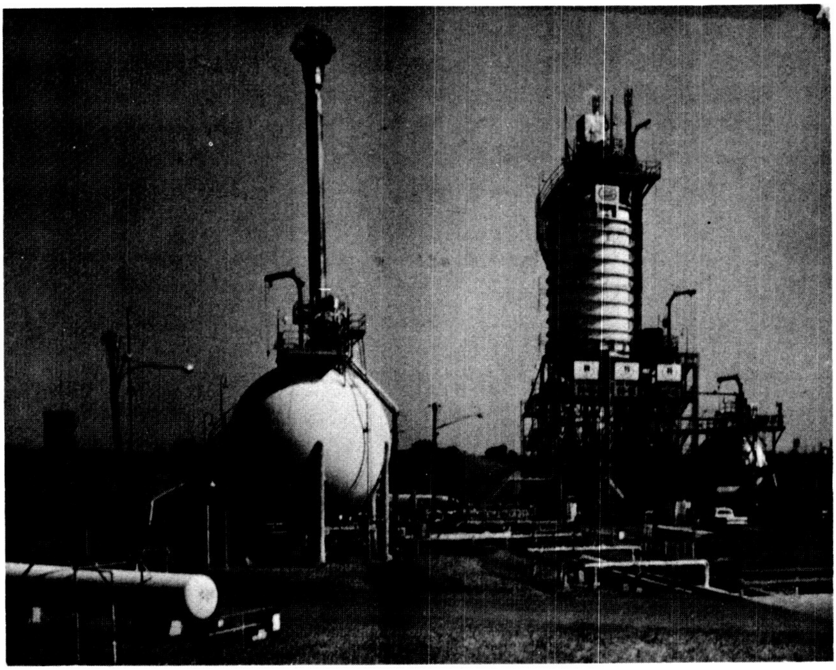

FIGURE 20.—*Typical cryogenic turbopump test facility.*

LIST OF SYMBOLS

C	Fluid absolute velocity, ft/sec
D	Diameter, bore in mm for bearings and inches for pump dimensions
F	Load, lbs
G or g	Acceleration due to gravity, ft/sec^2
H or ΔH	Pump stage head rise, ft
N	Pump speed, rpm
NPSH	Net positive suction head (fluid total head minus vapor head), ft
NPSP	Net positive suction pressure (fluid total pressure minus vapor pressure), psi
N_s	Specific speed, $NQ^{1/2}/H^{3/4}$, rpm (gpm)$^{1/2}$/(ft)$^{3/4}$
ΔP	Pressure differential, psi
Q	Pump volumetric flow rate, gpm
S	Suction specific speed, $NQ^{1/2}/NPSH^{3/4}$, rpm (gpm)$^{1/2}$/(ft)$^{3/4}$
U_t	Wheel or blade tip tangential velocity, ft/sec

α	Fluid absolute angle (measured from rotational direction), degrees
η	Pump efficiency, percent
ϕ	Flow coefficient, C_m/U_t
ψ	Stage head coefficient, $\Delta H/U_{2m}^2/g$

Subscripts

1	Inlet of impeller or inducer
2	Discharge of impeller
3	Vaned diffuser or volute throat
h	Blade hub
m	Meridional (axial plane) fluid component
t	Blade tip

REFERENCES

1. Brumfield, R. G., Optimum Design for Resistance to Cavitation in Centrifugal Pump. Memorandum, U.S. Naval Ordnance Test Station, Inyokern, California, February 1948.
2. *Results of Mark 25 Cavitation Tests.* Memorandum J–17–145–69, Los Alamos Scientific Laboratory, July 18, 1969.
3. Stepanoff, A. J., *Centrifugal and Axial Flow Pumps.* Second Ed., John Wiley and Sons, Inc. (New York), 1957.
4. Stodola, A., *Steam and Gas Turbines.* McGraw-Hill Book Co., Inc. (New York), 1927. Reprinted by Peter Smith (New York), 1945.
5. Barten, H. S., et al., *Study of Inducer Load and Stress.* PWA FR–3015, NASA CR–72514, April 2, 1969.
6. Walter, R. J., et al., "On the Mechanisms of Hydrogen-environment Embrittlement of Iron and Nickel-base Alloys". *Materials Sci. Eng.*, January 1970.
7. Wood, G. M., et al., Tip-Clearance Effects in Centrifugal Pumps. *ASME Trans.*, Paper 64 WA/FE 17, Ser. D, Vol. 87, December 1965.