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INSTRUMENTATION OF THE TRANSONIC TURBINE
TEST RIG TO DETERMINE THE PERFORMANCE
OF TURBINE INLET GUIDE VANES BY THE
APPLICATION OF THE MOMENTUM AND
MOMENT OF MOMENTUM EQUATIONS

PATRICK MICHAEL COMMONS

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INSTRUMENTATION OF THE TRANSONIC TURBINE
TEST RIG TO DETERMINE THE PERFORMANCE
OF TURBINE INLET GUIDE VANES BY THE APPLICATION
OF THE MOMENTUM AND MOMENT OF MOMENTUM EQUATIONS

by

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Lieutenant, United States Navy
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Submitted in partial fulfillment of the
requirements for the degree of
MASTER OF SCIENCE IN AERONAUTICAL ENGINEERING
from the
NAVAL POSTGRADUATE SCHOOL
September 1967

NTS 34211
1967
COMMONS, P.

ABSTRACT

The primary purpose of the Transonic Turbine Test Rig is to study the effects of axial and radial clearances on turbine performance. Additionally, the installation was designed to permit the determination of stator discharge velocities by the application of the momentum and moment of momentum equations.

Previous attempts to obtain stator discharge velocities by these methods have been unsuccessful. However, during this series of tests, with over 150 hours of useful operating time, the instrumentation and the data acquisition procedures were improved to the extent that the difference between stator discharge velocities obtained from a momentum analysis and those obtained from continuity were reduced to less than three percent.

Thesis by: Patrick Michael Commons entitled Instrumentation of the Transonic Turbine Test Rig to Determine the Performance of Turbine Inlet Guide Vanes by the Application of the Momentum and Moment of Momentum Equations.

ERRATA

| <u>Page</u> | <u>Line</u> | <u>Change</u> | <u>To</u> |
|-------------|-------------|---------------|--------------|
| 35 | 24/25 | exhaust-er | exhaus-ter |
| 36 | 24/25 | dynamom-eter | dynamo-meter |
| 71 | 8/9 | dynam-ic | dyna-mic |

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TABLE OF SYMBOLS

| <u>Symbols</u> | | <u>Fortran</u> |
|-----------------|--|----------------|
| a | Mean stator/rotor throat dimension (in.) | ASTAT/AROTR |
| A _{th} | Stator throat area (in. ²) | ATH |
| A ₁ | Annulus area, stator discharge (in. ²) | AAX |
| A ₂ | Annulus area, rotor discharge (in. ²) | AAXR |
| C _n | Constant in flow nozzle equation (dimensionless) | CN |
| C ₁ | Conversion Factor, 2gJ (ft. ² -lbm/sec. ² -BTU) | CL |
| C ₂ | Conversion Factor, 2gJc _p (ft. ² /sec. ² -°R) | C2 |
| C ₃ | Conversion Factor (γ -1) (dimensionless) | C3 |
| C ₄ | Conversion Factor γ/(γ -1) (dimensionless) | C4 |
| C ₅ | Conversion Factor (lbs./in. ² /in.-H ₂ O) | C5 |
| D _n | Flow nozzle diameter (in.) | DN |
| D ₁ | Pipe diameter at flow nozzle inlet (in.) | |
| F _{C1} | Axial force on the closure plate (lbs.) | CLAXIL |
| F _i | Pressure forces on the stator assembly | F1-6 |
| F _s | Axial force on the stator assembly (lbs.) | AXIL |
| g | Universal gravitational constant (ft.-lbm./sec. ² -lb.) | G |
| H | Total enthalpy (BTU/lbm.) | |
| H _e | Equivalent total enthalpy (BTU/lbm.) | |
| h | Blade height (in.) | |
| h | Static enthalpy (BTU/lbm.) | |
| h _w | Differential pressure across flow nozzle (in. H ₂ O) | DH |
| HP | Horsepower | |

| <u>Symbols</u> | | <u>Fortran</u> |
|--------------------------|--|----------------|
| J | Conversion factor (778.16 ft.-lb/BTU) | CJ |
| k_{iS} | Head coefficient (dimensionless) | XKIS |
| k_t | Factor to account for stator/rotor blade trailing edge thickness (dimensionless) | SKT/RKT |
| M_1 | Mach Number at stator discharge (dimensionless) | VMI |
| M | Moment (ft.-lb.) | |
| M_{C1} | Moment on closure plate (ft.-lb.) | CLTORQ |
| M_D | Dynamometer moment (ft.-lb.) | DYNA |
| M_S | Moment on stator assembly | TORQ |
| m_s | Mass flow rate (slugs/sec.) | |
| N | Rotational speed (RPM) | RPM |
| n | Polytropic exponent (dimensionless) | POLY |
| \bar{n} | Outward normal vector (dimensionless) | |
| P_{atm} | Manometer reference pressure (in.Hg.) | PATM |
| P_{bar} | Barometric pressure (in. Hg.) | PBAR |
| P_{hood} | Hood pressure (in. Hg.) | PHD |
| P_{hub} | Hub static pressure at stator discharge (in. Hg.) | PHUB |
| P_{noz} | Total pressure at flow nozzle inlet (in. Hg.) | PNOZ |
| P_{ref1} P_{ref2} | Manometer reference pressures (in. Hg.) | PRF1/PRF2 |
| P_{spl} | Total pressure in plenum (in. Hg.) | PSPL |
| P_{tpl} | Total pressure at stator inlet (in. Hg.) | PTPL |
| P_{tip} | Tip static pressure at stator discharge (in.Hg.) | PTIP |
| P_{14-20} | Shroud static pressures (in. Hg.) | P14-P20 |

SymbolsFortran

| | | |
|------------------|---|-------------|
| R | Gas constant for air (53.34 ft.-lb/lbm. ^{°R}) | CR |
| R _{hub} | Hub radius (in.) | RHUB |
| R _m | Mean radius (in.) | RM |
| R _{tip} | Tip radius (in.) | RTIP |
| r* | Theoretical degree of reaction (dimensionless) | REAC |
| S | Entropy (BTU/lbm- ^{°R}) | |
| S | Blade spacing (in.) | SSTAT/SROTR |
| TCL | Temperature in Cascade Lab (°F) | TCL |
| TCR | Temperature in Control Room (°F) | TCR |
| T _{noz} | Temperature at flow nozzle inlet (mv) | TNOZ |
| T _{tpl} | Temperature in plenum (mv) | TTPL |
| t | Blade trailing edge thickness (in) | THK |
| U | Peripheral velocity (ft./sec.) | U |
| V | Absolute velocity (ft./sec.) | V |
| W | Relative velocity (ft./sec.) | W |
| \dot{w}_1 | Plenum labyrinth leakage flow rate (lbm./sec.) | FLOWL |
| \dot{w}_t | Turbine mass flow rate (lbm./sec.) | FLOWT |
| Y ₁ | Expansion factor in flow equation (dimensionless) | Y |
| Z | Number of blades (dimensionless) | |

Greek Letters

| | | |
|------------|--|-------|
| α | Absolute discharge angle (deg.) | ALPH |
| α_n | Coefficient of thermal expansion (dimensionless) | ALPHN |
| β | Relative discharge angle (deg.) | BETA |

Greek LettersFortran

| | | |
|----------|---|-------|
| β | Ratio of nozzle minimum diameter to upstream pipe dimension (dimensionless) | BETA |
| γ | Specific heat ratio (c_p/c_v) (dimensionless) | GAM |
| δ | Referred pressure ($p_{t_0}/14.7$) (dimensionless) | DEL |
| ζ | Loss coefficient (dimensionless) | ZETA |
| η | Efficiency (dimensionless) | ETA |
| θ | Referred temperature $T_{tp1}/518.7$ (dimensionless) | THETA |
| ξ | Blockage factor (dimensionless) | XI |
| ρ | Density (slugs/ft ³) | |
| Φ | Non-dimensional flow function | PHI |
| Φ_i | Labyrinth seal pressure ratio function (dimensionless) | FLOWL |
| ω | Angular velocity (radians/sec.) | |

Subscripts

| | | |
|----|----------------------------------|--------------------|
| a | Axial direction | Suffix A |
| is | Isentropic from inlet conditions | |
| r | Rotor | Prefix or Suffix R |
| s | Stator | Prefix or Suffix S |
| u | Peripheral direction | Suffix U |
| o | Inlet to stator | |
| 1 | Stator discharge | Suffix 1 |
| 2 | Rotor discharge | Suffix 2 |

1. Introduction.

The Ares Mod II turbine is a single stage axial flow power unit designed by Dr. M. H. Vavra of the Department of Aeronautics, U. S. Naval Postgraduate School, Monterey, California, for use as a power generating device in a liquid rocket propulsion plant requiring a small high speed unit.

Tests were conducted on a scale model of this turbine to determine the effects of axial and radial clearances on the overall turbine performance. The Transonic Turbine Test Rig, hereafter referred to as TTR, was modified to accommodate this model.

The TTR was originally designed to provide a means of determining stator performance by the application of the momentum and moment of momentum equations, as well as obtaining the effects of axial and radial clearances on turbine performance without the need for probe installations between the stator and the rotor.

The tests were performed jointly with Lt. J. A. Messegee, U.S.N. The effects of axial and radial clearances on turbine performance are described by Messegee.¹

This thesis is concerned with the feasibility of determining stator performance by the methods mentioned above. Additionally, it describes the test installation, the test procedures and the data reduction methods for all tests performed.

¹Messegee, J. A., Influence of Axial and Radial Clearances on the Performance of a Turbine Stage with Blunt Edge Non-Twisted Blades (Naval Postgraduate School, 1967), Sec. 6.

The author is most appreciative for the time spent by Dr. M. H. Vavra in providing encouragement and counsel. In addition a special expression of appreciation is due to Mr. J. Hammer who willingly spent long hours helping to acquire the vast amount of data necessary to complete the tests.

2. Test Installation.

The test installation is composed of three sections: the Allis Chalmers 12 stage axial compressor with its associated piping, the exhauster assembly (hood, nozzle, turbine hood, flow inlet section) described by Monroe,² and the TTR, which will be described in detail.

The Allis Chalmers compressor is the source of drive air for both the exhauster assembly and the TTR. When operated independently, the compressor discharges to the atmosphere through a discharge valve. When run in conjunction with the TTR, compressor air is discharged into a settling tank located in the test cell. The supply of compressor air to the settling tank is controlled by an electrically operated butterfly valve.

Fig. 1 depicts the piping installation for the TTR. Turbine air passes from the first settling tank into an eight in. pipe, thence through a flow nozzle into a six

²Monroe, P. A., An Investigation of the Performance and Mixing Phenomena Associated with a Supersonic Exhauster Interacting with Subsonic Secondary Flow (Naval Postgraduate School, 1967), Sec. 2.

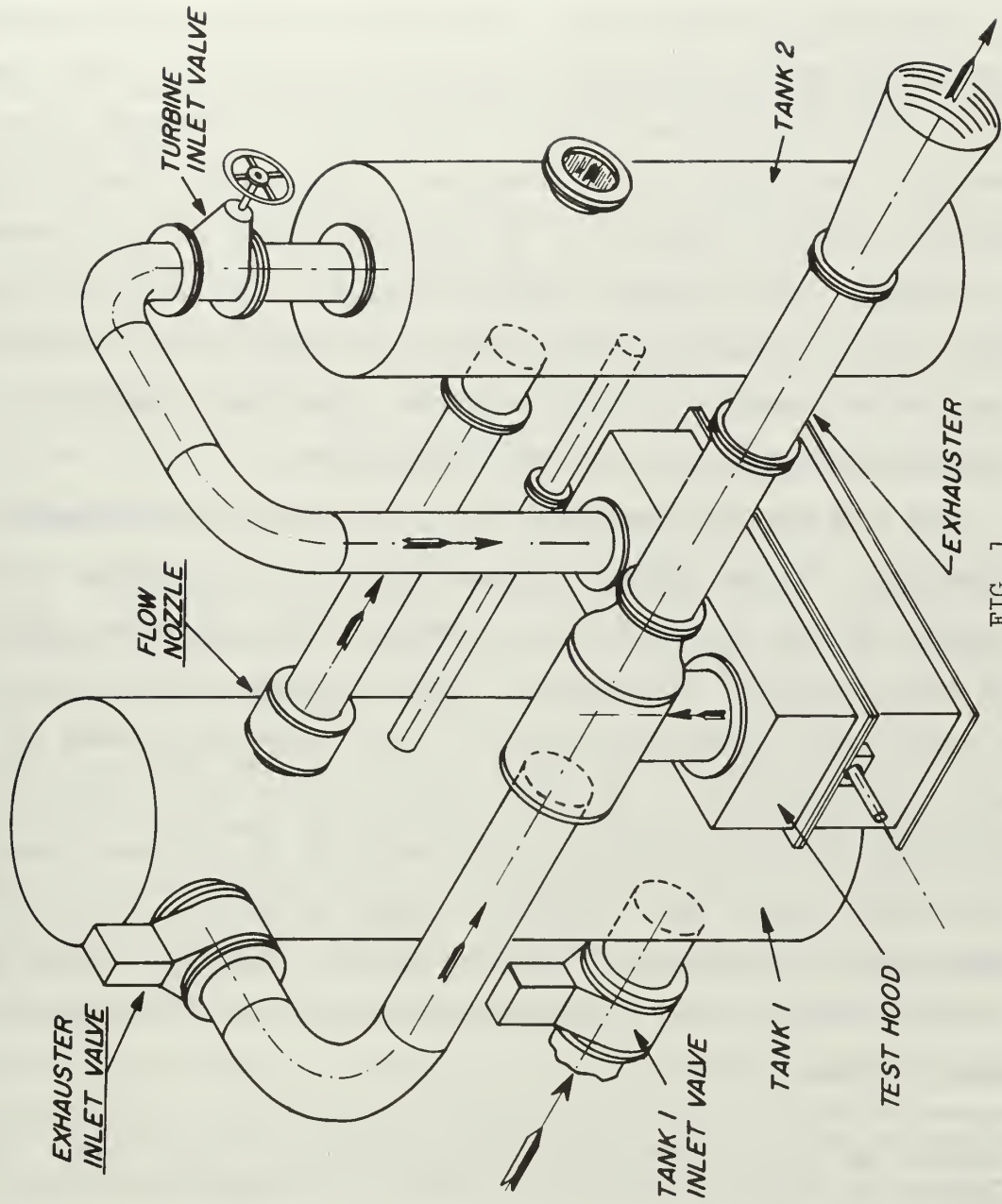


FIG. 1

PIPING INSTALLATION, TRANSONIC TURBINE TEST RIG

in. pipe discharging into a second settling tank. The flow nozzle is a non-standard flow measuring device. Therefore, nozzle calibration was necessary. This was first done by Eckert³ and then by Naviaux.⁴

Turbine air passes from the second settling tank through a manually operated control valve into a six in. pipe leading to the TTR plenum chamber. Exhauster air passes from the first settling tank through an electrically controlled butterfly valve into a six in. pipe leading to the exhauster nozzle. The exhauster nozzle converts the airflow into a high velocity low pressure stream whose momentum is sufficient to discharge the turbine flow rate from a partial vacuum inside the hood to the atmosphere.

The TTR may be operated with or without the exhauster assembly. In the latter configuration the butterfly valve leading to the exhauster is closed and the exhauster with the hood assembly is removed. The discharge side of the TTR is then open to the atmosphere. Tests were performed in both configurations. However, at pressure ratios above 1.5 operation with the exhauster is required to maintain turbine speed within an acceptable range, primarily because of dynamometer limitations. At the higher pressure ratios the turbine inlet pressure can be reduced so that the turbine power becomes smaller.

³Eckert, R. H., Performance Analysis and Initial Tests of a Transonic Turbine Test Rig (Naval Postgraduate School, 1966), Sec. 5.

⁴Naviaux, J. C., Transonic Turbine Test Rig Exhauster System Tests and Tests of a Reaction Turbine (Naval Postgraduate School, 1966), pp. 34 and 35.

The TTR is unique inasmuch as it permits the evaluation of stator performance by the application of the momentum and moment of momentum equations. This may be done by measuring the flow conditions existing ahead of, and the forces and moments acting on, the stator. The stator assembly is free floating to permit measurement of these forces and moments.

Fig. 2 depicts the support structure for the floating stator assembly. The cradle and stator assembly are supported on the test bed by two cradle supports. The cradle is attached to these supports by four 2.5 in. aluminum thrust flexures which enable the cradle with the stator assembly to move in the axial direction without changing the radial tip clearance of the turbine rotor. The stator assembly is hanging from an aluminum cross member welded to the cradle, as shown in Fig. 2. A 0.75 in. pin, extending from the center of the cross member, fits through a bracket designed to accommodate a Fafnir Self-Aligning Rod End. To its lower end is attached a specially designed bracket with two Fafnir Track Roller type bearings. These bearings ride on the inner periphery of a circular arc fitting attached to the upper portion of the stator housing. This design permits rotation of the stator assembly about its centerline and supports its weight. Two torque flexures, extending from either side of the stator housing, transmit the thrust acting on the stator assembly to the cradle while offering little resistance to torque and

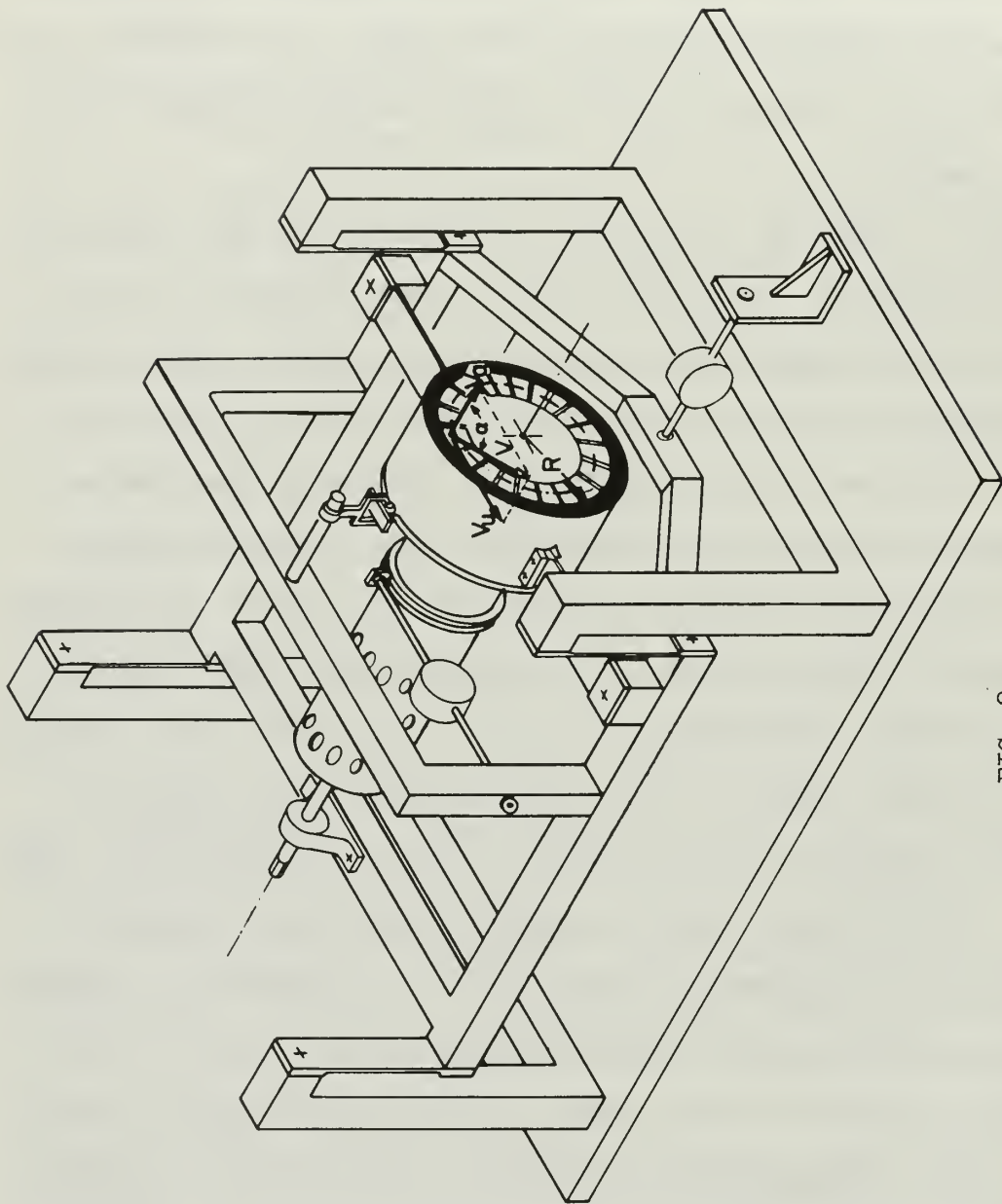


FIG. 2
FLOATING STATOR ASSEMBLY

maintain the proper alignment of the front part of the assembly. A 0.75 in. steel shaft extends from the plenum chamber. This shaft is supported by a pillow block attached to the cradle to guarantee the central alignment of the stator assembly. With the calibration stand the shaft is used also to apply known forces and moments to calibrate the force pickups used to measure the thrust and torque exerted on the stator during operation of the TTR. The force pickups used are reluctance type force capsules built by the Wiancko Engineering Company. Thrust is measured by a Model F1021 pickup with a range of 0-300 lbs. and torque by a Model F1009 pickup with a range of 0-100 lbs. The thrust pickup is mounted between the forward cross member of the cradle and a bracket affixed to the test bed. The torque pickup is attached to the weight support cross member and the plenum chamber. Fig. 3 is a scale drawing of the TTR illustrating the installation of the floating stator assembly.

The turbine rotor is attached to an overhung shaft which is supported by a separate bearing housing that is fixed to the test bed. The shaft runs in two matched sets of high-precision ball bearings that are lubricated by oil-mist. A steel quill shaft is used to connect the rotor shaft to a Vortec Air Dynamometer located outside the hood. The bearing housing may be moved axially to change the axial clearances between rotor and stator bladings from about zero to 1.5 in.

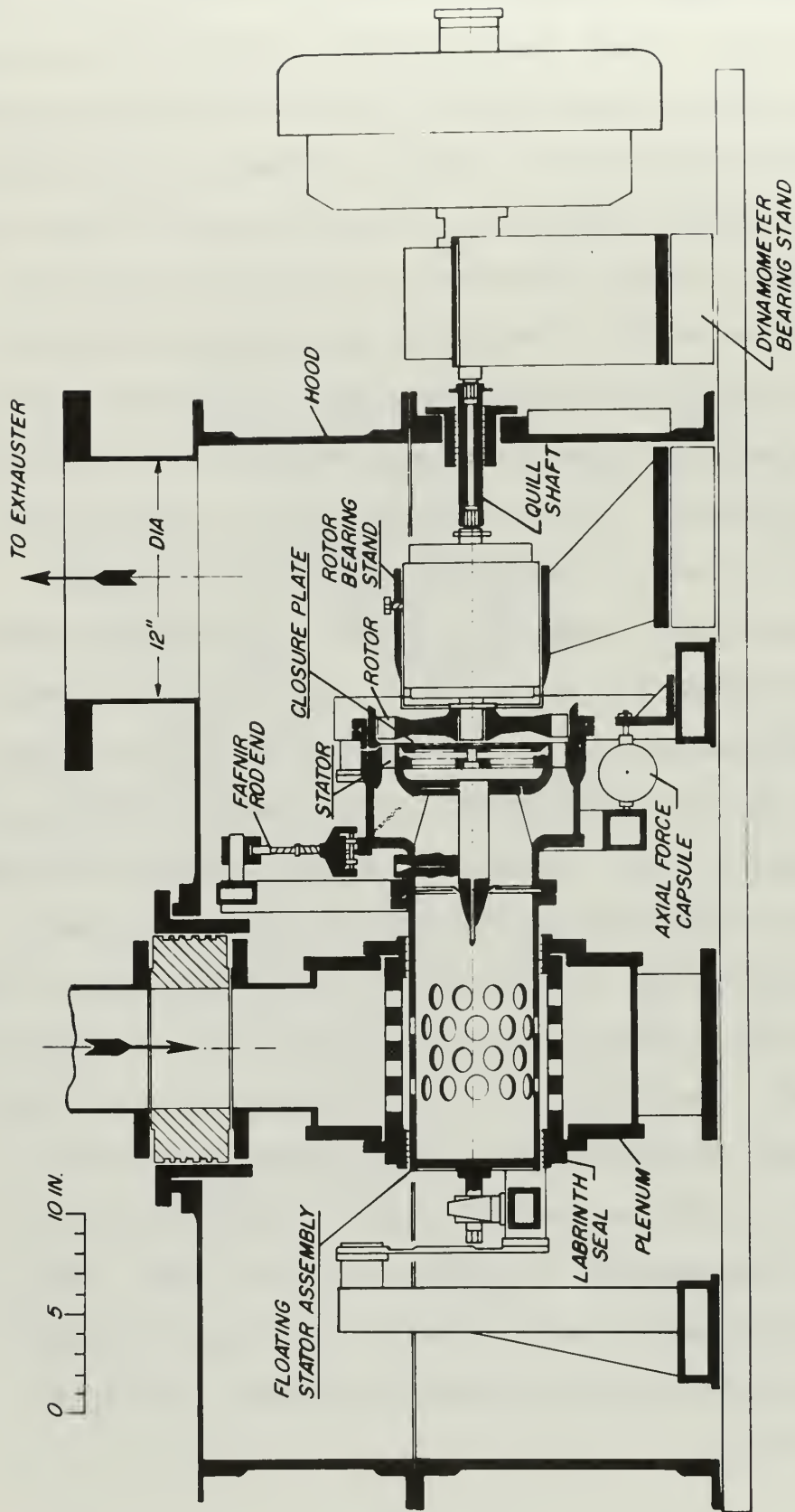
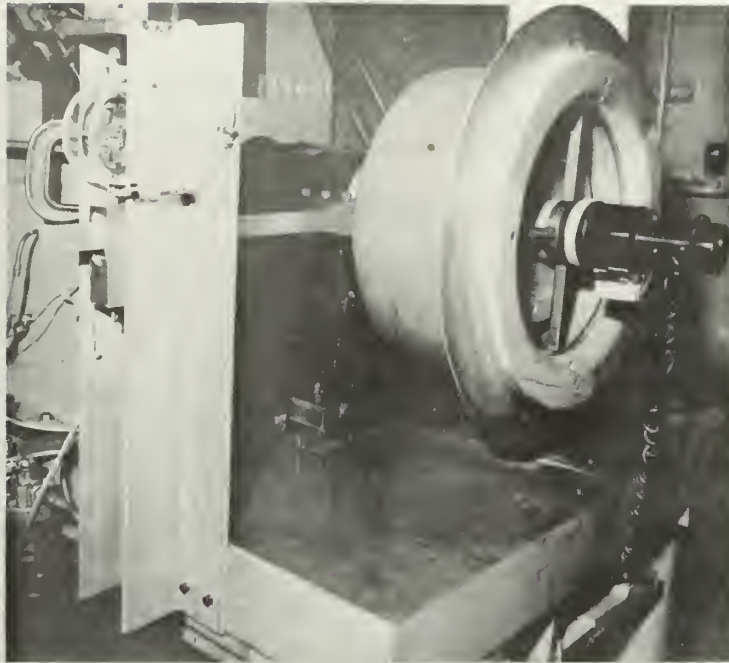


FIG. 3

TRANSONIC TURBINE TEST RIG

A six-lobe flux cutter is fitted to the end of the quill shaft outside of the hood. These lobes, in conjunction with a magnetic pickup provide a means for determining turbine speed. Turbine speed is read on a Hewlett-Packard electronic counter actuated by the pickup.

Turbine speed is controlled by varying the flow area of the dynamometer, thereby varying the load on the turbine. The dynamometer is designed so that its output torque may be measured in one of two ways, either by strain gage equipped electronic capsules or by a direct reading spring capsule. Neither of these methods proved entirely satisfactory. The electronic capsules were difficult to calibrate and repeatable results could not be obtained. This situation could be attributed to the wide temperature variations in the test cell during the tests. These temperature variations are due to high temperature air exhausting from the dynamometer. The spring capsule could only be read in the test cell. For these reasons a more reliable and temperature insensitive electronic torque measuring system was developed that could be calibrated with ease. Fig. 4 shows the installation of the measuring device that proved most satisfactory. The installation consists of a twenty in. long arm attached to the dynamometer housing which is used in conjunction with a Wiancko Model F1009 force capsule with a range of 0-50 lbs. The support structure for the force pickup consists of two aluminum extrusions attached to the test bed, with a bracket mounted between



(a)



(b)

FIG. 4

TORQUE MEASURING ASSEMBLY

(a) Dynamometer, Dynamometer Arm, Force Capsule Support Structure, Force Capsule. (b) Force Capsule with Dynamometer Arm in position.

them. The force pickup is attached to a Fafnir Self-Aligning Rod End mounted in the support bracket. To the lower end of the force pickup is fitted an aluminum block, milled out to allow the free end of the dynamometer arm to pass through it. The dynamometer arm has a ball fitting attached to the free end which rests in a hemispherical groove cut in the aluminum block. The hemispherical groove has a larger radius than the ball fitting to insure proper alignment between the dynamometer arm and the force capsule. Additionally, a fitting is attached to the bottom of the aluminum block for the calibrating of the force pickup with known weights. This assembly was successful for the measuring of the dynamometer torque independent of the ambient temperature in the test cell. The torque exerted on the dynamometer housing is transmitted to the arm and produces a tension load that is taken up by the force capsule. Only tensile loading will be indicated since the dynamometer arm is not rigidly attached to the force capsule, which in turn is not rigidly attached to the support structure.

The following paragraphs describe the flow of the air through the TTR and discuss the instrumentation necessary to measure the flow properties at different locations. Air enters the plenum chamber and passes through a large number of holes into the floating stator assembly. In the fixed plenum chamber the air temperature is measured by two thermocouples that extend into the airflow from opposite sides of the plenum chamber.

The mass flow of air entering the plenum is that measured by the flow nozzle. However, the airflow through the turbine is reduced by the leakage through the plenum labyrinth seals. Methods to determine the reduction in turbine mass flow due to this leakage are given by Eckert.⁵

A special probe housing extends from the central body of the floating stator assembly. This housing, referred to as the bullet probe, contains a total and static pressure probe and a thermocouple to measure total temperature. The total pressure measured by this probe was not used in these tests due to the losses incurred between this point and the stator.

Fig. 5 is an enlarged view of the stator section shown in Fig. 3. It illustrates the installation of the Ares Mod II Turbine in the test rig. Three conical screens are arranged ahead of the stator. They were installed to insure uniform flow distribution ahead of the stator. The total pressure at the stator inlet is therefore measured by six fixed Kiel probes, equally spaced around the periphery of the stator housing. Additionally, there are arranged two radially adjustable total pressure and total temperature Kiel probes just aft of the screens, as shown in Fig. 6.

A closure plate is attached to, and concentric with, the stator. The closure plate is installed to provide a means

⁵Eckert, R. H., Performance Analysis and Initial Tests of a Transonic Turbine Test Rig (Naval Postgraduate School, 1966), Sec. 5.

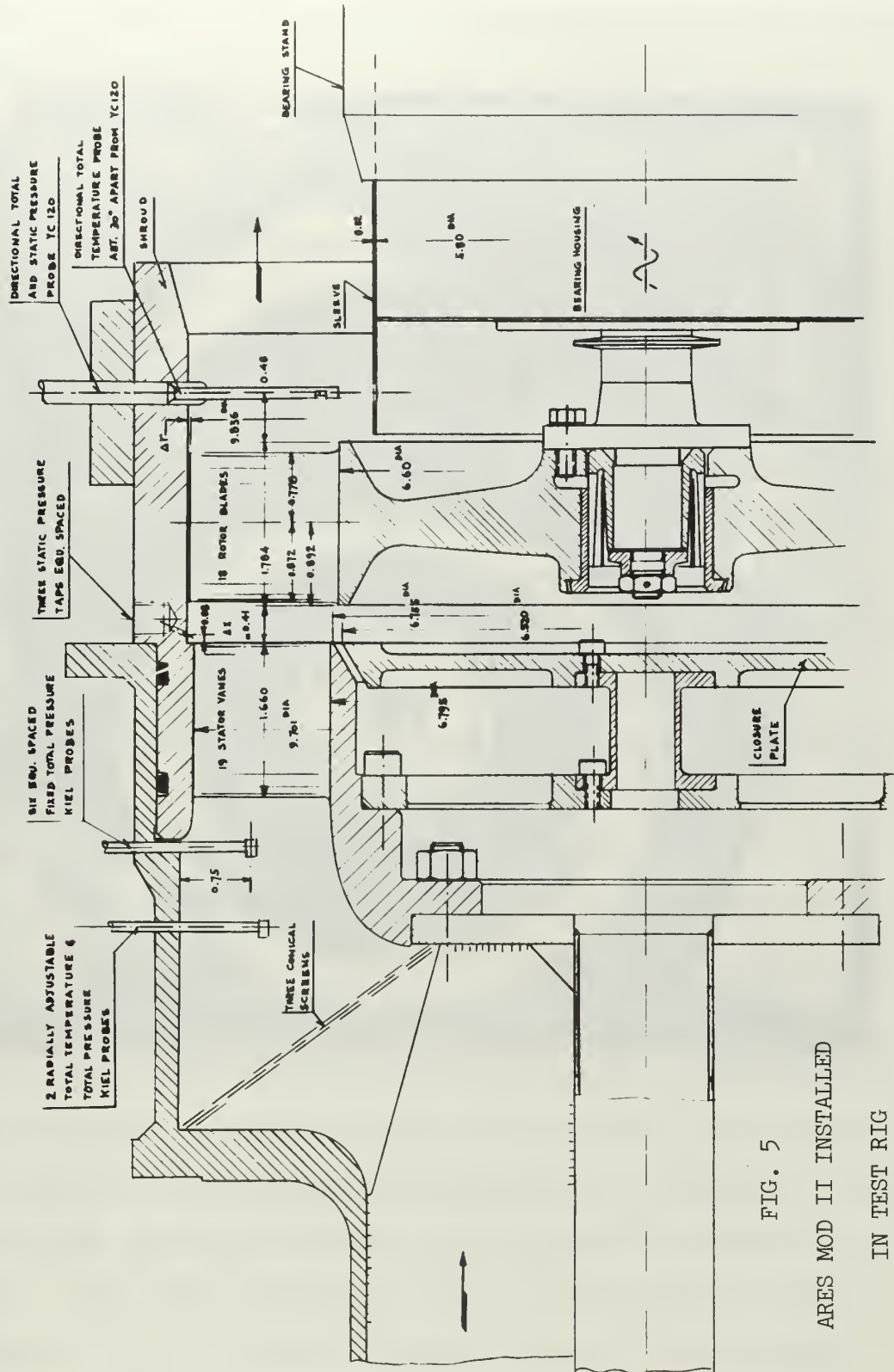


FIG. 5
 ARES MOD II INSTALLED
 IN TEST RIG

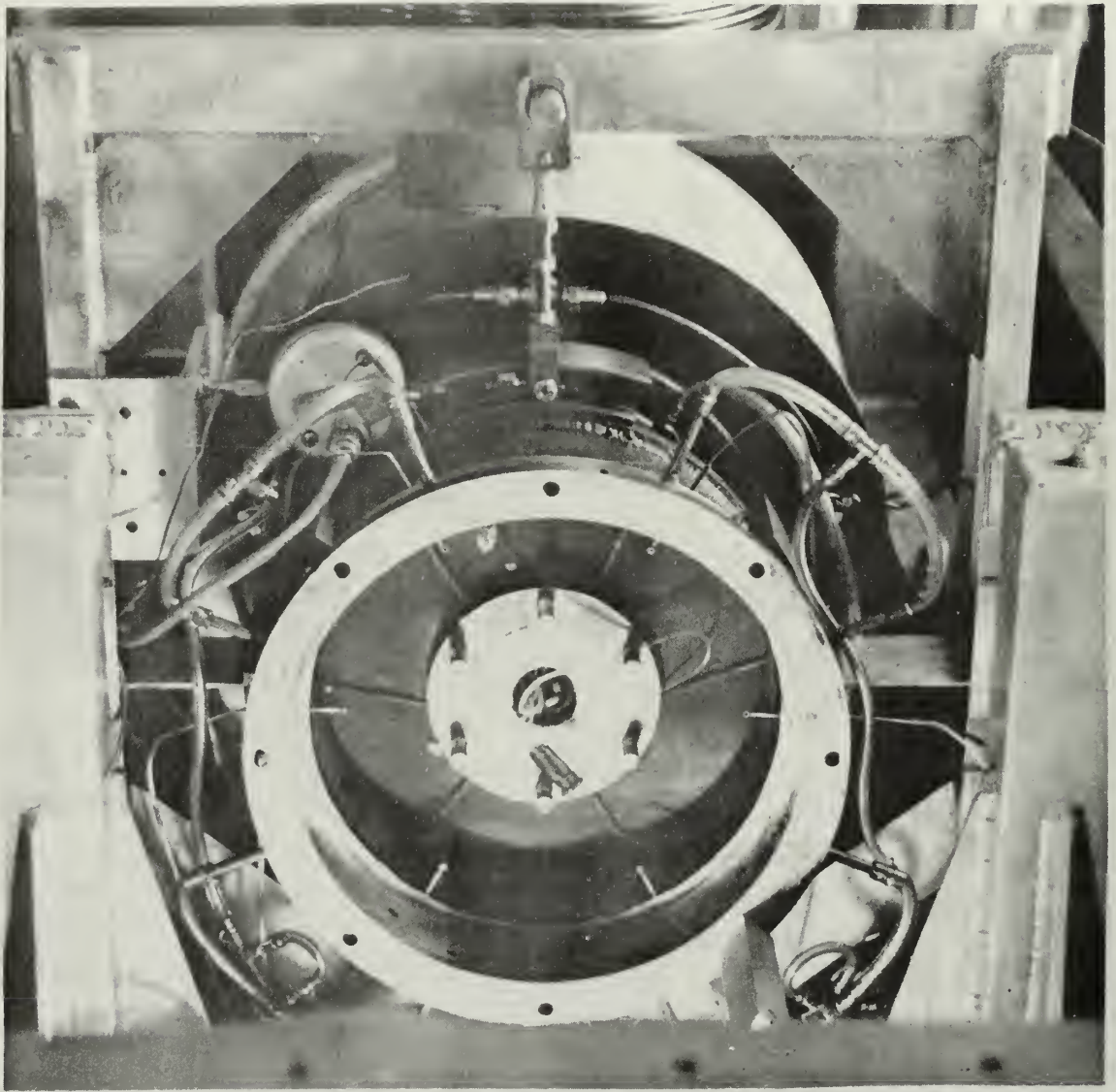


FIG. 6

STATOR INLET FLOW PASSAGE

of determining the static pressure force and moment on the stator assembly in the region between the stator hub and the turbine centerline.

Torque is exerted on the closure plate by the shear forces in the peripheral direction. These forces are the result of the rotational motion imparted to the fluid by the rotor.

The static pressure existing on the downstream side of the closure plate is different from that in the chamber behind the closure plate. Due to the rotational velocity imparted to the fluid by the rotor, the pressure on the downstream side of the closure plate varies with radius. Therefore, an accurate measurement of this pressure may not be readily obtained. However, by obtaining the force exerted on the closure plate, due to the pressure differential across it, an accurate measurement of the pressure force exerted on this area is obtained.

The closure plate is attached to a spool which in turn is attached to a spoked wheel type flexure. The spool and flexure are instrumented with strain gages to obtain the axial force and torque on the closure plate. The pressure in the chamber formed by the bullet probe support, stator and closure plate, P_{hub} , is measured on a manometer board with tubes that pass through the supports of the bullet probe. Then the net force on the closure plate is that caused by P_{hub} , less the measured force on the closure plate.

Initial tests were run with a three-spoke flexure. This flexure proved to be very sensitive to off center loadings and was difficult to calibrate. A four spoke flexure was manufactured and all four webs instrumented with strain gages to eliminate this deficiency. When the new flexure was installed, all the existing leads were used to obtain outputs from the four sets of strain gages installed on the webs. In order to obtain readouts from both the flexure and torque spool additional leads were required. However during the initial tests the torque on the closure plate was found to be less than one percent of the total torque on the stator assembly. Therefore, in the interest of saving time, new leads were not installed, and the strain gages on the torque spool were not connected during the final series of tests.

For data reduction purposes the torque on the closure plate was assumed to be constant, equal to 15 counts (0.276 ft.-lb.). The error in peripheral velocity caused by this assumption will be less than one-third of one percent.

Stator discharge pressure can be measured at the hub and tip radius only. The static pressure at the stator tip is measured by three equally spaced taps 0.25 in. downstream of the stator trailing edges. These taps are commoned to give an average tip static pressure.

The gap formed by the closure plate and the inner contour of the stator is 0.1275 in., and the outer edge of the closure plate is 0.020 in. inside of the inner flow passage of the stator discharge. The pressure existing in this gap

is assumed to be that at the hub of the stator. With the chamber formed by the bullet probe support, stator, and closure plate sealed from the atmosphere, the pressure in the gap is the pressure in the chamber.⁶

Tests were made at radial rotor tip clearances of 0.033 and 0.015 in. The larger clearance is obtained with the shroud of Fig. 5. This shroud has sixteen sets of three commoned static pressure taps, installed at 0.25 in. increments of the axial distance from the stator tip to the shroud exit. Additionally, one set of taps is arranged in the end face of the shroud.

For these tests the only pressures of interest were the stator tip static pressure and those that affect the axial force on the stator, namely, those measured by the two sets of taps on the bevel and the set on the end face of the shroud. With this shroud installed pressure and temperature surveys may be conducted at the rotor discharge.

A second shroud is available to permit surveys to be taken at the stator and rotor discharges, and in addition reduces the rotor radial tip clearance to 0.015 in. The pressure taps of this shroud are almost identical with those of the one previously described, with the difference that only two taps are arranged at each axial location. The second shroud has a double bevel at the exit instead of the single bevel shown in Fig. 5. Two sets of taps are installed in each bevel and at the end face of the shroud

⁶Vavra, M. H., Air Tests of the Mod I Ares Turbine, (Report Ares VA - T No. 8, 1966), p. 14.

to obtain a more accurate measurement of the pressures affecting the axial force on the stator.

All temperature probes are Iron-Constantan thermocouples. The cold junction is inserted in an ice bath to provide a 32° F. reference. Temperatures are read, in millivolts, on a 48 channel Brown potentiometer.

All pressures are measured on a 96 in. high mercury manometer board with 0.1 in. graduations.

An accelerometer mounted on the bearing stand provides a means of monitoring the vibration patterns of the test rig. The output signal of the accelerometer is displayed on a Singer Panoramic Analyzer and on an RMS millivoltmeter.

All instrument readouts are located in the control room. Fig. 7 is a schematic representation of the instrumentation of the TTR for the evaluation of the performance of the Ares Mod II Turbine.

3. Test Procedures.

Evaluation of stator performance using the momentum and moment of momentum equations requires a high degree of accuracy in determining the axial forces that are exerted on the stator assembly. Therefore all force measuring devices must be carefully calibrated to insure the required degree of accuracy.

The force capsules measuring the net force and moment on the stator assembly are calibrated prior to each test run. A system of lever arms has been devised to apply torque and axial loads on the stator assembly with known weights.

Axial force calibration is achieved by applying various axial loads on the stator assembly. The force capsule output is then linearized between ten and 12 KC. Linearization is accomplished by applying full load (160 lbs.) and adjusting the bandwidth to 12 KC, then removing the load to obtain a zero setting. Half load is then applied and the reading should be 11 KC. If the output is not 11 KC, the reading is changed in the direction of the error by an amount equal to three times the difference between 11 KC and the reading. Then the full and no load readings are readjusted to 12 and ten KC. This procedure is repeated until linearity is achieved. System hysteresis is checked by loading the assembly in 20 lb. increments from zero load to full load and then decreasing the load in like increments.

Stator torque is calibrated in the same manner as the axial force. In this instance, however, a torque load is obtained by applying the weights at the end of a 20 in. lever arm. Full load is attained with six 5 lb. weights resulting in a 50 ft.-lb. torque applied to the stator assembly.

Dynamometer calibration is performed with the dynamometer arm in position. This is taken as the zero load condition. The force capsule output is again linearized over a two KC range with the maximum reading corresponding to a dynamometer output of 800 in.-lb.

Initial tests were run with the dynamometer force capsule exposed, as shown in Fig. 4. During these tests a

great deal of difficulty was experienced in obtaining repeatable data. It was noted that, at the end of each test, the dynamometer output would not indicate zero load. Several attempts were made to determine the cause of this condition. It was finally ascertained that, even though the force capsule was said to be temperature insensitive in the test range, uneven heating or cooling caused large variations in the output signal. The source of the problem was exhaust from the dynamometer which discharges directly into the region where the capsule is mounted. Therefore, the force capsule was insulated from the dynamometer discharge air by installing an enclosure around it. To further insure that the capsule is maintained at a constant temperature, ambient air is blown through the enclosure by an electrically driven fan. This fan is situated outside the test cell to deliver air at almost constant temperature. The discharge of the fan is attached to a flexible tube of 3 in. diameter which is connected to the bottom of the enclosure.

After the dynamometer has been calibrated, its lever arm is lifted by hand to take a no-load tare reading of the force capsule. This tare provides a means of checking the capsule during the run for any long term temperature variation. The dynamometer arm is lifted periodically during the run and the reading obtained is compared with the tare reading. The dynamometer output is corrected by the difference of the two readings.

The axial force and torque on the closure plate can be calibrated only when the rotor is removed. However, these forces are measured by strain gages which do not require frequent calibration.

After the initial tests, it was found that the force capsules used to measure the axial force and torque on the stator assembly were affected in varying degrees by temperature changes also. However, these changes are not as radical as those experienced in the region of the dynamometer and may be easily compensated for. This is accomplished by noting the no load indication immediately after shutdown, and correcting the force and torque indication by an amount equal to the difference between the reading and the original calibration no-load reading. Since a relatively long temperature stabilization period is allowed prior to data acquisition, it is assumed that the force capsules reach temperature equilibrium also. During the initial test runs it was found that the torque on the closure plate was less than one percent of the net torque on the stator assembly. Additionally, the torque spool proved to be too rigid to achieve the desired sensitivity. Therefore, during the final tests, the closure plate torque was assumed to be constant.

Turbine Tests

During the test runs, periodic checks on the RPM measured by the electronic counter in the control room were made with a strobotac.

Thirty test runs were carried out at different pressure ratios, various axial clearances and radial clearances, over a range of speeds from 10,000 to 19,000 RPM. Mass flow rate variations at a given pressure ratio were achieved by operating with or without the exhauster assembly.

Runs 51 through 63 were made at a radial clearance of 0.033 in. and axial clearances of 0.2, 0.41, 0.64, 1.0 and 1.5 inches. Pressure ratios of 1.3, 1.4, 1.5, and 1.6 were chosen for the test operating conditions with the exhauster installed. These pressure ratios gave the widest range of data within the design regime of the Mod II turbine. When operating without the exhauster, pressure ratios of 1.3, 1.4, and 1.5 were chosen with only a limited operating range available at 1.5.

The pressure ratio across the turbine is controlled by two factors, namely, the vacuum produced in the hood by the exhauster, and the total pressure at the turbine inlet. For a high vacuum and a low inlet pressure, the mass flow rate through the turbine at a particular pressure ratio is smaller than that which occurs at a low vacuum and high inlet pressure, since the turbine flow rate is nearly proportional to the turbine inlet pressure.

If the inlet valve to the first settling tank is fully open, the total airflow through the turbine and the exhauster is regulated by the compressor discharge valve which controls the amount of compressor air delivered to the first settling tank of Fig. 1. The distribution of supply

air to the two components is controlled by an electrically operated exhauster inlet valve and a manually operated turbine inlet valve. Movement of either of these two valves alters the flow rate through both the exhauster and the TTR. An increase in airflow through the exhauster is accompanied by a decrease in flow through the turbine, and vice versa.

It was found that the most satisfactory results were obtained by leaving the exhauster inlet valve completely open at all times. With the exhauster inlet valve only partially open, large oscillations from the mean were noted in the pressure at the flow nozzle which resulted in flow instabilities throughout the TTR. If this valve is fully open, the pressure ratio across the turbine can be controlled by the turbine inlet valve and the compressor discharge valve.

To obtain the best possible data correlation, the turbine inlet conditions were set at chosen fixed values for each pressure ratio. These settings were used each time for a given pressure ratio. By operating in this manner any deviations from the norm could be easily recognized and corrected.

The rotor axial clearance was adjusted prior to each test run by moving the bearing housing in the rotor bearing stand. After adjusting the axial clearance, the dynamometer bearing housing is positioned to insure that the shaft splines of the rotor and dynamometer are engaged with the splines of the quill shaft over their whole axial length.

Just prior to commencing a test run, the two valves controlling the oil flow to the oil-mist producer for the rotor and dynamometer bearings and the air supply valve are turned on.

For a test run, the various control valves are set as follows:

1. Exhauster inlet - fully open
2. Turbine inlet - partially open
3. Compressor discharge - fully open
4. Inlet to the first settling tank - closed.

After the Allis-Chalmers compressor has been operating for a sufficient length of time to provide for temperature stabilization, the turbine is brought up to speed by opening the inlet valve to the first settling tank. When this valve is fully opened, the desired flow conditions are obtained by adjusting the compressor discharge and turbine inlet valves. At the beginning of each run the air dynamometer is usually set for maximum power absorption, to give minimum turbine speed. Once the desired flow conditions have been obtained, temperatures throughout the test installation must be allowed to stabilize prior to taking data.

The general procedures for taking data were as follows:

1. The pressure at the flow nozzle is recorded.
2. The following items may be recorded in any order, but must be taken when the nozzle pressure is at the point recorded in 1.

- a. Pressure drop across the flow nozzle.
 - b. RPM
 - c. Dynamometer torque
 - d. Stator torque
 - e. Stator axial force
 - f. Closure plate axial force
 - g. Closure plate torque
 - h. Shroud pressures
 - i. Total plenum pressures
 - j. Static plenum pressure
 - k. Hub pressure
 - l. Hood pressure.
3. The following temperatures are recorded:
- a. Temperature ahead of the flow nozzle
 - b. Total plenum temperatures (2)
 - c. Bullet probe temperature
 - d. Kiel temperatures (2)
4. The dynamometer arm is lifted and a no-load tare reading is recorded.
5. The dynamometer load absorption is reduced and steps (1) through (4) are repeated for the higher speed. Usually eight data points were recorded at each pressure ratio. If more than one pressure ratio was investigated during a run, the desired ratio was set and steps (1) through (5) were repeated.

This data taking procedure was not used during all tests. After the results of the first runs were studied, various

procedures were tried out to produce more repeatable results. In general it was recognized that, without the temperatures being stabilized, significant variations occurred while recording the data. However, if the temperatures were allowed to stabilize, they would remain constant while recording the remainder of the data. It should be noted here that whenever the mass flow rate through the turbine is altered significantly, by changing the turbine pressure ratio, the temperatures must again be allowed to reach a new steady state.

It was also noted that small changes in the pressure level ahead of the flow nozzle affected the RPM, the dynamometer reading, and the force capsule readouts almost instantaneously. However, total pressure at the turbine inlet, observed on the manometer board, lagged significantly behind the above mentioned readouts.

It is therefore necessary to set a definite value for the pressure at the flow nozzle inlet and to record the data only if this pressure can be maintained. Only with this procedure will it be possible to obtain test points for the proper flow rate.

The total pressure at the stator inlet varies somewhat if the RPM is changed. Therefore it is necessary to readjust this pressure to maintain a desired pressure ratio.

Pressure, temperature, and velocity surveys were conducted at the stator and rotor discharge during runs 64, 65, 66, 67, 74, 75, 76, and 79. For these surveys, the

exhauster assembly and the hood were removed. Prior to these tests the second rotor shroud was installed which has a radial rotor tip clearance of 0.015 in. Operation of the TTR without the exhauster is essentially the same as previously described. Then the exhauster inlet valve is closed, the turbine inlet valve is fully open, and the rotor discharges directly into the atmosphere. Thus the total pressure at the stator inlet is completely controlled by the compressor discharge valve.

The surveys were made with a United Sensor DA-125 five hole probe. With this probe it is possible to determine the pitch angle, yaw angle, total pressure, and dynamic head at the stator and rotor discharge. Fig. 8 is a picture of the probe and its supporting bracket as installed in the TTR.

An additional survey was made at the stator inlet with one of the two movable Kiel probes to obtain total pressure variations at various radial positions of the probe.

The surveys at the stator and rotor discharge give an indication of the flow discharge angles, pitch angles, and absolute velocities at these locations.

The total pressure tap (#1), and one of the two static pressure taps (#2) or (#3) of the probe of Fig. 8, were connected to either side of a mercury manometer. Additionally, the total pressure tap was connected to one side of a mercury manometer with the other side open to the atmosphere. The two static pressure taps, (#2) and (#3), and

TABLE I

MOD II TURBINE TEST RUNS 1967

Jan.-Aug.

| Run No. | r (in.) | X (in.) | $\frac{P_{t0}}{P_2}$ | N (RPM) | Test Purpose |
|---------|---------|---------|-------------------------------|-------------|---------------------------------|
| | | | (approx.) | **Varied | + = Performance Characteristics |
| ○ 51 | 0.033 | 0.410 | 1.4, 1.5 | **13-20,000 | + |
| * 52 | | 0.620 | 1.4, 1.5 | 14,000 | Test Hood Checkout |
| * 53 | | 0.620 | 1.3, 1.4 1.5 | **11-18,000 | + |
| * 54 | | 0.410 | 1.6, 1.8 | **15-18,000 | + |
| ○ 55 | | 0.410 | - | 0 | Dynamometer Calibration |
| * 56 | | 0.410 | 1.4 | **13-19,000 | " " |
| * 57 | | 0.410 | 1.4 | **13-19,000 | Spring Capsule Check |
| * 58 | | 0.620 | 1.3, 1.5 1.6 | **10-17,000 | + |
| * 59 | | 1.000 | 1.3, 1.4, 1.5, 1.6 | **10-17,000 | + |
| * 60 | | 0.200 | 1.3, 1.4 1.5, 1.6 | **10-17,000 | + |
| * 61 | | 0.410 | 1.3, 1.4 | **11-17,000 | + |
| * 62 | | 0.410 | 1.5, 1.6 | **13-17,000 | + |
| * 63 | | 1.500 | 1.3, 1.4 1.45, 1.5, 1.6 | **11-19,000 | + |
| ○ 64 | 0.015 | 1.000 | 1.4 | 14,600 | Stator Exit Survey |
| ○ 65 | | | 1.4 | 14,600 | " " " |
| ○ 66 | | | 1.4 | 14,600 | " " " |
| ○ 67 | | | 1.4 | 14,600 | Rotor Exit Survey |
| * 68 | | | 1.3, 1.4 1.5, 1.6 | **11-18,000 | + |
| * 69 | | 0.410 | 1.3, 1.4 1.5 | **11-19,000 | + |

TABLE I (CONT'D)

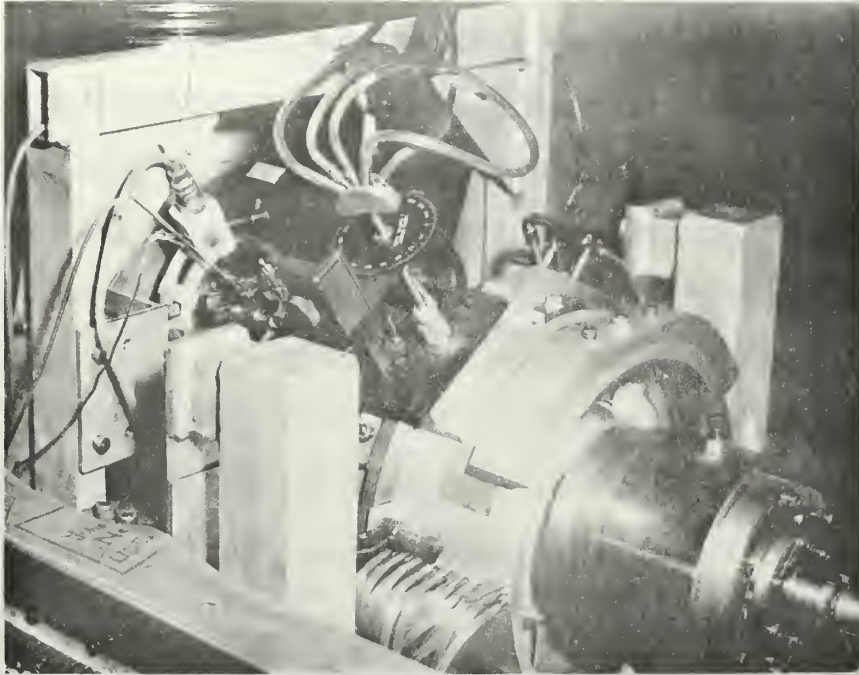
MOD II TURBINE TEST RUNS 1967

Jan.-Aug.

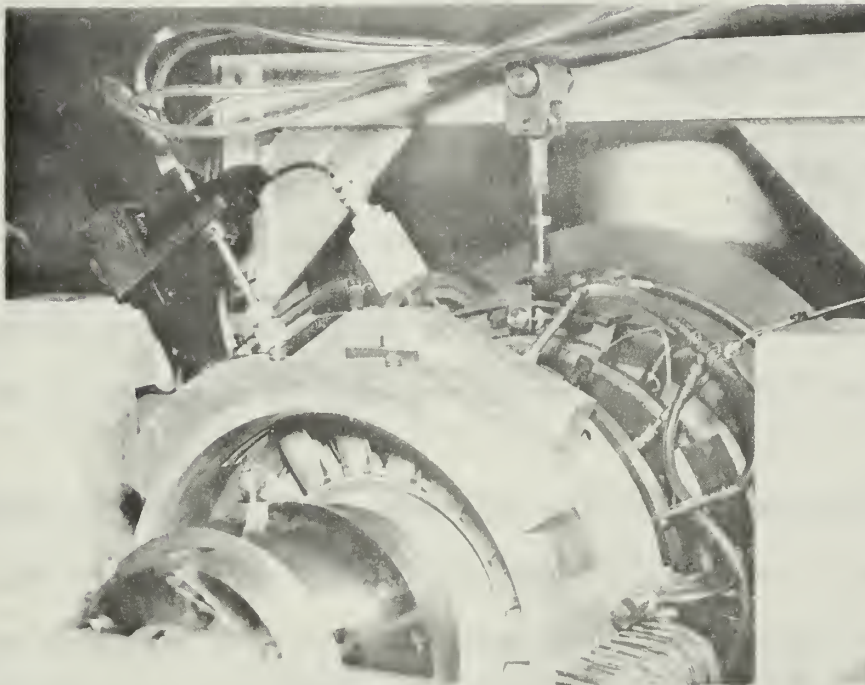
| Run No. | r (in.) | x (in.) | $\frac{P_{t0}}{P_2}$ (approx.) | N (RPM) | Test Purpose |
|---------|---------|-------------------------|-----------------------------------|-----------------|---------------------------------|
| | | | | **Varied | + = Performance Characteristics |
| *70 | 0.015 | 0.410 | 1.6 | **14-19,000 | + |
| *71 | | 0.200 | 1.3, 1.4 | **11-17,000 | + |
| *72 | | 0.410 | 1.3, 1.4 1.5 | **11-18,000 | Repeat Run 69 |
| *73 | | 0.200 | 1.3, 1.5 | | Repeat Run 71 |
| °74 | | 0.200 | 1.3, 1.4 | | Repeat Run 71 |
| °75 | | 0.410 | 1.3 | 11,500 | Stator Entrance & Exit Survey |
| °76 | | 0.410 | 1.3, 1.4 1.5 | **11,400-19,000 | Rotor Exit Survey |
| °77 | | 0.410 & 1.000 | 1.3, 1.4 | **11,400-19,000 | + |
| °78 | | .200, .410, 1.000 | 1.3 | **11,400-18,000 | Spring Capsule Check |
| °79 | | .200 | 1.3, 1.4 | **11-17,000 | Repeat Run 71 |
| °80 | | .410 | 1.3 | Varied | |

* With Exhauster $P_2 < P_{\text{barometer}}$

° Without Exhauster $P_2 = P_{\text{barometer}}$



(a)



(b)

FIG. 8

TRAVERSE PROBE INSTALLATION

(a) View showing Protractor. (b) View showing probe in blade passage.

the taps used to measure pitch angle, (#4) and (#5), were each connected to either side of a water manometer.

During the tests the probe was first turned about its axis until the pressures indicated by taps (#2) and (#3) were balanced to establish the flow direction. The flow angle is then read from a protractor attached to the probe, which is adjusted to read zero when the probe is aligned with the axis of the turbine.

Temperature surveys were also made at the stator and rotor discharge with a Kiel probe mounted in the same manner as the pressure probe.

Table I is a list of the runs performed, showing the radial and axial clearances, pressure ratios, speed ranges and test purpose for each run.

4. Data Reduction.

General

The method of analysis used to evaluate turbine performance is a one dimensional, mean streamline approach given by Vavra.⁷

The equations of continuity, momentum, moment of momentum and energy are used in the analysis. Of primary concern here is the use of the momentum and moment of momentum equations to accurately predict the flow conditions at the stator discharge. However, given in the following is also a

⁷Vavra, M. H., Aero-Thermodynamics and Flow in Turbomachines (New York, London: John Wiley and Sons, Inc., 1960) Chap. 15

complete description of the data reduction methods employed to obtain the overall turbine performance.

A program (TTR2) was written to reduce the data on the IBM 360 Computer of the Naval Postgraduate School. A program description and a sample program are given in Appendix A.

Flow Rate Determination

The turbine flow rate is equal to the flow through the flow nozzle less the plenum labyrinth seal leakage flow

$$\dot{w}_t = \dot{w}_n - \dot{w}_l \quad (1)$$

The nozzle flow rate is obtained from the basic flow equation for a flow nozzle:⁸

$$\dot{w}_n = 359.1 D_n^2 \alpha_n C_n Y_1 \sqrt{\rho_1 h_w} \quad (\text{lbm/hr})$$

where

D_n = nozzle throat diameter (in.)

α_n = coefficient of thermal expansion (dimensionless) based on D_n^2

C_n = discharge coefficient (dimensionless)

Y_1 = expansion coefficient (dimensionless)

ρ_1 = fluid density at the upstream tap (lbm/ft³)

h_w = differential pressure across taps (in. H₂O)

For the flow nozzle, made of 2024-T4 aluminum:⁹

$$\alpha_n = 1 + 0.00252 \frac{(T_{noz} - 527.7)}{100}$$

⁸Stearnes, R. F., et. al., Flow Measurement with Orifice Meters (New York: Van Nostrand Co., Inc., 1951), p. 6.

⁹Eckert, R. H., Performance Analysis and Initial Tests of a Transonic Turbine Test Rig (Naval Postgraduate School, 1966), p. 99.

The nozzle discharge coefficient, C_n , was first determined by Eckert and then by Naviaux. Naviaux shows that the nozzle discharge coefficient is constant, equal to 1.05, above a Reynolds Number of 9.75×10^5 .¹⁰

All tests were performed at mass flow rates corresponding to Reynolds Numbers in the region where C_n is a constant.

The expansion coefficient for a flow nozzle is

$$Y_1 = \left[\frac{\gamma}{\gamma-1} \frac{(P_2/P_{noz})^{2/\gamma} - (P_2/P_{noz})^{\gamma+1/\gamma}}{1 - (P_2/P_{noz})} \frac{1 - \beta^4}{1 - \beta^4 (P_2/P_{noz})^{2/\gamma}} \right]$$

where β is the diameter ratio D_n/D_1 .¹¹ The final form of the basic flow equation with all pressures in pounds per square inch, and temperatures in degrees Rankine, becomes:

$$\dot{w}_n = 0.8633 D_n^2 \alpha_n C_n Y_1 \sqrt{\frac{P_{noz} h_w}{T_{noz}}}$$

The leakage flow rate through the plenum labyrinths has been obtained from tests.¹²

$$\dot{w}_1 = 0.116 \frac{\phi_1 P_{sp1}}{\sqrt{T_{t1}}}$$

¹⁰Naviaux, J. C., Transonic Turbine Test Rig Exhauster System Tests and Tests of a Reaction Turbine. (Naval Postgraduate School, 1966), p. 68.

¹¹Asme Power Test Codes, Information on Instruments and Apparatus, Part 5, Chapter 4, Flow Measurement by Means of Standard Nozzles and Orifice Plates.

¹²Eckert, R. H., Performance Analysis and Initial Tests of a Transonic Turbine Test Rig (Naval Postgraduate School), 1966), p. 113.

where

$$\phi_1 = \sqrt{\frac{1 - p_{\text{hood}}/p_{\text{SPI}}}{4.0791}}$$

Then the turbine mass flow rate is found from Eq. (1).

Stator Analysis

To obtain the stator performance characteristics by the application of the momentum, and the moment of momentum equations, requires a high degree of accuracy in the measurement of the pressures and forces acting on the stator assembly. Since the force on the stator assembly is small, any error in the pressure readings manifests itself in a relatively large error in the net force which is used to determine the axial velocity. This is especially true for pressures at the hub and tip of the stator since they are multiplied by relatively large areas.

To evaluate the general momentum theorem as given by Vavra,¹³

$$\begin{aligned} \frac{d}{dt} \left[\int_R dm_s \bar{V} \right] + \int_{S_1} dm_s \bar{V}_1 - \int_{S_0} dm_s \bar{V}_0 \\ = \int_S -\bar{n} p dS + \int_S -\bar{t} \tau dS + \bar{G} \end{aligned} \quad (2)$$

and to apply the results to determine stator performance, certain simplifying assumptions will be made; namely:

¹³Vavra, M. H., Aero-Thermodynamics and Flow in Turbo-machines (New York, London: John Wiley and Sons, Inc., 1960), Chap. 5.

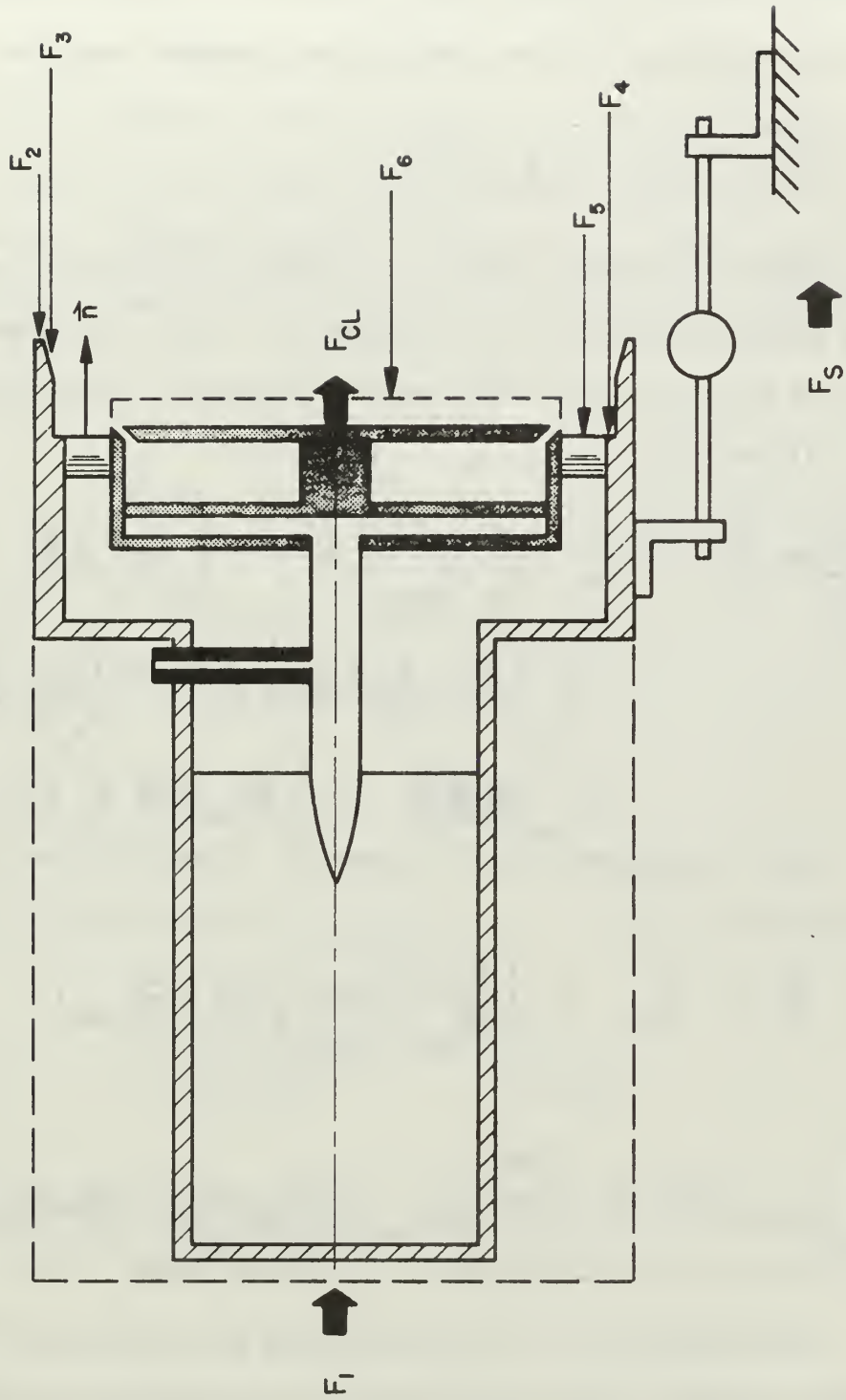
1. The flow is steady and adiabatic.
2. The flow is axisymmetric with no initial peripheral component and no radial component throughout.
3. The operating fluid is a perfect gas with a constant specific heat.
4. Shear stresses on the inner perimeter of the shroud and at the discharge annulus are ignored. The integral of the latter is small.

Since the test rig is designed so that the airflow enters the floating stator assembly without an axial component, Eq. (2) may now be reduced to

$$\int_{S_1} dm_s \nabla_1 = \int_S -\bar{n} p dS \quad (2a)$$

where the term representing the weight of the fluid particles in the control region has been ignored since it is small in relation to the inertia forces. For the control region shown in Fig. 9, the terms on the right hand side of Eq. (2a) become

$$\begin{aligned} \int_S -\bar{n} p dS &= \int_{A_2} -\bar{n} p_{17} dA + \int_{A_3} -\bar{n} \left(\frac{p_{14} + p_{15} + p_{16}}{3} \right) dA \\ &+ \int_{A_4} -\bar{n} p_{TIP} dA + \int_{A_5} -\bar{n} p_i dA + \int_{A_6} -\bar{n} p_{hub} dA \\ &+ \int_{A_1} +\bar{n} p_{hood} dA + \int_{S_w} -\bar{n}_w p_w dS_w + \int_{A_7} -\bar{n} (p_7 - p_{hub}) dA \end{aligned}$$



SCHMATIC OF STATOR FORCES
 FIG. 9

The last integral on the right-hand side of Eq. (3) is the measured force exerted on the closure plate, F_{Cl} . The seventh integral is the force exerted by all the surfaces in contact with the flow inside the control region.

$$\bar{F}_s = \int_{S_w} -\bar{n}_w p_w dS_w$$

The component of this term in the axial direction is the force measured by the force capsule. Since the remaining terms of Eq. (3) have only axial components, combining Eqs. (2) and (3)

$$\begin{aligned} \int_{A_5} dm_s V_{a_1} &= F_s + \int_{A_1} p_{hood} dA - \int_{A_2} p_{17} dA \\ &\quad - \int_{A_3} \left(\frac{p_{14} + p_{15} + p_{16}}{3} \right) dA - \int_{A_4} p_{tip} dA \\ &\quad - \int_{A_5} p_i dA - \int_{A_6} p_{hub} dA + F_{Cl} \end{aligned} \quad (4)$$

for an assumed linear pressure distribution from stator hub to tip

$$p_i = p_{hub} + \left(\frac{p_{tip} - p_{hub}}{R_{tip} - R_{hub}} \right) (R - R_{hub})$$

then

$$\int_{A_5} p_i dA = \int_{R_{hub}}^{R_{tip}} 2\pi \left[p_{hub} + \left(\frac{p_{tip} - p_{hub}}{R_{tip} - R_{hub}} \right) (R - R_{hub}) \right] R dR \quad (4a)$$

The remainder of the pressures are measured, and forces caused by them are simply the pressures times their respective areas. Solving Eq. (4) for V_{a_1}

$$V_{a_1} = \frac{\int dm_s V_{a_1}}{m_s} = \frac{g}{\dot{w}_t} \left[F_s + F_{c_1} + F_i - \sum_{i=2}^6 F_i \right] \quad (5)$$

where F_i are the pressure forces, and $m_s = \dot{w}_t/g$.

The peripheral component of the stator discharge velocity is obtained by applying the moment of momentum equation to the control region. Applying the same simplifying assumptions used in the previous development for determining V_{a_1}

$$M = \int_{s_0} dm_s R_0 V_{u_0} - \int_{s_1} dm_s R_1 V_{u_1}$$

Since the flow enters the stator axially, $V_{u_0} = 0$, and ignoring algebraic signs,

$$M = \int_{s_1} dm_s R_1 V_{u_1} \quad (6)$$

With the mean streamline approach

$$V_{u_1} = \frac{\int_{s_1} dm_s R_1 V_{u_1}}{m_s R_{m_1}} = \frac{M}{m_s R_{m_1}} \quad (7)$$

The moment measured by the force capsule is not equal to M . There is an additional moment imparted to the closure plate due to the shear forces in the peripheral direction. These forces are the result of the rotating motion imparted to the fluid by the turbine wheel. There also exists a moment on the inner face of the shroud caused by

shear forces in the peripheral direction. However, for this analysis this latter moment was ignored and the net moment created by the change in peripheral velocity is taken as

$$M = (M_S + M_{Cl}) \quad (8)$$

where M_{Cl} is the moment caused by shear forces on the closure plate, and M_S is the measured moment on the stator assembly. Then, combining Eqs. (7) and (8)

$$V_{u_1} = \frac{g(M_S + M_{Cl})}{R_{m_1} \dot{w}_t} \quad (9)$$

where

$$R_{m_1} = \sqrt{\frac{R_{TipL}^2 + R_{hubL}^2}{2}}$$

Using the results obtained from Eqs. (5) and (9)

$$V_1 = \sqrt{V_{u_1}^2 + V_{a_1}^2} \quad (10)$$

and the absolute discharge angle

$$\alpha_1 = \tan^{-1} (V_{u_1} / V_{a_1})$$

Since the flow is assumed to be steady and adiabatic, and because there is no work done in the stator, the total enthalpy along the mean streamline must remain constant. Therefore,

$$H = h_1 + \frac{V_1^2}{2gJ} \quad (11)$$

With the perfect gas assumption, Eq. (11) reduces to

$$T_{tpl} = T_1 + \frac{v_1^2}{2gJc_p}$$

Solving for T_1 ,

$$T_1 = T_{tpl} - \frac{v_1^2}{2gJc_p} \quad (12)$$

For an isentropic process

$$T_{1is} = T_{tpl} \left(\frac{p_{tpl}}{p_1} \right)^{\frac{\gamma-1}{\gamma}} \quad (13)$$

Thus, the stator loss coefficient is defined by

$$\zeta_s = \frac{T_1 - T_{1is}}{\Delta T_{1s}} = \frac{T_1 - T_{1is}}{T_{tpl} - T_{1is}} \quad (14)$$

and the stator efficiency is

$$\eta_s = 1 - \zeta_s$$

To insure that the solution obtained from momentum satisfies continuity as well, a check is performed by utilizing flow functions defined by Vavra.¹⁴

$$\Phi = \frac{\dot{w}_t}{p_{tpl} A_{th}} \sqrt{T_{tpl} \frac{R}{g}} = \left[\frac{2\gamma}{\gamma-1} \left(\left(\frac{p_{th}}{p_{tpl}} \right)^{2/n} - \left(\frac{p_{th}}{p_{tpl}} \right)^{\frac{n+1}{n}} \right) \right]^{1/2} \quad (15)$$

where

$$n = \frac{\gamma}{1 + \zeta_p(\gamma-1)} \quad (16)$$

and

$$A_{th} = aZ h$$

¹⁴Vavra, M. H., Problems of Fluid Mechanics in Radial Turbomachines (Rhode-Saint Genese, Belgium, Von Karman Institute of Fluid Dynamics, 1965), VKI Course Note 55a Equation C (7).

Φ represents a non-dimensional flow rate. The right hand side of Eq. (15) is a function of pressure ratio, the value of γ , and the polytropic loss coefficient ζ_p . The pressure at the throat is not a measured quantity. For want of a better understanding of the phenomena involved, it will be assumed that throat pressure and discharge pressure are equal. Fig. 10 shows the stator flow passage.

With the assumption that the static pressure at the throat equals that after the stator

$$\Phi = \left[\frac{2\gamma}{\gamma-1} \left(\left(\frac{p_1}{p_{tpl}} \right)^{\frac{2}{n}} - \left(\frac{p_1}{p_{tpl}} \right)^{\frac{n+1}{n}} \right) \right]^{\frac{1}{2}}$$

Fig. 11 represents an adiabatic expansion process with friction in a channel with varying cross sections. The derivation of the polytropic loss coefficient is given by Vavra¹⁵

$$dT_{is} = T \left[1 - \left(\frac{p-dp}{p} \right)^{\frac{\gamma-1}{\gamma}} \right] = T \left[\frac{\gamma-1}{\gamma} \frac{dp}{p} \right] \quad (17)$$

letting

$$1 - \zeta_p = \frac{dT}{dT_{is}} \quad (18)$$

ζ_p is assumed to be constant throughout the expansion process. Combining Eqs. (17) and (18)

$$\frac{dT}{T} = (1 - \zeta_p) \left(\frac{\gamma-1}{\gamma} \frac{dp}{p} \right) \quad (19)$$

¹⁵ Ibid., p. C23.

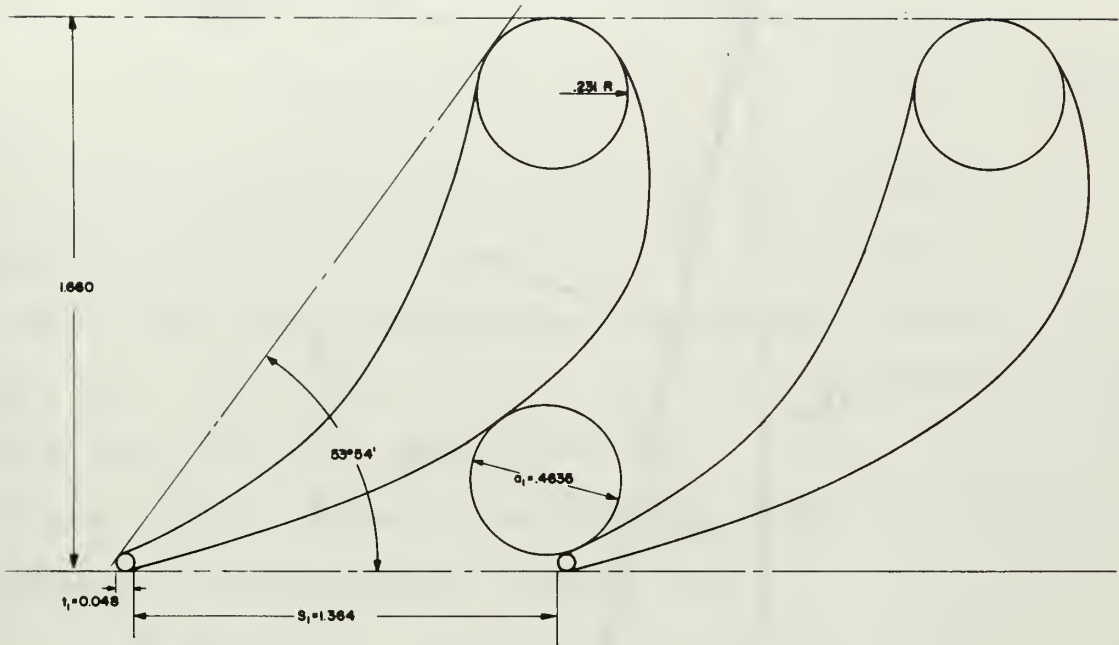


FIG. 10
 STATOR BLADE PROFILE

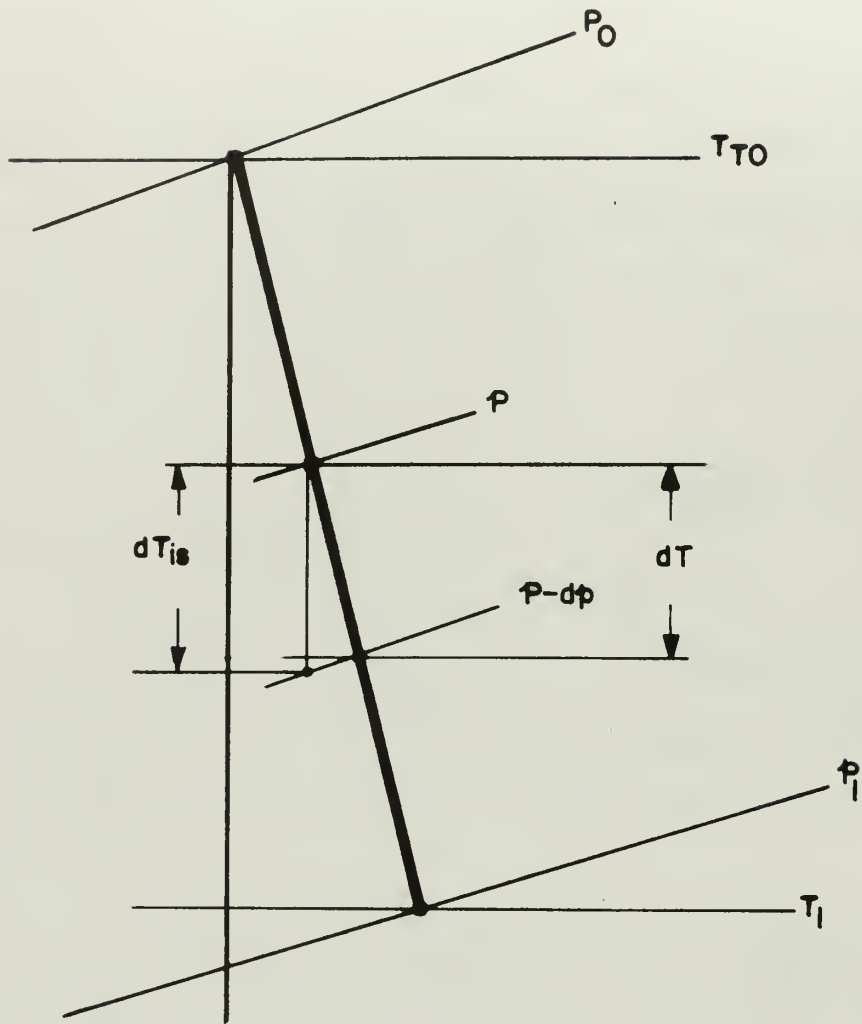


FIG. II
EXPANSION PROCESS WITH FRICTION

Integrating Eq. (19)

$$T = (\text{constant}) p^{(1-\zeta_p) \frac{\gamma-1}{\gamma}} \quad (20)$$

Eq. (20) may be expressed as

$$T = (\text{constant}) p^{\frac{n-1}{n}} \quad (21)$$

The expression shown in Eq. (16) is obtained by setting the exponents of Eq. (20) and Eq. (21) equal to each other. The actual stator loss coefficient is defined by Eq. (14). The polytropic loss coefficient is somewhat higher than the actual loss coefficient due to reheat. However, at low pressure ratios the difference is less than three percent. It is therefore assumed that

$$\zeta_p = \zeta_s \quad (22)$$

With this assumption Eq. (16) becomes

$$n' = \frac{\gamma}{1 + \zeta_s(\gamma-1)} \quad (23)$$

The three percent error introduced by Eq. (22) manifests itself in a change of less than one tenth of a percent in the polytropic exponent shown in Eq. (23).

Using the value of the polytropic exponent from Eq. (23) in Eq. (15)

$$\frac{\dot{W}_t}{T_{tpl} A_{th}} \sqrt{T_{tpl} \frac{R}{g}} = \left[\frac{2\gamma}{\gamma-1} \left(\left(\frac{p_i}{p_{tpl}} \right)^{\frac{2}{n'}} - \left(\frac{p_i}{p_{tpl}} \right)^{\frac{n'+1}{n'}} \right) \right]^{\frac{1}{2}} \quad (24)$$

The quantities on the left-hand side of Eq. (24) are either known or measured quantities; therefore, it is assumed

to be correct. P_1 , however, is not measured but is based on the assumption that the pressure distribution at the stator discharge is linear. If this assumption is incorrect, Eq. (24) will not be satisfied. Therefore an iterative process is adopted in which p_1 is changed until Eq. (24) is satisfied. Each time p_1 is incremented, V_{a1} is recomputed using Eq. (5). Since V_{u1} is unaffected by changes in p_1 , its value will remain constant. The quantities V_1 , T_1 , and T_{1is} are recomputed to obtain a new stator loss coefficient, which is then used to obtain a new value for the polytropic exponent which is introduced into Eq. (24) for a subsequent check. This process is repeated until Eq. (24) is satisfied. The solution finally obtained will satisfy both momentum and continuity within the limits of the adopted assumptions.

The isentropic flow function is defined by

$$\Phi_{is} = \left[\frac{2\gamma}{\gamma-1} \left(\left(\frac{p_1}{p_{tP1}} \right)^{\frac{2}{\gamma}} - \left(\frac{p_1}{p_{tP1}} \right)^{\frac{\gamma+1}{\gamma}} \right) \right]^{1/2} \quad (25)$$

This flow function holds for an isentropic expansion process from P_{tP1} to p_1 .

If the expansion process were isentropic, the area required to pass the flow would be less than that for the same process with friction. The area blockage factor is defined as

$$\xi = \frac{\Phi}{\Phi_{is}} \quad (26)$$

ξ represents the percentage of the throat area that would be required to pass the flow if the expansion were isentropic.

Rotor Analysis

The analysis of the rotor performance is based on the moment of momentum equation. For this application the moment of momentum equation is

$$M = \int_{S_1} dm_S R_1 V_{u1} - \int_{S_2} dm_S R_2 V_{u2} \quad (27)$$

The moment shown in Eq. (27) represents the moment exerted on the rotor by the fluid due to the change in peripheral velocity across the rotor, which is measured by the dynamometer.

The first integral on the right-hand side of Eq. (27) was evaluated in the previous section. Combining Eq. (7) with Eq. (27)

$$\int_{S_2} dm_S R_2 V_{u2} = m_S R_{m1} V_{u1} - M_D \quad (28)$$

Again assuming that the average peripheral component acts at the mean streamline

$$V_{u2} R_{m2} m_S = \int_{S_2} dm_S R_2 V_{u2} \quad (29)$$

Substituting Eq. (29) for the left hand side of Eq. (28) and solving for V_{u2}

$$V_{u2} = \frac{R_{m1}}{R_{m2}} V_{u1} - \frac{M_D g}{\dot{w}_T R_{m2}} \quad (30)$$

where $\dot{w}_t/g = m_s$.

The rotor analysis is greatly simplified by introducing the so-called equivalent total enthalpy and total pressure as defined by Vavra.¹⁶

The state point shown at E in Fig. 12 is defined by H_E and P_{t_E} . By defining a state point in this manner the expansion process through the rotor may be evaluated as if the flow passage were a stationary blade row. The equivalent total enthalpy is obtained from

$$H_E = h_1 + \frac{W_1^2}{2gJ} - \left(\frac{U_1^2 - U_2^2}{2gJ} \right) \quad (31)$$

where W_1 is the relative velocity of the flow at the rotor entrance. Its relationship to the absolute velocities at the stator discharge is shown in Fig. 13. With the known quantities V_1 , U_1 , and α_1 , W_1 may be found from

$$W_{a_1} = V_{a_1}$$

$$W_{u_1} = V_{u_1} - V_1$$

and

$$\beta_1 = \tan^{-1} \frac{W_{u_1}}{W_{a_1}}$$

Then

$$W_1 = \sqrt{W_{u_1}^2 + W_{a_1}^2}$$

The equivalent total enthalpy and temperature are established once the conditions at the stator discharge are known. However, the state point E cannot be immediately

¹⁶Ibid., p. G4.

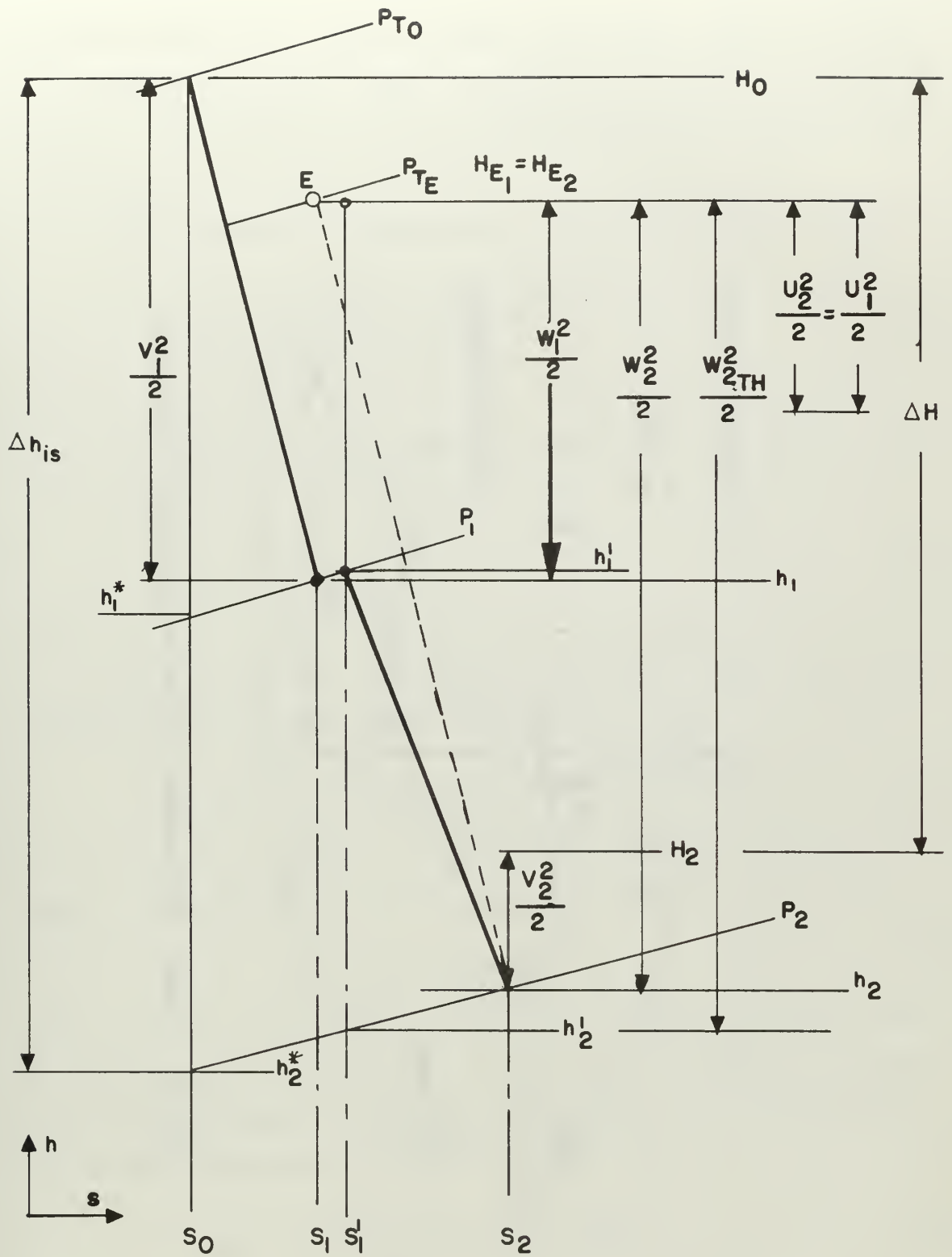


FIG.12 THERMODYNAMIC PROCESS OF FLUID IN TURBINE STAGE

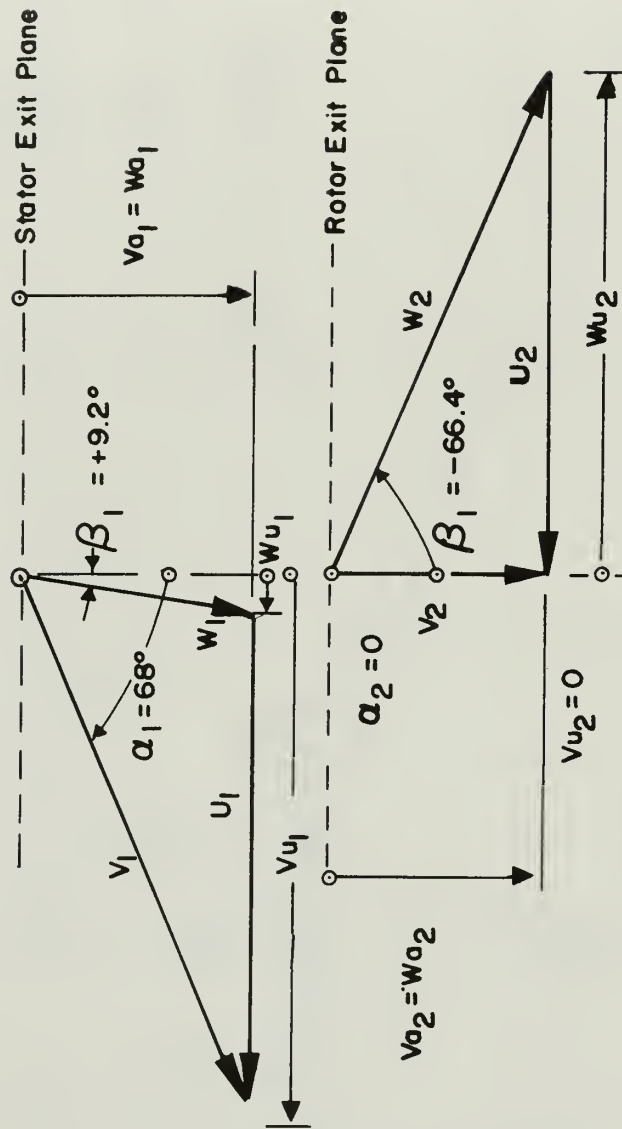


FIG. 13 MEAN STREAMLINE, DESIGN POINT, VELOCITY DIAGRAM

defined unless the losses between the blade rows are known. The conditions at the rotor discharge may still be obtained. However, the complexity of the calculations is somewhat increased.

The isentropic temperature change from T_{tE} to T_{2is} is

$$\Delta T_{is} = T_{tE} \left[1 - \left(\frac{p_2}{P_{tE}} \right)^{\frac{\gamma-1}{\gamma}} \right] \quad (32)$$

It is assumed that p_2 is equal to the hood pressure or ambient pressure when operating without the exhauster. Then the rotor efficiency is obtained from

$$\eta_r = \frac{T_{tE} - T_2}{\Delta T_{is}} \quad (33)$$

Solving Eq. (33) for T_2 and substituting for ΔT_{is} from Eq. (32) the resulting expression is

$$T_2 = T_{tE} \left\{ 1 - \eta_r \left[1 - \left(\frac{p_2}{P_{tE}} \right)^{\frac{\gamma-1}{\gamma}} \right] \right\} \quad (34)$$

Applying continuity at the rotor discharge

$$\dot{w}_t = \rho_2 w_{a2} (A_2 k_{tr}) \quad (35)$$

where A_2 is the annulus area at the rotor discharge and k_t is a restriction factor to account for the blade trailing edge thickness. The relationship for k_t , developed from data presented by Beer,¹⁷ is

$$k_t = 1 - \frac{2.7}{10^3} \left(\frac{t}{s} 100 \right)^{3.3} \frac{d}{s} \quad (36)$$

¹⁷Beer, R., Aerodynamic Design and Estimated Performance of a Two Stage Curtis Turbine for the Liquid Oxygen Turbopump of the M-1 Engine (NASA CR 54764 A6 C8800-12, 1965), Fig. 20.

where

t = the blade trailing edge thickness

S = blade spacing

a = throat dimension

Eq. (36) is not directly applicable to a rotating blade row due to the increase in flow area resulting from tip clearance. The increased area reduces the effect of area restriction caused by blade trailing edge thickness. If there were no boundary layer, k_t for the rotor would be

$$k_{t_r} = k_t \frac{A_2 + \Delta A_2}{A_2} \quad (37)$$

where ΔA_2 is the tip clearance area. However, experience has shown that there is some area restriction in ΔA_2 due to boundary layer buildup along the stationary walls. For this analysis it was assumed that the annulus area is increased by an amount equal to $0.5 (\Delta A_2)$. The resulting expression for k_{t_r} is

$$k_{t_r} = \left[1 - \frac{2.7}{10^3} \left(\frac{t}{S} 100 \right)^{3.3} \frac{a}{S} \right] \left(\frac{A_2 + 0.5 \Delta A_2}{A_2} \right) \quad (38)$$

Expressions for W_{a_2} and ρ_2 must be obtained to solve Eq. (35). From the energy equation

$$\frac{W_2^2}{2gJc_p} = \frac{W_{a_2}^2 + W_{u_2}^2}{2gJc_p} = \gamma_r \Delta T_{is} \quad (39)$$

Solving Eq. (39) for W_{a_2}

$$W_{a_2} = \left[2gJc_p \eta_r \Delta T_{is} - W_{u_2}^2 \right]^{1/2} \quad (40)$$

Combining Eqs. (32) and (40)

$$W_{a_2} = \left[2gJc_p \eta_r T_{t_E} \left(1 - \left(\frac{P_2}{P_{t_E}} \right)^{\frac{\gamma-1}{\gamma}} \right) - W_{u_2}^2 \right] \quad (41)$$

From the equation of state

$$\rho_2 = \frac{P_2}{RT_2}$$

Substituting Eq. (34) for T_2

$$\rho_2 = \frac{P_2}{RT_{t_E} \left\{ 1 - \eta_r \left[1 - \left(\frac{P_2}{P_{t_E}} \right)^{\frac{\gamma-1}{\gamma}} \right] \right\}} \quad (42)$$

Substituting Eq. (40) and (42) into Eq. (35) for W_{a_2} and ρ_2

$$\frac{\dot{W}_t}{A_2 K_{t_r}} = \left(\frac{P_2}{RT_{t_E} \left\{ 1 - \eta_r \left[1 - \left(\frac{P_2}{P_{t_E}} \right)^{\frac{\gamma-1}{\gamma}} \right] \right\}} \right) \left(2gJc_p \eta_r T_{t_E} \left[1 - \left(\frac{P_2}{P_{t_E}} \right)^{\frac{\gamma-1}{\gamma}} \right] - W_{u_2}^2 \right)^{1/2} \quad (43)$$

Rearranging Eq. (43)

$$\frac{\dot{W}_t \sqrt{T_{t_E}} \sqrt{\frac{R(\gamma-1)}{2\gamma}}}{P_2 A_2 K_{t_r} \sqrt{g}} = \frac{\left\{ \eta_r \left[1 - \left(\frac{P_2}{P_{t_E}} \right)^{\frac{\gamma-1}{\gamma}} \right] - \frac{W_{u_2}^2}{2gJc_p T_{t_E}} \right\}^{1/2}}{\left\{ 1 - \eta_r \left[1 - \left(\frac{P_2}{P_{t_E}} \right)^{\frac{\gamma-1}{\gamma}} \right] \right\}} \quad (44)$$

The left-hand side of Eq. (44) consists of known quantities and is a constant for a given set of operating conditions.

Therefore, with

$$\frac{\dot{W}_t}{P_2 A_2 K_{t_r} \sqrt{g}} \sqrt{\frac{R(\gamma-1)}{2\gamma}} = K$$

and expanding Eq. (44)

$$\left[\eta_r - \frac{1}{2} \left(\frac{2K^2 + 1}{K^2 \left[1 - \left(\frac{p_2}{P_{tE}} \right)^{\frac{\gamma-1}{\gamma}} \right]} \right) \right]^2 = \frac{1 + 4K^2 \left(1 - \frac{W_{u_2}^2}{2gJc_p T_{tE}} \right)}{4K^2 \left[1 - \left(\frac{p_2}{P_{tE}} \right)^{\frac{\gamma-1}{\gamma}} \right]^2} \quad (45)$$

Solving this quadratic equation for η_r yields

$$\eta_r = \frac{1 + 2K^2 \pm \sqrt{1 + 4K^2 \left(1 - \frac{W_{u_2}^2}{2gJc_p T_{tE}} \right)}}{2K^2 \left[1 - \left(\frac{p_2}{P_{tE}} \right)^{\frac{\gamma-1}{\gamma}} \right]} \quad (46)$$

By using an isentropic expansion from P_{tE} to p_2 for which the rotor efficiency would be equal to unity, it can be shown that the minus sign is applicable. With V_{u_2} known, there is

$$W_{u_2} = V_{u_2} - U_2$$

Therefore, all quantities on the right-hand side of Eq. (46) are known with the exception of the ratio p_2/P_{tE} . Multiplying both sides of Eq. (46) by the term containing this ratio, yields

$$\eta_r \left[1 - \left(\frac{p_2}{P_{tE}} \right)^{\frac{\gamma-1}{\gamma}} \right] = \frac{1 + 2K^2 - \sqrt{1 + 4K^2 \left(1 - \frac{W_{u_2}^2}{2gJc_p T_{tE}} \right)}}{2K^2} \quad (47)$$

Eq. (47) must be solved to obtain T_2 . Substitution into Eq. (34) gives

$$T_2 = T_{tE} \left[\frac{\sqrt{1 + 4K^2 \left(1 - \frac{W_{u_2}^2}{2gJc_p T_{tE}} \right)} - 1}{2K^2} \right] \quad (48)$$

With T_{tE} and T_2 known

$$W_2 = \left[2gJc_p(T_{tE} - T_2) \right]^{1/2} \quad (49)$$

Then

$$V_{a_2} = W_{a_2} = \left[W_2^2 - W_{u_2}^2 \right]^{1/2}$$

and

$$V_2 = \left[V_{a_2}^2 + V_{u_2}^2 \right]^{1/2}$$

The total temperature at the rotor exit may be found from

$$T_{t_2} = T_2 + \frac{V_2^2}{2gJc_p} = T_{tP1} - \frac{M_o \omega}{Jc_p \dot{W}_t} \quad (50)$$

where

ω = rotor speed (radians/sec)

The absolute and relative discharge angles are obtained from

$$\alpha_2 = \tan^{-1} \frac{V_{u_2}}{V_{a_2}} \quad (\text{absolute flow})$$

$$\beta_2 = \tan^{-1} \frac{W_{u_2}}{W_{a_2}} \quad (\text{relative flow})$$

Turbine Performance Analysis

In order to evaluate overall turbine performance additional parameters must be obtained.

The theoretical degree of reaction of a stage as defined by Vavra¹⁸ is

$$r^* = \frac{h_1^* - h_2^*}{\Delta h_{is}} \quad (51)$$

r^* represents the ratio of the isentropic enthalpy drop in the rotor to the isentropic enthalpy drop in the stage. The degree of reaction is obtained at the hub, tip, and mean blade positions. With these data it is possible to ascertain which portions of the blade are performing as designed. Additionally, low values of r^* are indicative of small pressure drops through the rotor. For an axial turbine low values of r^* may result in decelerated flow in the rotor. This effect is illustrated in Fig. 14 where W_2 is less than W_1 even when the rotor losses are low.

The head coefficient defined by Vavra¹⁹ is

$$k_{is} = \frac{\Delta h_{is}}{U_1^2/2gJ}$$

k_{is} is a dimensionless parameter relating the total isentropic enthalpy drop to the peripheral speed at the rotor entrance, and is an indication of the work capacity of a stage.

¹⁸Vavra, M. H., Aero-Thermodynamics and Flow in Turbo-machines, (New York, London: John Wiley and Sons, Inc., 1960), Chap. 5, p. 422.

¹⁹Ibid., p. 426.

The power output of the turbine is measured by the dynamometer. The horsepower developed is then obtained from

$$HP = \frac{M_D \omega}{550}$$

where

$$\omega = N \frac{2\pi}{60}$$

All the desired output quantities are converted to referred values by using

$$\Theta = \frac{\gamma R_g T_{tpl}}{\gamma R_g T_s} \quad (52)$$

$\delta = P_{tpl}/P_s$
 where T_s and p_s are 518.7 °R and 14.7 psia, respectively. Since the test fluid is air, and it is considered a perfect gas, Eq. (52) reduces to

$$\Theta = \frac{T_{tpl}}{T_s}$$

Referred values of the data are then

$$(\dot{W}_t)_{ref} = \frac{\dot{W}_t \sqrt{\Theta}}{\delta}$$

$$(M_D)_{ref} = \frac{M_D}{\delta}$$

$$(HP)_{ref} = \frac{HP}{\sqrt{\Theta} \delta}$$

$$(N)_{ref} = \frac{N}{\sqrt{\Theta}}$$

By using referred values, the data obtained at a given pressure ratio may be compared regardless of the inlet conditions. By using referred values, the data obtained at a given pressure ratio may be compared regardless of the inlet conditions.

Pressure and Velocity Surveys

The manometer readings taken during the surveys are used to obtain the total pressure, dynamic head, and pitch angle. The total pressure is obtained directly since it is compared to atmosphere at the manometer tube. The dynamic head is obtained from

$$q = p_1 - p_2$$

Calibration curves are provided with each probe by United Sensor and Control Corporation. These curves were used to obtain the pitch angle and Mach Number effects on the static pressure. The recorded dynamic head had to be corrected for the above mentioned effects. Since the curves were given at relatively low Mach Numbers, extrapolation was necessary to obtain applicable corrections. Unfortunately, no definitive factors for the influence of immersion depth of the sensing holes were available. Unfortunately, no defini-

5. Results and Discussion.

Each of the thirty test runs made was reduced with two different methods. Initially the data were reduced by using the momentum analysis outlined in Section 4. At that

time no attempt was made to satisfy continuity by the use of flow functions. Subsequently the same data were reduced by the analysis based on continuity which is described in the Appendix.

Comparison of the two methods indicated that the axial velocity at the stator discharge, obtained from the momentum analysis, was consistently higher than that obtained from continuity. Since the peripheral component of velocity is obtained in the same manner in both analyses, the resultant velocity obtained from momentum was also too high. The resulting stator loss coefficients were unrealistically low.

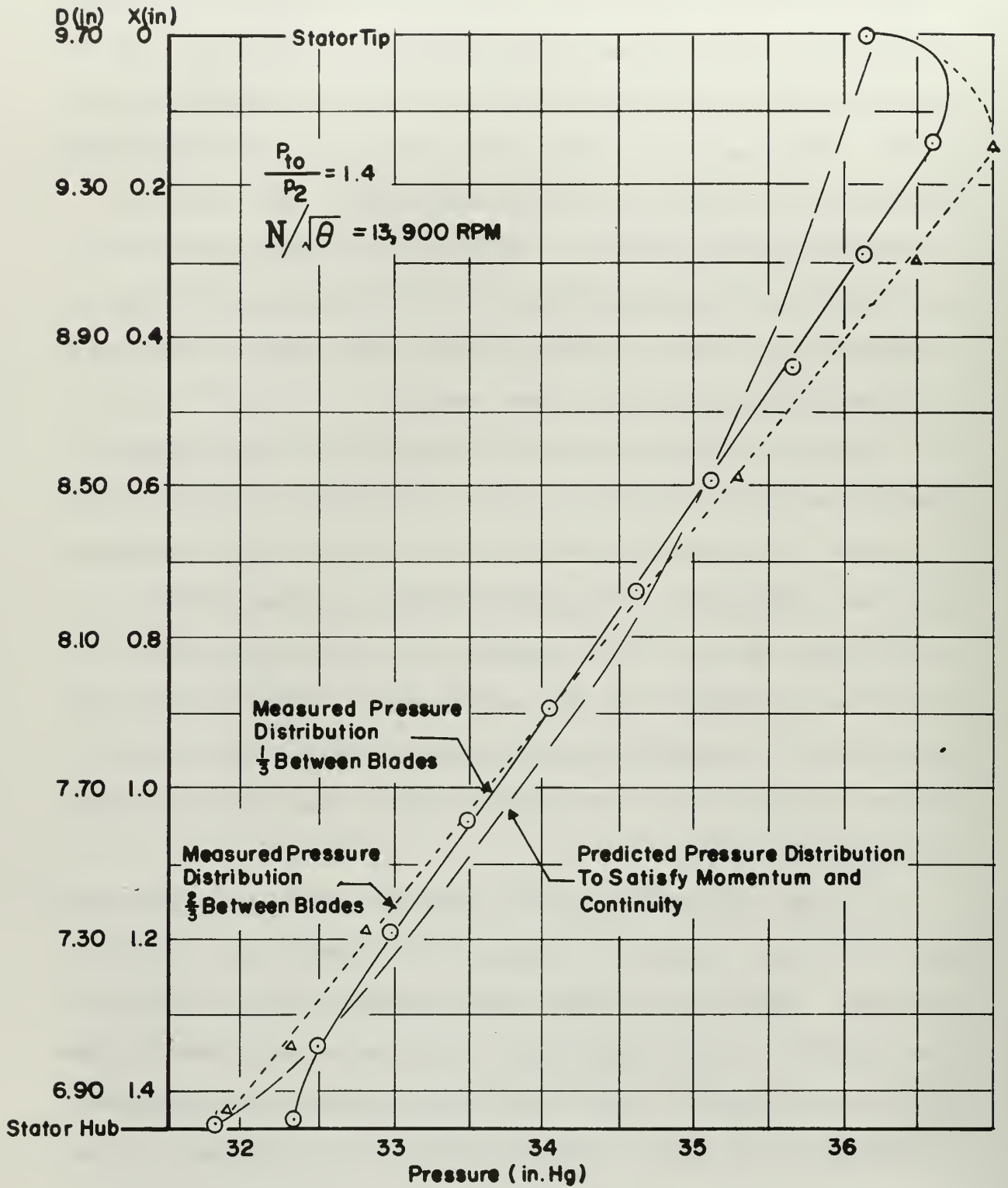
When using the momentum analysis, the axial velocity is a function of the axial forces on the stator assembly. All these forces are measured, with the exception of the pressure force acting on the stator blades. This force is determined from an assumed pressure distribution across the blade row. Initially the pressure was assumed to vary linearly from the stator hub to tip. When the velocities obtained with this pressure distribution were too high, a circular arc distribution was chosen in an attempt to obtain more realistic results. Although this pressure distribution yielded better results for the discharge velocities and angles, the resulting stator efficiencies were over one hundred percent.

In order to obtain more accurate pressure distributions at the stator discharge, flow surveys were made. Although

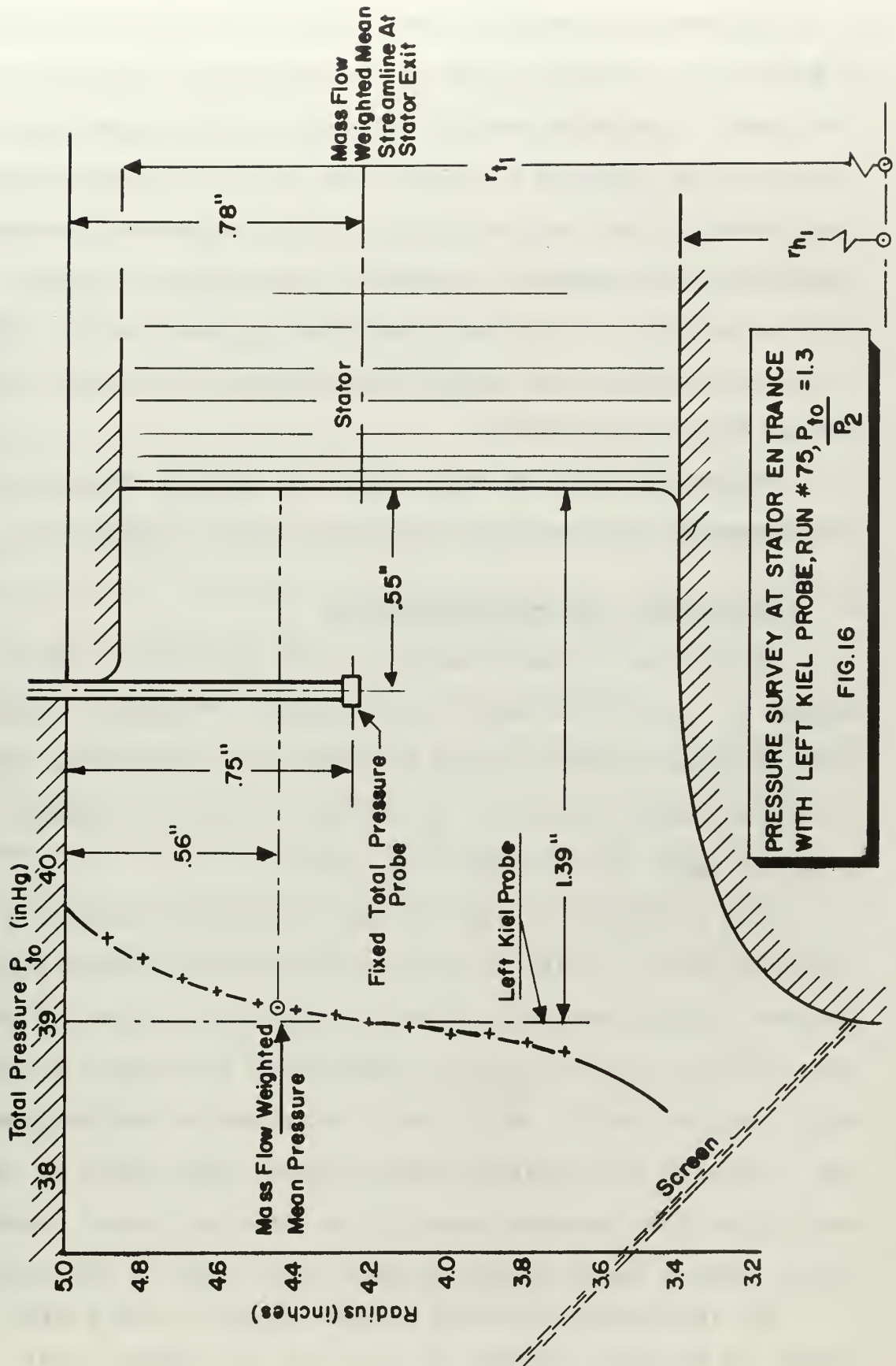
they were taken at several radial and peripheral positions, the results obtained were not conclusive. Only the discharge angles and total pressures could be considered correct. This was due to the probe calibration data which were available to obtain the static pressures. The available calibration curves hold for relatively low Mach Numbers, and had to be extrapolated for the data obtained in the present tests. The resulting corrections were of the order of twenty percent of the total reading.

These surveys did indicate however that the pressure variation from stator hub to tip was nearly linear. Fig. 15 shows the pressure distribution at the stator discharge. However, the pressure measured in the gap between the closure plate and the stator casing was significantly lower than that obtained from the survey data. This would indicate that, although a linear pressure distribution does exist, the net force obtained is higher than that obtained by using Eq. (4a).

It has already been noted that an increase in the force on the stator results in a stator efficiency of over 100 percent. However, from the total pressure survey ahead of the stator, it was found that the pressure indicated by the six fixed probes is lower than the average total pressure existing at the inlet. This results in an increase in the stator efficiency. The actual pressure distribution obtained from this survey is shown in Fig. 16.



STATOR DISCHARGE SURVEY
FIG. 15



PRESSURE SURVEY AT STATOR ENTRANCE
 WITH LEFT KIEL PROBE, RUN # 75, $P_{10} / P_2 = 1.3$
 FIG. 16

The analysis shown in Section 4 was given to indicate a method that could be used if the pressure distribution at the stator discharge were not known. A linear pressure distribution was assumed to obtain the initial values and then the iteration was carried out to achieve agreement between continuity and momentum. However, this analysis failed for the majority of the data recorded in these tests. This is due to the low loss coefficients obtained from the momentum portion of the analysis.

Tables II, III, IV and V show the results obtained from the momentum and continuity analysis for run number 79.

6. Conclusions and Recommendations.

Variations in the pressure at the flow nozzle inlet affect all other readings significantly. Therefore, each item of data recorded should be taken at a particular value of nozzle inlet pressure. If average values of certain data are used, data scatter may result.

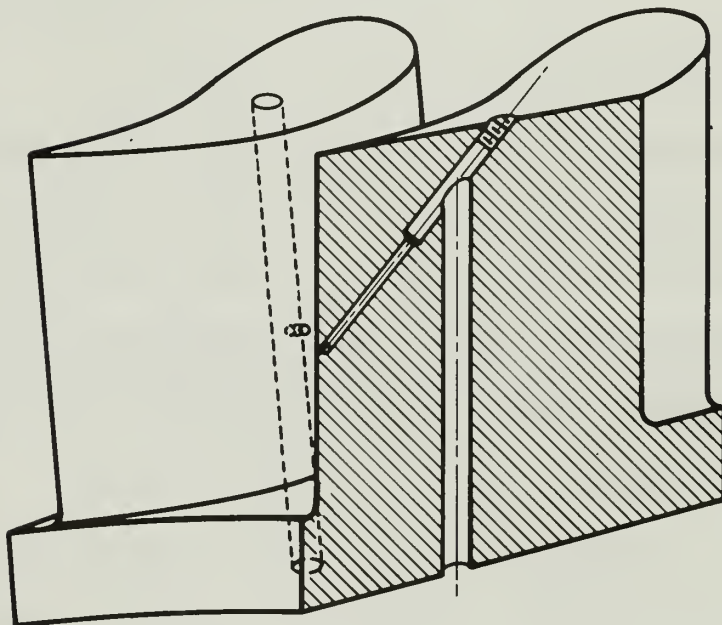
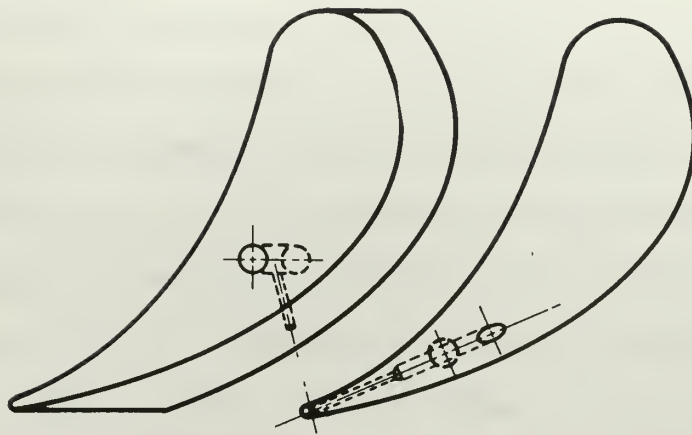
Data reduction by the methods outlined in Section 4 requires that all data be taken as accurately as possible. However, errors resulting from low indicated values of total inlet pressure will still be responsible for stator losses which are too small. Since this installation was designed for a turbine with smaller blade heights than those of the Mod II, the flow passage area of the floating stator assembly is almost equal to the annulus area ahead of the stator.

The installation of the screens ahead of the stator causes an entropy gradient to exist at the stator inlet.

Therefore, the assumption of uniform flow conditions is not valid. This difficulty could be overcome by increasing the size of the flow passage. The velocity in this region would then be reduced and the total pressure variation at the stator inlet would be reduced also. If the inlet total pressure is obtained accurately the method outlined in Section 4 will yield accurate results.

A further improvement could be made if the actual throat pressure were measured. This could be done by placing static taps at the hub, mean, and tip locations along a blade passage. Then the actual throat pressure could be used in the flow function rather than the stator discharge pressure. Fig. 17 is an illustration of how the pressure taps could be located at the throat passage.

Although it has been found that the stator discharge conditions may be predicted within three percent using a momentum analysis alone, it is felt that better results could be achieved with the analysis presented in Section 4, if the recommended changes in instrumentation were made.



STATOR THROAT PRESSURE TAPS
FIG. 17

GENERAL RESULTS CONTINUITY

VELOCITIES (FT/SEC)
 RUN NUMBER 79

| PCINT | V1 | V2 | W1 | W2 | U1 |
|-------|-----------|-----------|-----------|-----------|-----------|
| 1 | 579.73145 | 201.66962 | 235.15054 | 391.20850 | 418.05078 |
| 2 | 576.14990 | 207.20213 | 222.46715 | 396.11133 | 437.78369 |
| 3 | 575.85014 | 217.74500 | 214.44550 | 393.60205 | 458.24805 |
| 4 | 574.86818 | 226.35335 | 207.20920 | 392.97559 | 474.32666 |
| 5 | 573.58521 | 238.50925 | 201.64336 | 394.10571 | 494.79102 |
| 6 | 574.13232 | 263.14331 | 201.76196 | 363.18286 | 495.52173 |

TEMPERATURES (DEG R)

| PCINT | PLENUM TOTAL | STATOR DISCHARGE | ROTOR DISCHARGE | ISENTROPIC FROM T1 | TOTAL DISCHARGE |
|-------|-----------------|---------------------|--------------------|-----------------------|--------------------|
| 1 | 565.85156 | 527.88501 | 529.75366 | 526.49048 | 533.13794 |
| 2 | 565.68091 | 538.05884 | 529.12354 | 526.40503 | 532.69604 |
| 3 | 565.85156 | 538.25439 | 529.19263 | 526.41064 | 533.13794 |
| 4 | 565.85156 | 538.35034 | 529.07593 | 526.38867 | 533.32936 |
| 5 | 565.51025 | 538.13354 | 528.59619 | 525.99146 | 533.32983 |
| 6 | 565.68091 | 538.25195 | 530.66724 | 526.03784 | 536.42920 |

FLOW ANGLES (DEGREES FROM AXIAL) CONTINUITY

| PCINT | ALPHA 1 | ALPHA 2 | BETA 1 | BETA 2 | DELTA BETA |
|-------|----------|----------|----------|-----------|------------|
| 1 | 70.02647 | 21.72693 | 32.63414 | -61.38754 | 94.02168 |
| 2 | 70.02522 | 25.36276 | 27.78581 | -61.79211 | 89.57793 |
| 3 | 69.89960 | 20.90891 | 22.64500 | -61.66359 | 84.30859 |
| 4 | 70.02046 | 24.56361 | 18.56224 | -61.68404 | 80.24628 |
| 5 | 69.93495 | 28.34132 | 12.59791 | -61.66217 | 74.26007 |
| 6 | 69.93703 | 44.52836 | 12.52937 | -58.90594 | 71.43553 |

EFFICIENCIES AND LOSSES

| PCINT | EFFICIENCY TOTAL STATIC | ZETA ROTOR | ZETA STATOR |
|-------|-------------------------|------------|-------------|
| 1 | 0.80277 | 0.20385 | 0.04830 |
| 2 | 0.80739 | 0.17219 | 0.05517 |
| 3 | 0.80051 | 0.17734 | 0.05016 |
| 4 | 0.79334 | 0.17278 | 0.05345 |
| 5 | 0.78351 | 0.16754 | 0.05485 |
| 6 | 0.71133 | 0.29649 | 0.05231 |

TABLE III.

GENERAL RESULTS MOMENTUM

VELOCITIES (FT/SEC)
RUN NUMBER 79

| POINT | V1 | V2 | W1 | W2 | U1 |
|-------|-----------|-----------|-----------|-----------|-----------|
| 1 | 592.76221 | 201.66962 | 265.65869 | 391.20850 | 418.05078 |
| 2 | 586.11572 | 207.20213 | 247.13321 | 396.11133 | 437.78369 |
| 3 | 587.22534 | 217.74500 | 243.25128 | 393.60205 | 458.24805 |
| 4 | 587.01147 | 226.35335 | 238.79266 | 392.97559 | 474.32666 |
| 5 | 584.34204 | 238.50925 | 230.46890 | 394.10571 | 494.79102 |
| 6 | 585.39941 | 263.14331 | 231.88907 | 363.18286 | 495.52173 |

TEMPERATURES (DEG R)

| POINT | PLENUM TOTAL | STATOR DISCHARGE | ROTOR DISCHARGE | ISENTROPIC FROM T1 | TOTAL DISCHARGE |
|-------|--------------|------------------|-----------------|--------------------|-----------------|
| 1 | 565.85156 | 536.61353 | 529.75366 | 525.24585 | 533.13794 |
| 2 | 565.68091 | 537.09497 | 529.12354 | 525.46191 | 532.69604 |
| 3 | 565.85156 | 537.15723 | 529.19263 | 525.33765 | 533.13794 |
| 4 | 565.85156 | 537.17822 | 529.07593 | 525.24268 | 533.33936 |
| 5 | 565.51025 | 537.09692 | 528.59619 | 524.97827 | 533.32983 |
| 6 | 565.68091 | 537.16479 | 530.66724 | 524.97534 | 536.42920 |

TABLE IV.

FLOW ANGLES (DEGREES FROM AXIAL) MOMENTUM

RUN NUMBER 79

| POINT | ALPHA 1 | ALPHA 2 | BETA 1 | BETA 2 | DELTA BETA |
|-------|----------|----------|----------|-----------|------------|
| 1 | 66.80782 | 21.72693 | 28.51198 | -61.38754 | 89.89952 |
| 2 | 67.49745 | 25.36276 | 24.81186 | -61.79211 | 86.60397 |
| 3 | 67.06747 | 30.90891 | 19.84192 | -61.66359 | 81.50551 |
| 4 | 66.98535 | 34.56361 | 16.03537 | -61.68404 | 77.71941 |
| 5 | 67.22206 | 38.34132 | 11.00112 | -61.66217 | 72.66328 |
| 6 | 67.10764 | 44.53836 | 10.88014 | -58.90594 | 69.78609 |

EFFICIENCIES AND LOSSES

| POINT | EFFICIENCY TOTAL STATIC | ZETA ROTOR | ZETA STATOR |
|-------|-------------------------|------------|-------------|
| 1 | 0.80277 | 0.26132 | 0.00503 |
| 2 | 0.80739 | 0.21890 | 0.02220 |
| 3 | 0.80051 | 0.23006 | 0.01239 |
| 4 | 0.79334 | 0.22961 | 0.01311 |
| 5 | 0.78351 | 0.21853 | 0.01907 |
| 6 | 0.71133 | 0.34135 | 0.01474 |

TABLE V.

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APPENDIX

COMPUTER PROGRAM TO OBTAIN TURBINE PERFORMANCE

The computer program (TTR2) is composed of an executive routine and 11 subroutines. Two of these subroutines appear as subroutines of STATOR.

The comment cards give the program name, purpose, and indicate the units of each input item.

The executive routine reads the number of data sets to be reduced and provides the calling sequence for the subroutines. Any number of data sets may be reduced. However each set may consist of a maximum of 50 data points.

Subroutine INPUT is then called. This subroutine reads in the input data. The input data consist of the following entries:

Single Entries

1. Number of test run (NRUN)
2. Axial and radial clearance (AXCLR and RADCLR)
3. Barometric pressure
4. Number of data points
5. Temperatures in the Cascade Lab and Control Room

Multiple Entries

6. Pressure change across the flow nozzle
7. Total pressure flow nozzle
8. Total temperature, flow nozzle
9. Plenum total pressure-average of the six Kiel probes

10. Hub static pressure
11. Tip static pressure
12. Inlet total temperature
13. Reference pressure for, PNOZ, PHUB, PSPL, PHD
14. Rotational speed
15. Stator torque
16. Stator axial force
17. Dynamometer torque
18. Static pressure in hood-use p_{atm} when operating without the hood
19. Reference pressure for p_{tip} , p_{14-20}
20. Plenum static pressure
21. Closure plate axial force
22. Closure plate torque
23. Reference pressure for six plenum Kiel probes
24. Pressures around end of shroud

All pressures are entered as they are read on the manometer board with the exception of the actual pressure differential across the flow nozzle in inches of water (DH).

The multiple entries are entered with eight data points per card. After the input data have been read, they are printed out before exiting the subroutine.

In the following subroutine descriptions, the item numbers refer to the numbers listed on the left side of the enclosed print-out.

SETCON is the subroutine that establishes all the constants that are used in the program. All quantities in this subroutine have been discussed, with the exception of

item 30. This factor accounts for the effect of blade trailing edge thickness and will be used in the continuity analysis of STATOR.

Following SETCON, the input data are changed from subscripted to non-subscripted variables, which are denoted by Q. This is done to avoid the use of subscripts in all subroutines.

CONVERT is the subroutine that changes all units to a single system (psia, lb., ft., sec., $^{\circ}$ R). All areas remain in square inches.

Item 2 is the equation used to convert the temperatures, from millivolts, to degrees Rankine.

Item 3 is a conversion factor which accounts for the variation of water density due to temperature changes.

Items 5 and 8 are conversion factors which account for the temperature affects on the density of mercury.²⁰

Items 26 through 29 depend on the calibration methods used to calibrate the torque and axial force on the stator assembly and closure plate.

Item 30 depends on the range over which the dynamometer is calibrated.

FLORAT is the subroutine which computes the turbine flowrate. The formula development is given in Section 3.

STATOR is the subroutine which computes the flow characteristics at the stator discharge. In this subroutine

²⁰ Stearnes, R. F., et. al., Flow Measurement with Orifice Meters (New York: Van Nostrand Co., Inc., 1951), p.261

either MOMENT or CONTIN is called to compute the absolute discharge velocity. The procedure followed in MOMENT has been outlined in Section 4. The continuity analysis is identical with that given in Section 4 for the rotor, with the exception that absolute velocities are used. The absolute velocity at the stator discharge is then obtained from

$$V_1 = \left[2gJc_p(T_{t_{p1}} - T_1) \right]^{1/2}$$

where T_1 is obtained from

$$T_1 = \frac{T_{t_{p1}}}{2K^2} \left\{ \left[1 + 4K^2 \left(1 - \frac{V_{u_1}^2}{2gJc_p T_{t_{p1}}} \right) \right]^{1/2} - 1 \right\}$$

with

$$K = \frac{\dot{w}_t \sqrt{T_{t_{p1}}}}{A_1 K_{ts} P_1} \sqrt{\frac{R}{g} \frac{\gamma-1}{2\gamma}}$$

Then

$$V_{a_1} = \sqrt{V_1^2 - V_{u_1}^2}$$

The other flow characteristics at the stator discharge are determined as shown in Section 4 under Stator Analysis.

ROTOR is the subroutine used to compute rotor exit conditions. The formula development for this subroutine may be found in Section 4 under Rotor Analysis.

PERFORM is the subroutine used to compute the various performance parameters outlined in Section 4 under Turbine Performance Analysis.

Following PERFORM, the executive routine converts the Q variables back into a one-dimensional array for output.

OUTPTA is a subroutine that gives a printout of all pertinent performance data. This printout consists of the following items:

1. Flow rate
2. RPM
3. Pressure ratio
4. Horsepower
5. k_{is}
6. Referred flow rate
7. Referred torque
8. Referred power
9. Referred speed
10. Absolute and relative discharge velocities and angles for the stator and rotor
11. Total plenum temperature
12. Static temperature at stator and rotor discharge
13. Static temperature achieved by an isentropic expansion from T_{tpl}
14. Total temperature at the rotor discharge
15. Flow deflection in the rotor ($\Delta \beta$)
16. Turbine efficiency (Total to static)
17. Rotor and stator loss coefficients
18. Degree of reaction at the hub, mean, and tip
19. Absolute and relative Mach Numbers at the stator discharge
20. Pressure ratio across the stator
21. Stator blockage factor (ξ_s)

OUTPUT prints out Items 3, 5, 17, 6, 7, 8, 9, 19 of OUTPTA in a report format. The turbine tested, radial and axial clearance, test run number, date of test, and data reduction method are given in the heading.

A sample program is shown in this appendix. Two stator subroutines are shown. The second contains the iterative process explained in Section 4.

```

0001 PROGRAM TTR2 1
0002 COMPUTES EXPERIMENTAL PERFORMANCE CHARACTERISTICS OF THE
0003 TURBINE TEST RIG
0004 INPUT DATA DH(IN H2O), PNOZ(IN HG), TNOZ(MV), PTPL(IN HG), PHUB(IN HG)
0005 PTIP(IN HG), PBPP(IN HG), TTPL(MV), PATM(IN HG), RPM(RPM), TORQR(COUNTS)
0006 AXIL(COUNTS), DYNAR(COUNTS), PHD(IN HG), PRFL(IN HG), PSPL(IN HG),
0007 CLAXIL(COUNTS), CLTRQR(COUNTS), PRF2(IN HG), THD(MV), P14 THRU P17
(IN HG), TCL(DEGREES F), TCR(DEGREES F),
DIMENSION DH(50), PNOZ(50), TNOZ(50), PTPL(50), PHUB(50), PTIP(50),
1PBPP(50), TTPL(50), PATM(50), RPM(50), TORQR(50), AXIL(50), DYNAR(50),
2PHD(50), PRFL(50), PSPL(50), CLAXIL(50), CLTRQR(50), PRF2(50),
3, P18(50), P15(50), P16(50), P17(50), P19(50), P20(50)
DIMENSION FLOWT(50), PR(50), HP(50), XKIS(50), COEFL(50), COEFFM(50),
1COEFP(50), COEFS(50), V1(50), V2(50), W1(50), W2(50), U1(50), U2(50),
2T1(50), T2(50), T2IS(50), TT2(50), ALPH1(50), ALPH2(50), BETA1(50),
3BETA2(50), DBETA(50), ETA(50), ETAT(50), ZETAR(50), ZETAS(50), ZETA(50),
4REACHB(50), REACMN(50), REACTP(50), VMI(50), WMI(50), PHI(50),
5 XI(50), PRS(50)
COMMON G,CR,C2,C3,RM1,RM2,AAX,ATH,QFLOWT,QTTP,TLIS,QI1,QTPL,
1QPTIP,QPHUB,P1,QPI8,QPI5,QPI7,QPRS,TORQ,CLTORQ,QV1,VAL,VU1,
2QZETAS,QPHI,QXI,GAM,QALPH1,QBETA1,QRPM,QU1,QW1,QVMI,QWMI,QPHD,
3 CLFAX, FAX, RTIP1, RHUB1, SKT, PAMB, QP19, QP20
READ 20, MM
20 FORMAT(I10)
CALL INPUT(NRUN, AXCLR, RADCLR, PBAR, N, TCL, TCR, DH, PNOZ, TNOZ, PTPL,
1PHUB, PTIP, TTPL, PATM, RPM, TORQR, AXIL, DYNAR, PHD, PRF1, PSPL,
2CLAXIL, CLTRQR, PRF2, P18, P15, P16, P17, P19, P20)
CALL SETCON(BETA, CN, DN, AAX, AAXR, ATH, CR, GAM, G, CP, CJ, C1, C2, C3, C4,
1C5, RTIP1, RHUB1, RTIP2, RHUB2, RM1, RM2, RADCLR, SKT, RKT)
DO 90 I=1, N
QDH=DH(I)
QPNOZ=PNOZ(I)
QTNOZ=TNOZ(I)
QPTPL=PTPL(I)
QPHUB=PHUB(I)
QPTIP=PTIP(I)
QPBPP=PBPP(I)
QTTP=TTPL(I)
QPATM=PATM(I)
QRPM=RPM(I)
QTORQR=TORQR(I)
QAXIL=AXIL(I)
QDYNAR=DYNAR(I)
QPHD=PHD(I)
QPRF1=PRF1(I)
QPSPL=PSPL(I)
QCLAXIL=CLAXIL(I)

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| 0027 | QLTRQR=CLTRQR(I) |
| 0028 | QPRF2=PRF2(I) |
| 0029 | QP18=PI8(I) |
| 0030 | QP15=PI5(I) |
| 0031 | QP16=PI6(I) |
| 0032 | QP17=PI7(I) |
| 0033 | QP19=PI9(I) |
| 0034 | QP20=PI20(I) |
| 0035 | CALL CNVERT(QDH,QPNOZ,QTNOZ,OPTPL,QPHUB,OPTIP, QTTPL,QPATM, 1 QRP, QTRQR,QAXIL,QDYNAR,QPHD,QPRF1,QPSPL, QLTRQR, QCLXIL,P1,P2, 2 QP18,QP15,QP16,QP17,TORQ,CLTORQ,FAX,CLFAX,DYNA,PBAR,QPRF2,TCL, 3 YCR,C5,PAMB,QP19,QP20) 1 QFLOWT) |
| 0036 | CALL STATOR |
| 0037 | CALL ROTOR(G,CR,CP,CJ,C2,C3,RM1,RM2,AAXR,QFLOWT, QTTPL,QRPM,DYNA, 1 QDP,P1,P2,QT1,QT2, QTT2,QT2IS,VAL,VU1,QV2,QW1,QW2,QU1,QU2,QALPH2, 2 QBETA1,QBETA2,C4,RKT) |
| 0039 | CALL PERFORM(G,CR,CP,C1,C2,C3,GAM, QTTPL,OPTPL,QPHUB,OPTIP,QPHD, 1 QRP, QFLOWT,DYNA,QDP,P2,T1IS,QT1,QT2IS,QPR,QETA,QXKIS,QCOEFL, 2 QCOEFM,QCOEFP,QCOEFS,QEACMN,QEACHB,QEACTP,QZETAR,QALPH1,QALPH2, 3 QBETA1,QBETA2,QDBETA,QHP,QU2,QW1,QW2) FLOWT(I)=QFLOWT |
| 0040 | PR(I)=QPR |
| 0041 | HP(I)=QHP |
| 0042 | XKIS(I)=QXKIS |
| 0043 | COEFL(I)=QCOEFL |
| 0044 | COEFM(I)=QCOEFM |
| 0045 | COEFP(I)=QCOEFP |
| 0046 | COEFS(I)=QCOEFS |
| 0047 | V1(I)=QV1 |
| 0048 | V2(I)=QV2 |
| 0049 | W1(I)=QW1 |
| 0050 | W2(I)=QW2 |
| 0051 | U1(I)=QU1 |
| 0052 | U2(I)=QU2 |
| 0053 | TTPL(I)=QTTPL |
| 0054 | T1(I)=QT1 |
| 0055 | T2(I)=QT2 |
| 0056 | T2IS(I)=QT2IS |
| 0057 | TT2(I)=QTT2 |
| 0058 | ALPH1(I)=QALPH1 |
| 0059 | BETA1(I)=QBETA1 |
| 0060 | BETA2(I)=QBETA2 |
| 0061 | DBETA(I)=QDBETA |
| 0062 | ETA(I)=QETA |
| 0063 | ZETA(I)=QZETA |
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ZETAS(I)=QZETAS  
REACHB(I)=QEACHB  
REACMN(I)=QEACMN  
REACTP(I)=QEACTP  
VMI(I)=QVMI  
WMI(I)=QWMI  
PHI(I)=QPHI  
XI(I)=QXI  
PRS(I)=QPRS  
PTPL(I)=QPTPL  
90 CONTINUE  
CALL OUTPUTA (NRUN,AXCLR,RADCLR,FLOWT,PR,HP,XKIS,COEFL,COEFM,COEFP,  
1COEFS,V1,V2,W1,W2,U1,U2,T1,T2,T2IS,TT2,ALPH1,ALPH2,BETA1,BETA2,  
2DBETA,ETA,XI,PRS,RPM,TTPL,N)  
3PHI,  
CALL OUTPUT(NRUN,AXCLR,RADCLR,N,PR,XKIS,ETA,COEFL,COEFM,COEFP,  
1COEFS,REACHB,REACTP,PTPL)  
99 CONTINUE  
END
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0001 SUBROUTINE INPUT (NRUN,AXCLR,RADCLR,PBAR,N,TCL,TCR,DH,PNOZ,PHUB,PTIP,PSPL,
1PTPL,PHUB,PTIP,PRF2,PI8,PI5,PI6,PI7,PI9,P20)
0002 T TPL,PAATM,RPM,TORQR,AXIL,DYNAR,PHD,PRF1,PSPL,
2CLAXIL,CLTRQR,DH(50),PNOZ(50),TNOZ(50),PTPL(50),TORQR(50),AXIL(50),DYNAR(50),
DIMENSION DH(50),PATM(50),RPM(50),TORQR(50),AXIL(50),CLTRQR(50),PRF2(50),
1 TPL(50),PSPL(50),CLAXIL(50),CLTRQR(50),PI9(50),P20(50)
2PHD(50),PRF1(50),PI5(50),PI6(50),PI7(50),PI9(50),P20(50)
3PI8(50),PI5(50),PI6(50),PI7(50),PI9(50),P20(50)
READ (5,101)NRUN
READ (5,102)AXCLR,RADCLR
READ (5,102)PBAR
READ (5,101)N
READ (5,102)TCL,TCR
READ (5,102)(DH(I),I=1,N)
READ (5,102)(PRF1(I),I=1,N)
READ (5,102)(PTIP(I),I=1,N)
READ (5,102)(PI5(I),I=1,N)
READ (5,102)(PI6(I),I=1,N)
READ (5,102)(PI7(I),I=1,N)
READ (5,102)(PI8(I),I=1,N)
READ (5,102)(PI9(I),I=1,N)
READ (5,102)(P20(I),I=1,N)
READ (5,102)(PTPL(I),I=1,N)
READ (5,102)(PATM(I),I=1,N)
READ (5,102)(PNOZ(I),I=1,N)
READ (5,102)(PSPL(I),I=1,N)
READ (5,102)(PHUB(I),I=1,N)
READ (5,102)(PHD(I),I=1,N)
READ (5,102)(TNOZ(I),I=1,N)
READ (5,102)(TTP( I),I=1,N)
READ (5,102)(AXIL(I),I=1,N)
READ (5,102)(TORQR(I),I=1,N)
READ (5,102)(DYNAR(I),I=1,N)
READ (5,102)(RPM(I),I=1,N)
READ (5,102)(CLAXIL(I),I=1,N)
READ (5,102)(CLTRQR(I),I=1,N)
WRITE (6,105)NRUN,PBAR
WRITE (6,107)(DH(I),I=1,N)
WRITE (6,107)(PRF1(I),I=1,N)
WRITE (6,107)(PTIP(I),I=1,N)
WRITE (6,107)(PI5(I),I=1,N)
WRITE (6,107)(PI6(I),I=1,N)
WRITE (6,107)(PI7(I),I=1,N)
WRITE (6,107)(PI8(I),I=1,N)
WRITE (6,107)(PI9(I),I=1,N)
WRITE (6,107)(P20(I),I=1,N)
WRITE (6,107)(PTPL(I),I=1,N)
WRITE (6,107)(PATM(I),I=1,N)
WRITE (6,107)(PNOZ(I),I=1,N)
WRITE (6,107)(PSPL(I),I=1,N)
WRITE (6,107)(PHUB(I),I=1,N)
WRITE (6,107)(PHD(I),I=1,N)
WRITE (6,107)(TNOZ(I),I=1,N)
WRITE (6,107)(TTP( I),I=1,N)
WRITE (6,107)(AXIL(I),I=1,N)
WRITE (6,107)(TORQR(I),I=1,N)
WRITE (6,107)(DYNAR(I),I=1,N)
WRITE (6,107)(RPM(I),I=1,N)
WRITE (6,107)(CLAXIL(I),I=1,N)
WRITE (6,107)(CLTRQR(I),I=1,N)

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WRITE(6,107) (PATM(I), I=1,N)
WRITE(6,107) (PNOZ(I), I=1,N)
WRITE(6,107) (PSPL(I), I=1,N)
WRITE(6,107) (PHUB(I), I=1,N)
WRITE(6,107) (PHD(I), I=1,N)
WRITE(6,107) (TNOZ(I), I=1,N)
WRITE(6,107) (TTPL(I), I=1,N)
WRITE(6,107) (AXIL(I), I=1,N)
WRITE(6,107) (TORQR(I), I=1,N)
WRITE(6,107) (DYNAR(I), I=1,N)
WRITE(6,107) (RPM(I), I=1,N)
WRITE(6,107) (CLAXIL(I), I=1,N)
WRITE(6,107) (CLTRQR(I), I=1,N)
FORMAT(I10)
FORMAT(8F10.4)
FORMAT(IH1, //5X, 11H RUN NUMBER, I5, //5X, 40H INPUT DATA, CARD IMAGE,
1 LINE = 1 CARD //, 5X, 6H PBAR=, F6.3/)
FORMAT(8F15.5)
RETURN
END

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CARD IMAGE,

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SUBROUTINE SETCUN(BETA,CN, DN, AAX, AAXR, ATH, CR, GAM, G, CP, CJ, C1, C2,
1 C3, C4, C5, RTIP1, RHUB1, RTIP2, RHUB2, RM1, RM2, RADCLR, SKI, RKT)
BETA=.533
CN=1.05
DN=4.25
AAX=37.6499
AAXR=41.7729
ATH=12.70512
CR=53.34
GAM=1.401
G=32.174
CP=0.24
CJ=778.16
C1=2.*G*CJ
C2=2.*G*CJ*CP
C3=(GAM-1.)/GAM
C4=1./C3
C5=.0361269
RTIP1=9.701/2.
RHUB1=6.795/2.
RTIP2=9.836/2.
RHUB2=6.600/2.
RM1=SQRT((RTIP1**2+RHUB1**2)/2.)
RM2=SQRT((RTIP2**2+RHUB2**2)/2.)
THKS=.048
SSTAT=1.36378
ASTAT=.4635
THKRR=.048
SROTR=1.4343
AROTR=.540
SKI=1.-(2.7/1000.)*(THKS/SSTAT*100.)**3.3*ASTAT/SSTAT
RKT=(1.-(2.7/1000.))*((THKR/SROTR*100.)**3.3*AROTR/SROTR)*((AAXR+
13.14159*RADCLR*(RTIP2+RADCLR/2.))/AAXR)
13. RETURN
END

```

0001 SUBROUTINE  CNVERT (DH,PNOZ,TNOZ,PIPL,PHUB,PTIP,TTPL,PATM,RPM,
1TORQR,AXIL,DYNAR,PHD,PRF1,PSPL,CLTRQR,CLAXIL,PI,P2,P18,P15,P16,
2P17,TORQ,CLTORQ,FAX,CLFAX,DYNA,PBAR,PRF2,TCL,TCR,C5,PAMB,P19,P20)
TEMP(X)=32.+35.98*X-.435*X**2
GH20=1.00013+TCR*.078/1000.-TCR**2.*.0014/1000.
DH=GH20*C5
CHGC=.49280-TCL*.5/10000.
PAMB=PBAR
PAMB=PAMB*CHGC
CHGCR=.49280-TCR*.5/10000.
PNOZ=CHGCR*(PATM-PNOZ)+PAMB
PTPL=CHGCR*(PRF2-PTPL)+PAMB
PHUB=CHGCR*(PATM-PHUB)+PAMB
PTIP=CHGCR*(PRF1-PTIP)+PAMB
P18=CHGCR*(PRF1-P18)+PAMB
P15=CHGCR*(PRF1-P15)+PAMB
P16=CHGCR*(PRF1-P16)+PAMB
P17=CHGCR*(PRF1-P17)+PAMB
P19=CHGCR*(PRF1-P19)+PAMB
P20=CHGCR*(PRF1-P20)+PAMB
PSPL=CHGCR*(PATM-PSPL)+PAMB
PHD=CHGCR*(PATM-PHD)+PAMB
EPS=0.0
P1=((PTIP+PHUB)/2.)*(1.+EPS)
P2=PHD
TTPL=TEMP(TTPL)+459.7
TNOZ=TEMP(TNOZ)+459.7
TORQ=.3*TORQR/12.
CLTORQ=.2207*CLTRQR/12.
FAX=.08*AXIL
CLFAX=CLAXIL
DYNA=DYNAR/30.
RETURN
END

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0001 SUBROUTINE FLORAT (CR,C5,DN,CN,BETA,PNOZ,PSPL,TNOZ,TTPL,PHD,DH,
1FLOWT)
0002 ALPHN=1+.00252*(TNOZ-527.7)/100.
0003 Z=1.9+.24*((TNOZ-459.7)/100.-1.)
0004 X=DH/PNOZ
0005 XR=1.-X
0006 B=(4.25/7.975)**4.
0007 Y=SQR((XR**(2./1.4))*3.5*(1.-XR**(.4/1.4))/(1.-XR)*(1.-B)/(1.
1-B*XR**(.2./1.4)))
0008 FLOW=.8633*DN**2*ALPHN*CN*Y*SQR(PNOZ*DH/TNOZ)
0009 PRL=PHD/PSPL
0010 FLOWS=SQR((1.-PRL)/4.0791)
0011 FLOWL=.116*FLOWS*PSPL/SQR(TTPL)
0012 FLOWT=FLOW-FLOWL
0013 RETURN
0014 END

```



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0001 SUBROUTINE STATOR
0002 CGMMON G,CR,C2,C3,RM1,RM2,AAX,ATH, FLOWT, TTPL, TIIS, T1, PTPL,
1 PTIP, PHUB, P1, P18, P15, P17, PRS, TORQ, CLTORQ, VI, VAI, VUI,
2 ZETAS, PHI, XI, GAM, ALPH1, BETAI, RPM, U1, W1, VMI, PHD,
3 CLFAX, FAX, RTIPI, RHUB1, SKI, PAM9, PI9, P2C
PBAR=PAMB
CALL CONTING, C2, AAX, RM1, PHD, PTIP, P1, PHUB, P18, P15, P16, P17, FLOWT,
1 TORQ, CLFAX, FAX, TTPL, VUI, VAI, V1, ALPH1, T1, RTIPI, RHUB1, CLTORQ,
2 RADCLR, PBAR, DELP1, C3, SKI, CR, P19, P2C)
PRS=PI/PTPL
TIIS=TTPL*PRS*C3
ZETAS=(T1-TIIS)/(TTPL-TIIS)
PHI=FLOWT/PTPL/ATH *SQRT (TTPL*CR/G)
98 XI=PHI/SQRT(2./C3*(PRS**((GAM+1.)/GAM))-PRS**((GAM+1.)/GAM))
RAD=.10472*RPM
U1=RM1*RAD/12.
WUI=VUI-U1
WAI=VAI
BETAI=ATAN (WUI/WAI)
WI=WAI/COS (BETAI)
VMI=VI/SQRT (GAM*G*CR*TI)
WMI=W1/SQRT (GAM*G*CR*TI)
RETURN
0019 END

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0001 SUBROUTINE STATOR
0002 COMMON G,CR,C2,C3, RM1, RM2, AAX, ATH, FLOWT, TTPL, TIIS, T1, PTPL,
0003 PTIP, PHUB, P1, P18, P15, P16, P17, PRS, TORQ, CLTORQ, V1, VAL, VU1,
0004 ZETAS, PHI, XI, GAM, ALPH1, BETA1, RPM, U1, W1, VMI, PHD,
0005 CLFAX, FAX, RTIPI, RHUB1, SKT, PAMR, P19, P20
0006 PBAR=PAMB
0007 R=0.0
0008 EPS=R
0009 P1=((PHUB+PTIP)/2.)*(1.+EPS)
0010 CALL MOMENT(G,C2,AAX,RM1,PHD,PTIP,P1,PHUB,P18,P15,P16,P17,
0011 FLOWT,TORQ,CLFAX,FAX,TTPL,VU1,VAL,V1,ALPH1,T1,RTIPI,RHUB1,CLTORQ,
0012 2RADCLR,PBAR,DELP1,C3,SKT,CR,EPS,P19,P20)
0013 PRS=P1/PTPL
0014 TIIS=TTPL*PRS*C3
0015 ZETAS=(T1-TIIS)/(TTPL-TIIS)
0016 PHI=FLOWT/PTPL/ATH #SORT (TTPL*CR/G)
0017 POLY=GAM/(1.+ZETAS*(GAM-1.))
0018 PHIP=ZETAS*(2./C3*(PRS*(2./POLY)-PRS*(POLY+1.)/POLY))
0019 IF(ABS(PHI-PHIP)-0.001)98,98,99
0020 R=R+0.0001
0021 GO TO 97
0022 XI=PHI/SORT(2./C3*(PRS*(2./GAM)-PRS*(GAM+1.)/GAM))
0023 WRITE(6,995)EPS
0024 FORMAT(F10.4)
0025 RAD=.10472*RPM
0026 U1=RMI*RAD/12.
0027 WU1=VU1-U1
0028 WAI=VAL
0029 BETA1=ATAN (WU1/WAI)
0030 W1=WAI/COS (BETA1)
0031 VMI=V1/SORT (GAM*G*CR*TI)
0032 WMI=W1/SORT (GAM*G*CR*TI)
0033 RETURN
0034 END

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0001 SUBROUTINE CONTIN(G,C2,AA,AX,RM1,PHD,PTIP,P1,PHUB,P18,P15,P16,P17,
0002 1FLOWT,TORQ,CLFAX,FAX,TTPL,VUI,VA1,V1,ALPH1,T1,RTIPI,RHUB1,CLTORQ,
0003 2RADCLR,PBAR,DELP1,C3,SKT,CR,P19,P20)
0004 VUI=(TORQ+CLTORQ)*G/FLOWT/RM1*12.
0005 THETA2=VUI**2./C2/TTPL
0006 SK=FLOWT/SKT/AA*SQRT(CR/G*C3/2.*TTPL)/P1
0007 T1=TTPL*(SQRT(1.+4.*SK**2*(1.-THETA2))-1.)/2./SK**2
0008 V1=SQRT((TTPL-T1)*C2)
0009 VA1=SQRT(V1**2 -VUI**2)
0010 ALPH1=ATAN(VUI/VA1)
      RETURN
      END
.997
      999

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0001 SUBROUTINE MOMENT(G,C2,AA,X,RM1,PHD,PTIP,P1,PHUB,P18,P15,P16,P17,
0002 1FLOWT,TORQ,CLEFAX,FAX,TIPL,VU1,VAL,V1,ALPHI,T1,RTIPI,RHUB1,CLTORQ,
0003 2RADCLR,PBAR,DELPI,C3,SKT,CR,EPS,P19,P20)
0004 VU1=(TORQ+CLTORQ)*G/FLOWT/RM1*12.
0005 PI=3.14159
0006 A1=PI/4.*11.0**2
0007 A2=A1-PI/4.*10.132**2
0008 A3=A1-PI/4.*9.932**2
0009 A4=A1-PI/4.*(9.836+2.*RADCLR)**2.-A2-A2A
0010 A6=PI*RHUB1**2
0011 F1=A1*PHD
0012 F2=A2*(P19+P20)/2.
0013 F3=A3*(P15+P16)/2.
0014 F4=A4*PTIP
0015 F5=PI*(1+EPS)*(PHUB*(RTIPI**2-RHUB1**2)+2*((PTIP-PHUB)/(RTIPI-
0016 RHUB1))*(RTIPI**3/3.-RHUB1*RTIPI**2/2.+RHUB1**3/6.))
0017 F6=A6*PHUB
0018 FNET=F1-F2-F3-F4-F6+CLFAX+FAX-F2A-F5
0019 VAL=G*FNET/FLOWT
0020 ALPH1=ATAN(VU1/VAL)
0021 V1=SQRT(VAL**2+VU1**2)
0022 T1=TTPL-V1**2/C2
0023 RETURN
END

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0001 SUBROUTINE ROTOR(G,CR,CP,CJ,C2,C3,RM1,RM2,AAXR,FLOWT,TTPL,RPM,
0002 1,DYNA,DP,PI,P2,T1,T2,TT2,T2IS,VAL,VU1,V2,W1,W2,U1,U2,ALPH2,BETA1,
0003 2,BETA2,C4,RKT)
0004 DP=.10472*RRPM
0005 DP=DYNA*RAD/CJ
0006 VU2=RM1/RM2*VU1-DYNA*G/RM2/FLOWT*12.
0007 TT2=TTPL-DP/CP/FLOWT
0008 T2IS=T1*(P2/PI)*C3
0009 U2=RM2*RAD/12.
0010 WU2=VU2-U2
0011 TE=T1+(W1**2.-U1**2.+U2**2.)/C2
0012 TETA2=(WU2**2)/TE/C2
0013 RK=FLOWT/RKT/AAXR*SQR(CR/G*C3/2.*TE)/P2
0014 T2=TE*(SQR(1.+4.*RK**2.*(1.-TETA2))-1.)/2./RK**2.
0015 V2=SQR(T2*(TT2-T2)*C2)
0016 VA2=SQR(V2**2-VU2**2)
0017 WA2=VA2
0018 W2=SQR(WA2**2+WU2**2)
0019 ALPH2=ATAN(VU2/VA2)
0020 BETA2=ATAN(WU2/WA2)
0021 RETURN
0022 END

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0001 SUBROUTINE PERFRM (G,CR,CP,C1,C2,C3,GAM,TTPL,PTPL,PHUB,PTIP,PHD,
1RPM,FLOWT,DYNA,DP,P2,IIS,I1,T2IS,PR,ETA,XKIS,COEFL,COEFFM,COEFP,
2COEFS,REACMN,REACHR,REACTP,ZETAR,ALPH1,ALPH2,BETA1,BETA2,DBETA,
3HP,U2,W1,W2)
  PR=PTPL/P2
  T2TH=TTPL*(PHD/PTPL)**C3
  DHIS=CP*(TTPL-T2TH)
  ETA=DP/DHIS/FLOWT
  XKIS=DHIS*C1/U2**2
  XKIS=XKIS*4.2048**2/4.125**2
  DEL=PTPL/14.69
  THETA=SQRT(GAM*CR*TTPL)/196.8107
  COEFL=FLOWT*THETA/DEL
  COEFFM=DYNA/DEL
  HP=3600./2545.*DP
  COEFP=HP/(DEL*THETA)
  COEFS=RPM/THETA
  REACMN=1.0-CP*(TTPL-IIS)/DHIS
  REACHB=((PHUB/PHD)**C3-1.)/((PTPL/PHD)**C3-1.)
  REACTP=((PTIP/PHD)**C3-1.)/((PTPL/PHD)**C3-1.)
  SQW2TH=C2*(I1-T2IS)+W1**2
  ZETAR=1.-W2**2/SQW2TH
  ALPH1=ALPH1*57.2957779
  ALPH2=ALPH2*57.2957779
  BETA1=BETA1*57.2957779
  BETA2=BETA2*57.2957779
  DBETA=BETA1-BETA2
  RETURN
0026 END

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0001 SUBROUTINE OUTPTA (NRUN,AXCLR,AXCLR,RADCLR,FLOWT,PR,HP,XKIS,COEFL,COEFLM,
1 COEFP,COEFS,V1,V2,W1,W2,U1,U2,T1,T2,T2IS,TT2,ALPH1,ALPH2,BETA1,
2 BETA2,DBETA,ETA, ZETAR,ZETAS,ZETA,REACHB,REACMN,REACTP,
3 VM1,WMI,PHI, XI,PRS,RPM,TTPL,N)
0002 DIMENSION FLOWT(50),PR(50),HP(50),XKIS(50),COEFL(50),COEFLM(50),
1 COEFP(50),COEFS(50),V1(50),V2(50),W1(50),W2(50),U1(50),U2(50),
2 T1(50),T2(50),T2IS(50),TT2(50),ALPH1(50),ALPH2(50),BETA1(50),
3 BETA2(50),DBETA(50),ETA(50),ZETA(50),ZETAS(50),ZETA(50),
4 REACHB(50),REACMN(50),REACTP(50),VM1(50),WMI(50),PHI(50),
5 XI(50),PRS(50),RPM(50),TTPL(50)
0003 WRITE(6,805)
0004 FORMAT(1H1//22H ARES TURBINE MODEL II //)
0005 WRITE(6,806)NRUN,AXCLR,RADCLR
0006 FORMAT(11H RUN NUMBER I3,8H SPACING F10.3,11H TIP CLEAR. F10.3/)
0007 WRITE(6,807)
0008 FORMAT(5X,6H POINT 6X,10H FLOW RATE 9X,4H RPM
1 5X,15H PRESSURE RATIO 5X,11H HORSEPOWER /)
0009 DO 808 I=1,N
0010 WRITE(6,809)I,FLOWT(I),RPM(I),PR(I),HP(I)
0011 FORMAT(15,12X,4(F10.4,5X))
0012 WRITE(6,810)
0013 FORMAT(//20X,37H DIMENSIONLESS PERFORMANCE PARAMETERS //
15X,6H POINT 11X,2H K 10X,9H REFERRED 8X,9H REFERRED
27X,9H REFERRED 7X,9H REFERRED 6X,
317X,11H ISENTROPIC 5X,10H FLOW RATE 8X,7H TORQUE
4 9X,6H POWER 10X,6H SPEED 9X,/)
0014 DO 811 I=1,N
0015 WRITE(6,812) I,XKIS(I),COEFL(I),COEFLM(I),COEFS(I)
0016 FORMAT(15,12X,6(F10.4,6X))
0017 WRITE(6,813)NRUN
0018 FORMAT(1H1//5X,16H GENERAL RESULTS //20X,20H VELOCITIES (FT/SEC:
1//20X,11H RUN NUMBER I5/
2//5X,6H POINT 10X,3H V1 12X,3H V2 12X,3H W1 12X,3H W2
3 12X,3H U1 12X,3H U2 /)
0019 DO 814 I=1,N
0020 WRITE(6,815) I,V1(I),V2(I),W1(I),W2(I),U1(I),U2(I)
0021 FORMAT(15,12X,6(F10.5,5X))
0022 WRITE(6,816)
0023 FORMAT( /20X,23H TEMPERATURES (DEG R ) //
1 5X,6H POINT 8X,7H PLENUM 6X,7H STATOR 8X,6H ROTOR
2 9X,11H ISENTROPIC 4X,6H TOTAL /19X,6H TOTAL 7X,
310H DISCHARGE 5X,10H DISCHARGE 5X,8H FROM T1 7X,
410H DISCHARGE /)
0024 DO 817 I=1,N
0025 WRITE(6,818) I,TTPL(I),T1(I),T2(I),T2IS(I),TT2(I)
0026 FORMAT(15,12X,5(F10.5,5X))
0027 WRITE(6,819)NRUN
0028 FORMAT(1H1//20X,33H FLOW ANGLES (DEGREES FROM AXIAL) //

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0029 120X, 11H RUN NUMBER I5///
0030 2 5X, 6H POINT 7X, 8H ALPHA 1 8X, 8H ALPHA 2 7X, 7H BETA 1
0031 3 8X, 7H BETA 2 6X, 12H DELTA BETA / )
0032 DO 820 I=1,N
0033 WRITE(6,821) I, ALPH1(I),ALPH2(I),BETA1(I),BETA2(I),DBETA(I)
      .821 FORMAT ( I5, 12X, 5(F10.5,5X) )
      822 WRITE(6,822) //20X,24H EFFICIENCIES AND LOSSES //
      FORMAT POINT 10X, 11H EFFICIENCY 17X, 5H ZETA 20X, 5H ZETA/
      12X, 6H ROTOR 18X, 6H STATOR 16X, 7H STATOR//
      DO 823 I=1,N
      823 WRITE(6,824) I, ETA(I), ZETAR(I), ZETAS(I)
      824 FORMAT (I5, 12X, 3(F10.5, 15X) )
      825 WRITE(6,825) NRUN MACH NUMBERS AND DEGREE OF REACTION //
      FORMAT (1H1, 36H MACH NUMBERS AND DEGREE OF REACTION //
      120X, 11H RUN NUMBER I5///
      25X, 6H POINT 8X, 9H REACTION 6X, 9H REACTION 7X, 9H REACTION
      37X, 9H ABSOLUTE / 23X, 4H HUB 9X, 5H MEAN 11X,
      44H TIP 12X, 7H MACH 1 9X, 7H MACH 1 // )
      DO 826 I=1,N
      826 WRITE(6,827) I, REACHB (I), REACMN(I), REACTP (I), VMI(I), WMI(I)
      827 FORMAT (I5, 12X, 5(F10.4, 6X) )
      828 WRITE(6,828) //49H STATOR PRESSURE RATIO AND THROAT BLOCKAGE FACTOR//
      FORMAT (//49H STATOR PRESSURE RATIO AND THROAT BLOCKAGE /
      12X, 6H POINT 11X, 9H PRESSURE 17X, 9H BLOCKAGE /
      221X, 6H RATIO 19X, 7H FACTOR / )
      DO 829 I=1,N
      829 WRITE(6,830) I, PRS(I), XI(I)
      830 FORMAT(I5, 12X, 2(F10.5, 15X) )
      RETURN
      END
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0001 SUBROUTINE OUTPUT(NRUN,AXCLR,RADCLR,N,PR,XKIS,EFF,COEFL,COEFM,
0002 COEFP,COEFS,REACHB,REACTP,PTPL)
0003 DIMENSION PR(50),XKIS(50),EFF(50),COEFL(50),COEFM(50),COEFP(50),
0004 COEFS(50),REACHB(50),REACTP(50),PTPL(50),NPTS(8),
0005 NPTS(8),ICOEFS(50)
0006 READ(5,905)DATE
0007 READ(5,906) REPRTA ,REPRTB ,TTYPEA,TTYPEB,CONFIG,METHOD
0008 READ(5,907)RMEAN
0009 READ(5,908)J (NPTS(K),NPTS(K),K=1,8)
0010 DO 919 I=1,N
0011 EFF(I)=100.*EFF(I)
0012 DO 950 JJ=1,2
0013 IF (N+3*J-43) 920,920,921
0014 NPP=1
0015 NXX=N
0016 GO TO 922
0017 NPP=2
0018 NXX=43
0019 NP=1
0020 NX=1
0021 K=1
0022 WRITE(6,900) REPRTA ,REPRTB ,NP,NPP,TTYPEA,TTYPEB,CONFIG,RADCLR,
0023 WRITE(6,901)RMEAN
0024 DO 940 I=NX,NXX
0025 ICOEFS(I)=COEFS(I)
0026 I=I
0027 WRITE(6,902) I,PR(I),XKIS(I),EFF(I),COEFL(I),COEFM(I),COEFP(I),
0028 ICOEFS(I),REACHB(I),REACTP(I)
0029 IF (J-K) 940,924,924
0030 IF (I-NPTS(K)) 940,925,940
0031 LL=NPTS(K)
0032 LLL=LL-LL+1
0033 INTGER =LLL-LL+1
0034 AVG=INTGER
0035 AVGATM=0.0
0036 AVGPR=0.0
0037 DO 926 L=LL,LLL
0038 AVGATM=AVGATM+PTPL(L)/14.69 /AVG
0039 AVGPR=AVGPR+PR(L)/AVG
0040 LLL=LLL-1
0041 COMPAR =PR(LL)
0042 DO 931 L=LL,LLL
0043 IF (COMPAR -PR(L+1)) 930,931,931
0044 COMPAR =PR(L+1)
0045 CONTINUE
0046 DEVHI =100.*(COMPAR -AVGPR)/AVGPR

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0044 COMPAR =PR(LL)
0045 DO 934 L=LL,LLL
0046 IF (CCMPAR-PR(L+1)) 934,934,933
0047 COMPAR =PR(L+1)
0048 CONTINUE
0049 DEVLOW=100.*(AVGPR-COMPAR )/AVGPR
0050 WRITE(6,903) NPTS(K),NPTSS(K),AVGPR,DEVHI ,DEVLOW,AVGATM
0051 IF (NPTSS(K+1)+3*(K+1)-43) 936,936,935
0052 IF (NP-2) 944,936,944
0053 K=K+1
0054 CONTINUE
0055 IF (N-II) 950,950,945
0056 NP=2
0057 NX=I+1
0058 NXX=N
0059 WRITE(6,904)NP
0060 GO TO 923
0061
0062
900 FORMAT(IH1,/ ,9X,7H REPORT,2X,A8,A2,77X,6H SHEET,1X,I1,3H OF,1X,I1
1,/,/,33X,53H TURBO PROPULSION LABORATORY USNPGS, MONTEREY, CALIF. TR
2,/,/,21X,79H REDUCED PERFORMANCE DATA OF TURBINE FROM TESTS WITH FIG
3ANSONIC TURBINE TEST RIG // ,13H TURBINE TYPE,1X,A8,A3,14H CONFIGU
4RATION,1X,A5,1X,25H RADIAL ROTOR TIP CLEAR.=,24X,13H TEST RUN NO.,1X,I3
5L CLEAR. STATOR-ROTOR=,1X,F5.3,4H IN./ ,24X,13H TEST RUN NO.,1X,I3
6,3X,13H DATE OF TEST,1X,A8,22H DATA REDUCTION METHICY,1X,A4,/,/)
901 1 REFERRED REFERRED ISENTROPIC EFFICIENCY / ,15X, 96H
2 RATIO HEAD COEFF. REFERRED DEGREE OF TORQUE / ,15X, 96H
3 SPEED REACTION / ,24X, 4H RPM
4 CENT LBM/SEC
5 //)
902 FORMAT(9X,I2,5X,F6.4,5X,F6.4,8X,F5.2,6X,F6.4,5X,F6.3,4X,F6.3,4X,
115,5X,F5.4,6X,F5.4)
903 1=,1X,F6.4,18H , MAX.DEVIATION +, F5.3,7H PCT. -, PRESSURE RATIO
2 ,PAVG./PATM0=,1X,F6.4 /)
904 1,PAVG./PATM0=,1X,F6.4 /)
905 FORMAT(A8)
906 FORMAT(A8,A2,10X,A8,A3,9X,A5,5X,A4)
907 FORMAT(F5.3)
908 FORMAT(I1)
909 FORMAT(8(2I5))
950 CONTINUE
RETURN
END

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RUN NUMBER 79

INPUT DATA, CARD IMAGE, 1 LINE = 1 CARD

PBAR=30.050

37.89999 37.59999 37.70000 37.59999 37.59999
50.00000 50.00000 50.00000 50.00000 50.00000
46.03000 45.96999 45.90999 45.87999 45.81999
50.26999 50.25999 50.25000 50.23999 50.23000
50.28999 50.28000 50.26999 50.26999 50.26999
50.25000 50.25000 50.23999 50.23999 50.23000
50.12000 50.12000 50.10999 50.10999 50.10999
50.01999 50.01999 50.01999 50.01999 50.01999
50.00000 50.00000 50.00000 50.00000 50.00000
41.00000 40.96999 40.96999 40.93999 40.90999
70.00000 70.00000 70.00000 70.00000 70.00000
48.89999 48.89999 48.89999 48.89999 48.89999
52.64999 52.64999 52.64999 52.64999 52.64999
69.29999 69.25000 69.23000 69.20999 69.18999
70.00000 70.00000 70.00000 70.00000 70.00000
2.19500 2.19500 2.19500 2.19500 2.17500
2.11500 2.11500 2.11500 2.11500 2.10500
955.00000 960.00000 975.00000 983.00000 987.00000
899.00000 890.00000 890.00000 888.00000 886.00000
589.00000 565.00000 536.00000 514.00000 488.00000
11440.00000 11980.00000 12540.00000 12980.00000 13540.00000
12.06000 12.44000 13.04000 13.31000 13.75000
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37.59999 37.59999 37.59999 37.59999 37.59999
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45.87999 45.87999 45.87999 45.87999 45.81999
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983.00000 983.00000 983.00000 983.00000 987.00000
888.00000 888.00000 888.00000 888.00000 886.00000
514.00000 514.00000 514.00000 514.00000 488.00000
12980.00000 12980.00000 12980.00000 12980.00000 13540.00000
13.31000 13.31000 13.31000 13.31000 13.75000
15.00000 15.00000 15.00000 15.00000 15.00000

37.59999 37.59999 37.59999 37.59999 37.59999
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45.79999 45.79999 45.79999 45.79999 45.79999
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70.00000 70.00000 70.00000 70.00000 70.00000
48.89999 48.89999 48.89999 48.89999 48.89999
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| | | | |
|---|---|--|--|
| 1. ORIGINATING ACTIVITY (Corporate author) Naval Postgraduate School Monterey, California 93940 | | 2a. REPORT SECURITY CLASSIFICATION UNCLASSIFIED | |
| | | 2b. GROUP | |
| 3. REPORT TITLE | | | |
| 4. DESCRIPTIVE NOTES (Type of report and inclusive dates) Thesis, M.S., September 1967 | | | |
| 5. AUTHOR(S) (Last name, first name, initial) Commons, Patrick M., LT, USN | | | |
| 6. REPORT DATE September 1967 | 7a. TOTAL NO. OF PAGES 108 | 7b. NO. OF REFS 10 | |
| 8a. CONTRACT OR GRANT NO. | 9a. ORIGINATOR'S REPORT NUMBER(S) | | |
| 8b. PROJECT NO. | | | |
| 8c. | 9b. OTHER REPORT NO(S) (Any other numbers that may be assigned this report) | | |

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|-------------------------|--|
| 11. SUPPLEMENTARY NOTES | 12. SPONSORING MILITARY ACTIVITY Commander Naval Air Systems Command Department of the Navy Washington, D. C. 20360 |
|-------------------------|--|

13. ABSTRACT

The primary purpose of the Transonic Turbine Test Rig is to study the effects of axial and radial clearances on turbine performance. Additionally, the installation was designed to permit the determination of stator discharge velocities by the application of the momentum, and moment of momentum equations.

Previous attempts to obtain stator discharge velocities by these methods have been unsuccessful. However, during this series of tests, with over 150 hours of useful operating time, the instrumentation and the data acquisition procedures were improved to the extent that the difference between stator discharge velocities obtained from a momentum analysis and those obtained from continuity were reduced to less than three percent.

| 14. KEY WORDS | LINK A | | LINK B | | LINK C | |
|-----------------------------|--------|----|--------|----|--------|----|
| | ROLE | WT | ROLE | WT | ROLE | WT |
| Turbine Instrumentation | | | | | | |
| Turbine Test Rig | | | | | | |
| Inlet Guide Vanes | | | | | | |
| Momentum Equation | | | | | | |
| Moment of Momentum Equation | | | | | | |

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