Performance and Market Evaluation of the Bladeless Turbine

**B. A. Garrett-Price
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October 1982

**Prepared for the U.S. Department of Energy
under Contract DE-AC06-76RLO 1830**

**Pacific Northwest Laboratory
Operated for the U.S. Department of Energy
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PACIFIC NORTHWEST LABORATORY
operated by
BATTELLE
for the
UNITED STATES DEPARTMENT OF ENERGY
under Contract DE-AC06-76RLO 1830

Printed in the United States of America
Available from
National Technical Information Service
United States Department of Commerce
5285 Port Royal Road
Springfield, Virginia 22151

NTIS Price Codes
Microfiche A01

Printed Copy

Pages	Price Codes
001-025	A02
026-050	A03
051-075	A04
076-100	A05
101-125	A06
126-150	A07
151-175	A08
176-200	A09
201-225	A010
226-250	A011
251-275	A012
276-300	A013

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1.0 SUMMARY

This section summarizes the results of the technical and market evaluation of the Gamell bladeless turbine.

The three-inch diameter prototype turbine was tested with air over a range of inlet pressures from 20 to 100 psia and speeds of 10, 20, 30 and 40 thousand rpm. The peak efficiency of 22.5 percent was recorded at a pressure of 98 psia and a speed of 40,000 rpm. Efficiency increased slightly with speed and inlet pressure over the range of test conditions.

The test program was somewhat hindered by mechanical failures. The turbine bearings in particular were unreliable, with two instances of outright failure and numerous cases of erratic performance. The test apparatus, methodology and results are described in detail in Section 3.0.

A model of the bladeless turbine was developed to aid in interpreting the experimental results. A macroscopic approach, incorporating several favorable assumptions, was taken to place a reasonable upper bound on turbine efficiency. The model analytically examines the flow through the air inlet nozzles and the interaction between the fluid jet and the turbine blades. The analysis indicates that the maximum possible efficiency of a tangential flow turbine with straight axial blades is 50 percent. This is a direct consequence of turning the fluid only 90 degrees relative to the turbine blade. Performance modeling and prediction are discussed in detail in Section 4.0.

The adoption of the bladeless turbine as the expander in an Organic Rankine Cycle (ORC) will depend to a great extent on the efficiency of the turbine. The market potential for ORC technology will also impact the adoption of the bladeless turbine. Other expanders have demonstrated efficiencies of 60 to 80% in ORC systems. The Gamell turbine had a peak test efficiency of 22.5% and a maximum theoretical efficiency of 50%. Costs of the turbine are highly uncertain, relying to a great extent on cost reductions achieved through quantity production and through learning. The market analysis is discussed in more detail in Section 5.0.

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2.0 INTRODUCTION

The Gamell turbine is a proprietary bladeless device that is essentially a knurled wheel inside a multi-port housing. The turbine operates on a combined "drag" and "impulse" principle. A portion of the energy of the working fluid is imparted to the rotor through the action of shear forces on the periphery of the rotor. A part of the kinetic energy of the fluid is transferred, via impulse, to the turbine rotor. That is, the Gamell turbine uses both the frictional forces (drag) of the working fluid and the velocity head (impulse) to transfer energy from the working fluid to the turbine rotor.

The prototype turbine, with an attached 1000:1, four-stage reduction gear train, was previously tested by Gardner-Denver for the Franklin Research Center (FRC). Using air as the working fluid, the output torque, speed, and volumetric air flow rate were measured over a range of inlet pressures. The peak efficiency of the turbine with gear box attached was found to be 17.5%. Since the reducing gear was not tested by itself, data from gear manufacturers were used by FRC to estimate an efficiency of 27% for the reduction gear. This led to an estimated peak efficiency for the prototype Gamell turbine of 64.5% at an inlet pressure of 50 psig.⁽¹⁾

In their analysis, FRC indicated that the turbine has potential feasible application in the following areas: low-head hydroelectric power generation, solar-powered irrigation, air-powered generator for coal mining operations, and vehicle propulsion.

Pacific Northwest Laboratory (PNL) was contracted by the Department of Energy, Office of Industrial Programs to assess the potential of the Gamell turbine to compete with other commercial or prototype turbines. Objectives of the program included obtaining performance test data for the turbine without reduction gear attached to eliminate the uncertainty involved in estimating the gear box efficiency. The program was also intended to assess the technical and market potential of the turbine for specific applications selected from a list of candidate applications provided by the turbine proponent.

The purpose of this report is to present the results of PNL's analysis of the potential of the Gamell turbine in waste energy utilization applications.

Both the technical performance of the turbine, as determined through experimental tests and the expected market performance are discussed. The market analysis is focused on two specific applications: waste energy utilization from an internal combustion engine to supercharge a 200 horsepower air compressor and waste energy recovery from the exhaust of an internal combustion engine to turbocharge the engine itself. In both cases, the Gamell turbine acts as an expander in an organic Rankine cycle.

3.0 PROTOTYPE TURBINE PERFORMANCE EVALUATION

A three-inch diameter prototype Gamell turbine was evaluated experimentally by Pacific Northwest Laboratory (PNL) using air as the working fluid. Data were recorded at rotational speeds of 10, 20, 30 and 40 thousand rpm; a rotational speed of 62,000 rpm was achieved with no load on the turbine. Inlet pressures ranged from 20 to 100 psia. The turbine exhausted to atmosphere. Rotational speeds were measured with a non-contact optical sensor. The torque produced by the turbine was measured with a cradle dynamometer bearing upon a precision load cell. A tangential flow air brake, coaxial with the turbine, absorbed the power transmitted. The object of the tests was to determine the turbine efficiency as a function of rotational speed and load. This section includes a description of the prototype Gamell turbine and the test stand used to evaluate it. The experimental method and data reduction procedures are described. Experimental results in the form of turbine efficiency versus inlet pressure at four rotational speeds are presented and discussed.

3.1 APPARATUS

The equipment used in this evaluation consisted of the prototype Gamell turbine and a test stand and air supply network designed and built specifically for this experiment. The prototype was supplied by Joseph Gamell Industries, Inc. and was modified only as required to facilitate testing. The test stand design was complicated by the exceedingly high rotational speeds and low torques characteristic of the Gamell turbine. A thorough search failed to reveal any device that could simultaneously load the turbine and quantify the torque (or power) being transmitted; thus, the torque output was measured indirectly with a cradle dynamometer. A non-contact optical sensor provided reliable readings of the rotational speed. This section includes descriptions of the prototype Gamell turbine, turbine test stand, air supply network, and all peripheral devices required to perform the evaluation.

3.1.1 Prototype Gamell Turbine

An exploded view of the aluminum prototype tested by PNL is presented in Figure 3.1. A photograph of the turbine rotor is shown in Figure 3.2.

FIGURE 3.1. Exploded View of Prototype Gamell Turbine

Significant dimensions are listed in Table 3.1. The Gamell turbine is a tangential flow turbomachine consisting of a double-ring housing and a toothed wheel. Flow enters through a radial port in the outer housing ring and is distributed throughout an annular plenum bounded by the outer and inner housing rings. O-rings hold the two housing rings together. Flow passes from the plenum through twelve equally-spaced nozzles machined into the inner housing ring. The nozzles are oriented tangent to a circle of 2.88-inch diameter, slightly less than that of the turbine wheel (2.98 inches). The 185 blades on the turbine wheel are straight and parallel to the turbine axis; thus, the fluid jets leaving the nozzles strike the blades at an incidence angle of 90 degrees. The flow is turned axially and radially outward, transferring momentum to the turbine wheel via impulse and drag as the fluid passes over the knurled blades. To reach the six exit ports on each side of the casing, the flow turns radially inward and passes through an axial gap between the turbine wheel and side plate.

FIGURE 3.2. Knurled Turbine Rotor

TABLE 3.1. Gamell Turbine Prototype Dimensions

Housing Outer Ring Outside Diameter, in.	3.750
Housing Inner Ring Outside Diameter, in.	3.250
Air Inlet Plenum Height, in.	0.105
Air Inlet Plenum Width, in.	0.382
Housing Inner Ring Inside Diameter, in.	3.000
Number of Air Inlet Nozzles	12
Inlet Nozzle Throat Diameter, in.	0.046
Nozzle Tangent Circle Diameter, in.	2.880
Turbine Wheel Major Diameter, in.	2.984
Number of Blades	185
Blade Height, in.	0.015
Tip Clearance, in.	0.008
Blade Width, in.	0.503
Housing Width, in.	0.575
Turbine Wheel Axial Clearance, in.	0.036
Number of Exit Ports Per Side	6
Exit Port Diameter, in.	0.312
Exit Port Circle Diameter, in.	2.530

The turbine was delivered to PNL with Miniature Precision Bearing Company SR6 enclosed-raceway ball bearings. Because the age and condition of these bearings were unknown, PNL purchased open-raceway bearings from the same manufacturer. The open-raceway bearings reportedly run cooler and with less drag than closed-raceway bearings. Both types of bearings were used during testing.

3.1.2 Test Stand and Air Supply Network

The Gamell turbine prototype was tested in PNL's Engineering Development Laboratory in Richland, Washington. The air supply network and test stand were designed and built specifically for this task. Clean, dry, oil-free compressed air at 100 psig was provided by the laboratory air system. The air supply network supplied both the turbine and the air brake against which it was driven. Carbon steel 1/2-inch pipe was used for most of the air system. Reinforced Tygon tubing of 1/4-inch inside diameter and 1/32-inch wall thickness was used to connect the piping to the turbine and air brake housings. Air flow to the brake was unmetered and was controlled with a brass globe valve.

Air flow to the turbine was measured with two Schutte & Koerting Safeguard high-pressure rotameters. The calibration of these rotameters performed by PNL is presented in Appendix A. Brass globe valves were used to control flow. The air temperature at the rotameters and at the turbine inlet was measured with

Type J thermocouples connected to a Fluke model 2166A Digital Thermometer. These instruments and all others used in the turbine testing were calibrated traceable to National Bureau of Standards references; copies of the appropriate calibration documents are included in Appendix B.

Two instruments were used to measure the air pressure at the rotameters and turbine inlet. During the early testing, a Validyne P305D pressure transducer was used; however, the transducer failed when water (used for leak testing) penetrated a seal enclosing electronic circuits. A digital-readout Heise pressure gage was used for subsequent tests. Excitation for the Validyne transducer was by a Power Design Inc. direct current power supply. A Keithley 191 digital multimeter displayed the transducer output.

The test stand, shown in an exploded view in Figure 3.3, was designed to provide simultaneous rotational speed and torque measurements as the turbine operated under load. Rotational speeds were measured with a Pioneer phototachometer, a non-contact optical device. The torque output was measured with a cradle dynamometer, a device widely used in turbomachinery evaluations.^(2,3) With the turbine operating under load, the housing experiences a torque equal in magnitude and opposite in direction to that transmitted by the shaft. The housing is "cradled" in bearings so that the torque is transmitted via a moment arm to a load cell or similar device.

The test stand, like the turbine, was fabricated largely of aluminum. The turbine housing was supported by two ball bearings which, in turn, were seated in 3/4-in. thick side plates. A clamp with an integral moment arm transmitted the housing torque to an Interface Minibeam MB-5-4 load cell. Excitation for the load cell was supplied by a Power Design Inc. direct current power supply. The load cell output and excitation voltage were displayed on the same Keithley digital multimeter used with the pressure transducer.

The turbine power output was transmitted to the air brake via a stainless steel shaft. The load on the turbine was controlled by varying the air flow to the brake. The original test stand design included a third shaft bearing on the outboard end of the air brake; however, vibration caused the 3-axis linear translation stage used to position the bearing to wander aimlessly, making the bearing ineffectual.

FIGURE 3.3. Exploded View of Turbine Test Stand

A 0.375-in. thick Lexan safety shield was used during testing to prevent injury in the event of catastrophic failure.

3.2 EXPERIMENTAL METHOD

The object of the test program was to evaluate turbine efficiency at a variety of loads and rotational speeds. These two parameters could be varied independently by balancing the air flows to the turbine and brake. Turbine load was indicated by the inlet pressure. Rotational speed was monitored directly by the photo-tachometer. Each data point required about 15 seconds

to record once the valves had been properly adjusted. Two individuals were present at all times during testing.

Several activities were completed before rolling the turbine. Excitation voltages for the load cell and pressure transducer were set. The atmospheric pressure was noted. The zero reading on the Heise pressure gage, which varied by about 0.7 psi, was recorded. The load cell reading with the turbine stationary (the static zero load cell reading) was recorded. The system was pressurized. A noise-reduction housing and the Lexan safety shield were set in place. The investigators donned protective goggles and headphones.

The turbine was rolled by opening the throttling valve slowly, causing a gradual acceleration to the desired speed (usually 10,000 rpm). Flow to the air brake was not initiated immediately; at this stage, the only load on the turbine was the brake windage loss. The turbine air flow was adjusted until the desired rotational speed was achieved. This was often a time consuming process for the following reasons: 1) the significant inertia of the rotating element slowed its response to air flow changes, 2) the air supply pressure varied because of compressor cycling, and 3) erratic bearing performance sometimes resulted in abrupt speed fluctuations. With the turbine rotating at the desired speed, a "no-load" (no air flow to the brake) data set was recorded. To best approximate steady-state conditions, the most rapidly varying readings were recorded first: 1) rotational speed, 2) turbine inlet pressure, 3) rotameter pressure, 4) load cell reading, 5) volumetric flow rate, 6) turbine inlet and rotameter temperatures, and 7) excitation voltages. Variations in the temperatures and excitation voltages were almost nonexistent. A complete data set required about 15 seconds to record.

Following the no-load data sets, air brake flow was initiated and balanced against the turbine flow until the desired combination of load and speed was obtained. Data were recorded as described above. The load (and thus inlet pressure) was increased progressively until no higher turbine inlet pressure could be achieved. Flow to the air brake was then shut off and the turbine speed increased 10,000 rpm. Testing proceeded by varying load first and then rotational speed, until the maximum achievable speed had been reached. At least two data sets were recorded for each condition tested.

In addition to the data sets described above, two other series were completed: operational acceptance tests and dynamic zero tests. The largely informal operational acceptance tests were conducted to demonstrate the efficacy of various components. Of particular concern were the cradling mechanism and the load cell; their task was to resolve torques on the order of 0.01 ft-lb_f. The system was tested by placing small masses on the moment arm directly above the load cell under both static and dynamic conditions (turbine stationary and rotating). The 1/4-in. steel washers used most often weighed 0.003 lb_m, the equivalent of a torque of about 0.0015 ft-lb_f transmitted through the 0.476 ft moment arm. The data (tabulated in Appendix C and discussed in more detail with the experimental results) indicated that the mechanism could indeed quantify such minute torques.

The dynamic zero tests were conducted to provide a reference load cell reading at each speed tested. Experiments under static conditions showed that the starting torque in the cradling bearings sometimes biased the load cell reading. However, the vibration present when the turbine was rotating overcame the starting torque and yielded reproducible reference readings. These reference values, or dynamic zeroes, were recorded by running the turbine at 10, 20, 30 and 40 thousand rpm with the air brake completely removed. Their importance in evaluating the turbine performance is clearly illustrated in the following section on data reduction.

3.3 DATA REDUCTION PROCEDURE

The object of the data reduction procedure was to determine the turbine efficiency given the raw data recorded during testing. Turbine efficiency as defined herein is expressed as:

$$\eta = \frac{(\tau)(\omega)}{(m)(\Delta h)_s} \quad (3.1)$$

where τ is the torque delivered to the load, ft-lb_f; ω is the rotational speed, rad/sec; m is the air mass flow rate, lb_m/sec; and $(\Delta h)_s$ is the change in enthalpy corresponding to an isentropic expansion of air across the turbine. The balance of this section consists of the methods used to calculate each of these four factors.

3.3.1 Torque Output

The torque produced by the prototype was evaluated indirectly by measuring the reaction torque on the housing. The reaction torque, τ , may be expressed as:

$$\tau = (L)(F_{\text{net}}) \quad (3.2)$$

where L is the moment arm length, 0.476 ft, and F_{net} is the net force as indicated by the load cell. F_{net} is calculated as:

$$F_{\text{net}} = \frac{1}{S} \left(\frac{LC_{\text{out}}}{LC_{\text{ex}}} - \frac{DZ_{\text{out}}}{DZ_{\text{ex}}} \right) \quad (3.3)$$

where S is the load cell sensitivity, 0.6 mV/V_{excitation}/lb_f; LC_{out} is the load cell output signal, mV; and LC_{ex} is the excitation voltage, Volts. DZ_{out} and DZ_{ex} are the output signal and excitation voltage recorded during the dynamic zero test at the appropriate rotational speed. The ratio $\left[\frac{DZ_{\text{out}}}{DZ_{\text{ex}}} \right] (S)$ represents the force on the load cell due to causes other than the turbine torque.

3.3.2 Rotational Speed

The rotational speed, ω , in rad/sec is calculated from the photo-tachometer reading, N , in rpm as:

$$\omega = N \left(\frac{\text{rev}}{\text{min}} \right) \left(\frac{2\pi \text{ rad}}{\text{rev}} \right) \left/ \left(\frac{60 \text{ sec}}{\text{min}} \right) \right. \quad (3.4)$$

3.3.3 Mass Flow Rate

The air mass flow rate is most conveniently calculated as:

$$m = Q\rho \quad (3.5)$$

where Q is the volumetric flow rate, ft³/sec, and ρ the air density, lb_m/ft³. The volumetric flow rate is based on the rotameter calibration curve (Appendix A) and a correction factor that accounts for operating at non-standard temperature and pressure:

$$Q = \left[(0.986) \text{ RR} - 0.179 \right] \sqrt{\frac{P_s}{T_s} \frac{T_r}{P_r}} \quad (3.6)$$

where RR is the rotameter scale reading; P_s is standard pressure, 14.6 psia; T_s is standard temperature, 530^oR; P_r is the rotameter inlet pressure, psia; and T_r is the rotameter inlet temperature, ^oR. The air density is calculated via the ideal gas law:

$$\rho = (P_r) (R_{air}) (T_r) \quad (3.7)$$

where R_{air} is the gas constant for air, 53.3 ft-lb_f/lb_m^oR.

3.3.4 Enthalpy Change

The enthalpy change for any working fluid operating between temperatures T_{in} and T_{out} is simply

$$h = C_p (T_{in} - T_{out}) \quad (3.8)$$

where the specific heat, C_p , ft-lb_f/lb_m^oR, is assumed constant and T_{in} and T_{out} are the inlet and outlet temperatures, ^oR. The inlet temperature is determined by system constraints; the outlet temperature, on the other hand, depends upon the thermodynamic process. Since the objective is to calculate the maximum possible power output from an expanding gas, an isentropic expansion is the logical choice. The following relations are applicable:⁽⁴⁾

$$T_{out} = T_{in} \left(\frac{P_{out}}{P_{in}} \right)^{(\gamma-1)/\gamma} \quad (3.9)$$

$$C_p = \frac{\gamma}{\gamma-1} (R_{air}) \quad (3.10)$$

where P_{in} and P_{out} are the turbine inlet and outlet pressures, psia, and $\gamma = 1.4$ is the ratio of specific heats for air. Inserting Equations 3.9 and 3.10 into Equation 3.8 yields:

$$\Delta h = \left(\frac{\gamma}{\gamma-1} \right) R_{air} T_{in} \left[1 - \frac{P_{out}}{P_{in}}^{(\gamma-1)/\gamma} \right] \quad (3.11)$$

The outlet pressure, P_{out} , is atmospheric. The inlet pressure is measured as close to the turbine inlet plenum as practical and corrected for losses between the pressure tap location and inlet plenum (typically 2 to 3 psi).

3.4 EXPERIMENTAL RESULTS AND DISCUSSION

The prototype was tested during the one-month period from April 6 to May 6, 1982. Three types of tests employing five slightly different turbine configurations were completed. Turbine efficiencies were evaluated at rotational speeds of 10, 20, 30, and 40 thousand rpm with inlet pressures ranging from 20 to 100 psia. A peak efficiency of 22.5 percent was recorded at an inlet pressure of 98 psia and 40,000 rpm. This section includes descriptions of the configurations tested and the results achieved with each. Emphasis is placed upon the final configuration tested for which the most thorough results were recorded.

3.4.1 Summary of Testing Program

A description of the five turbine configurations tested and the types of tests run with each is presented in Table 3.2. The configurations differed

TABLE 3.2. Turbine Configurations Tested

<u>Configuration</u>	<u>Description</u>	<u>Types of Tests Completed</u>		
		<u>Operational Acceptance</u>	<u>Dynamic Zero</u>	<u>Data</u>
A	Original shaft, open-raceway bearings	X	X	X
B	Identical to A, but cleaned & reassembled		X	X
C	Original shaft, closed-raceway bearings		X	X
D	Replacement shaft, closed-raceway bearings		X	
E	Replacement shaft, one open- and one closed-raceway bearing		X	X

only by the turbine shaft and bearing types used. The original turbine shaft failed and was replaced with an identical unit. Both open- and closed-raceway bearings were used in the interest of providing a thorough and unbiased evaluation.

A chronological summary of the test program appears in Table 3.3. The early tests were primarily operational acceptance tests that demonstrated the efficacy of the torque measurement apparatus. The static operational acceptance tests (tabulated in Appendix C) showed that the starting torque of the cradling bearings sometimes prevented loads from being transmitted to the load cell. Under dynamic conditions, however, sufficient vibration was present to maintain the bearings in a nearly constant state of motion; these small movements prevented any binding and allowed the full load to be transmitted. Unfortunately, the vibration also caused oscillations in the load cell output signal that degraded readability.

The configuration A data tests were of a preliminary nature. A bearing on the outboard end of the air brake was removed after sets A1 and A2 because its position could not be reliably fixed. Tests at 30,000 and 40,000 rpm were hampered by bearing problems as evidenced by loud and irregular noises. The resulting efficiencies (Appendix C) were quite low and obviously not representative.

Testing was halted for two weeks until suitable noise-reduction measures had been taken. The configuration B dynamic zero and data tests were also hampered by bearing noise at high rotational speeds. Although the performance was improved slightly, subsequent tests (configuration E) demonstrated that better efficiencies at high rotational speeds were achievable.

Configuration C and D tests were conducted with the closed-raceway bearings originally supplied with the prototype. Unfortunately, neither set produced useful data. The configuration C tests were halted when it became apparent that the air inlet hose had been improperly installed. The hose was oriented so that it exerted a slight lifting force on the load cell probe, preventing the full turbine torque from being transmitted. In subsequent tests, the hose was oriented to provide a slight pre-load on the load cell.

The configuration D tests were delayed by the fatigue failure of the turbine shaft and the failure of the pressure transducer because of an electrical

TABLE 3.3. Chronology of Prototype Testing

4-6-82. Operational acceptance tests A1 through A3 demonstrated that loads as small as 0.003 lb_f could be resolved in static configuration. Data sets A1 and A2 were completed with outboard bearing in service.

4-7-82. Static (A4, A5, A6) and dynamic (A7, A8, A9) operational acceptance tests investigated the effect of preloading the load cell and further demonstrated the test stand efficacy. Data sets A3 through A6 were completed without the outboard bearing because of positioning problems. A maximum speed of 51,000 rpm was achieved.

4-9-82. Testing was halted by the building manager because of excessive noise.

4-21-82. Testing resumed with noise-absorbing materials in place. Dynamic zero tests A1 through A8 were completed.

4-22-82. Operational acceptance tests A10 and A11 demonstrated that preloading of load cell had no effect on test stand performance. Disassembled, cleaned, lubricated, and reassembled turbine and test stand. Dynamic zero tests B1 and B4 were hindered by sporadic bearing grinding resulting in speed and torque changes.

4-26-82. Data sets B1 through B4 were completed. Maximum speed of 48,000 rpm achieved.

4-27-82. Dynamic zero tests C1 through C5 and data sets C1 and C2 completed. Flat torque curves indicated that improperly installed air hose restricted turbine housing movement. A maximum speed of 57,800 rpm was achieved.

4-28-82. Pressure transducer failed when leak-testing water penetrated a seal and was replaced with a digital-readout pressure gage.

4-29-82 Turbine shaft failed.

5-4-82. Dynamic zero tests D1 through D5 were completed with closed-raceway bearings. Bearing seized during maximum speed test at 62,000 rpm. Dynamic zero tests E1 through E4 and data set E1 were completed with one open- and one closed-raceway bearing.

5-5-82. Data sets E2 through E5 were completed.

5-6-82. Data sets E6 through E13 were completed. The closed raceway bearing operated erratically at 40,000 rpm (set E13) and eventually seized.

short. Following completion of the dynamic zero tests, a closed-raceway bearing seized while operating at 62,000 rpm. Thus, no data tests were completed with configuration D.

The final set of tests was by far the most successful. The configuration E tests were conducted over a three-day period with one open- and one closed-raceway bearing. Three series of tests spanning the full range of rotational speeds were completed before another closed raceway bearing failure forced an end to all testing. Data from the configuration E tests are plotted and discussed in the following section.

3.4.2 Configuration E Test Results

The turbine efficiencies determined in the three series of configuration E tests are plotted in Figures 3.4 through 3.7 as functions of turbine inlet pressure. Data for the four speeds tested are plotted separately for clarity. Series 3 data at 40,000 rpm were recorded and are tabulated in Appendix C; however, the bearing failure that halted testing was obviously in progress as those data were taken, reducing the efficiencies sharply.

The trends in turbine efficiency with inlet pressure and rotational speed are clear from the figures. With the exception of the Series 3 data at 10,000 rpm (to be discussed shortly), efficiency increases gradually with inlet pressure. Efficiency also increases with rotational speed, ranging from approximately 10 percent at 10,000 rpm to a peak value of 22.5 percent at 40,000 rpm. Based upon these trends, one would expect higher efficiencies at increased speed and inlet pressure. The relative flatness of the curves, however, suggests that the increase with inlet pressure would be slight. Testing at higher speeds and pressures was not possible because of air system limitations. Further comments on efficiency trends appear in Section 4 where a performance model is discussed.

Scatter in the data within individual series is minimal, particularly at higher inlet pressures where the torque readings are increased. A worst-case uncertainty analysis (Appendix D) indicates that the uncertainty in individual data points ranges up to 15.6 percent. Taking redundant data points, however, reduces the uncertainty to about 11 percent. Significant contributors to uncertainty include: 1) poor readability of the load cell output due to vibration,

FIGURE 3.4. Turbine Efficiency Versus Inlet Pressure at 10,000 RPM

FIGURE 3.5. Turbine Efficiency Versus Inlet Pressure at 20,000 RPM

FIGURE 3.6. Turbine Efficiency Versus Inlet Pressure at 30,000 RPM

FIGURE 3.7. Turbine Efficiency Versus Inlet Pressure at 40,000 RPM

2) pressure changes due to compressor cycling, and 3) limited resolution of the rotameter scale, particularly at low flow rates. The relative magnitudes of all significant contributing factors to the uncertainty in turbine efficiency decrease with increasing inlet pressure.

Scatter between series is significantly greater than that within a given series, particularly at 10,000 rpm. The three series of configuration E tests are distinct only because of temporal separation; no modification of any sort was made to the turbine or test stand between series. However, long experience with the turbine indicated that factors beyond direct control, particularly bearing performance, could affect the testing significantly.

This fact was graphically illustrated while recording configuration E dynamic zeroes at 40,000 rpm. The turbine bearings were causing large oscillations in output torque as well as a great deal of noise. The turbine became quiet and simultaneously increased in speed to 48,000 rpm; the "set" of the bearings had obviously improved significantly. To reduce the rotational speed to 40,000 rpm again required a 25 percent reduction in air flow. Although a worst case, this example shows the potential effect of changes in turbine components beyond direct control.

The discrepancy at 10,000 rpm between the series 3 data and the previous two series decreases sharply with increasing inlet pressure. The series 3 data are highly suspect because they conflict with the trends exhibited at all speeds by the balance of the data. A variable that is significant at low inlet pressures but much less so at high pressures is clearly at fault. Such a variable is the dynamic zero, since efficiency is proportional to the net transmitted force calculated as:

$$F_{\text{net}} = \frac{1}{S} \left(\frac{LC_{\text{out}}}{LC_{\text{ex}}} - \frac{DZ_{\text{out}}}{DZ_{\text{ex}}} \right) \quad (3.12)$$

At low inlet pressures the load cell output, LC_{out} , is only slightly greater than DZ_{out} , the dynamic zero output. (Excitation voltages were virtually identical.) At high pressures, however, the difference between LC_{out} and DZ_{out} is substantial, so the absolute value of DZ_{out} is much less significant. The same set of dynamic zero readings was used to reduce all

three series of configuration E tests. However, in view of the many hours of running time that transpired between the dynamic zero tests and the series 3 tests, it is not unreasonable to suspect a slight change in the dynamic zero value. Calculations based on the data tabulated in Appendix C show that only a 10 percent increase in the dynamic zero for the series 3 tests is required to provide excellent agreement between all three series at all inlet pressures. Thus, the aberrant series 3 results at 10,000 rpm are probably due to a slight error in the dynamic zero value used in data reduction.

In summary, the Gamell turbine prototype was tested with air over a range of inlet pressures from 20 to 100 psia and speeds of 10, 20, 30 and 40 thousand rpm. The peak efficiency of 22.5 percent was recorded at a pressure of 98 psia and a speed of 40,000 rpm. Efficiency increased slightly with speed and inlet pressure over the full range tested. The test program was hindered somewhat by mechanical failures. The turbine bearings in particular were quite troublesome, with two instances of outright failure and numerous cases of erratic performance. Any future testing should be undertaken only after bearings suitable for prolonged high-speed operation are procured.

4.0 PERFORMANCE MODELING AND PREDICTION

A model of the Gamell turbine was developed to assist in interpreting the experimental results. To rigorously analyze the complex phenomena occurring in the turbine was beyond the scope of this effort; a macroscopic approach incorporating several favorable assumptions was taken instead to place a reasonable upper bound on the turbine efficiency. The predicted efficiency as a function of rotational speed is compared with the experimental values described in Section 3.4.

The evaluation described herein is of limited use unless the results can be extrapolated to conditions other than those tested. Dimensional analysis is the tool most frequently used to perform such extrapolations. This section also includes a discussion of turbomachine scaling parameters, emphasizing those that apply directly to the Gamell turbine.

4.1 GAMELL TURBINE PERFORMANCE MODEL

The Gamell turbine is described by its proponents as a combination impulse-drag device. As a partial admission tangential flow machine, it is similar in some respects to the Pelton wheel, a widely used impulse turbine.⁽⁵⁾ To adequately model the impulse and drag effects occurring in the Gamell turbine is an exceedingly complex task; the model described herein circumvents that difficulty by treating only the gross effects of momentum change rather than the detailed mechanisms responsible. Two flow features are examined in the model: 1) flow through the air inlet nozzles, and 2) the interaction between the fluid jet and the turbine blade. Significant loss mechanisms are also described. Turbine efficiency as a function of rotational speed is calculated and compared with experimental results.

4.1.1 Air Inlet Nozzle Flow

The Gamell turbine prototype is equipped with twelve air inlet nozzles in the inner housing ring as shown in Figure 3.1. Each nozzle consists of a 0.046-in. diameter hole drilled obliquely through the ring. A 0.093-in. diameter "spotface" is used to provide a square inlet as shown in Figure 4.1. The nozzle exit is roughly elliptical, although machining imperfections in the prototype distort the shape somewhat.

FIGURE 4.1. Gamell Turbine Air Inlet Nozzle and Three Alternative Models

Three configurations were evaluated as models for the prototype nozzles. The three, illustrated in Figure 4.1, include: 1) a converging nozzle in which flow is isentropic, 2) a straight cylindrical nozzle with a uniform velocity profile occupying the full area, and 3) a straight cylindrical nozzle in which the sharp-edged inlet creates a "vena contracta" resulting in higher jet velocities.

The three configurations were evaluated by calculating nozzle exit velocities and comparing them to turbine blade tip velocities at the four rotational speeds tested. Inlet plenum pressures and temperatures from actual tests were used. The reduced-area cylindrical nozzle model yielded the highest velocities and was therefore the most acceptable. However, the peak velocity of 630 ft/sec predicted by this model was still far below the maximum experimental blade tip velocity (810 ft/sec at 62,000 rpm). Since the jet velocity must exceed the blade tip velocity, none of the three models was satisfactory. Therefore, in keeping with the objective of placing an upper bound on the turbine efficiency, a favorable assumption was made: the nozzle exit velocity was assumed sonic, the maximum achievable in a converging nozzle or constant-area duct.⁽⁶⁾ The effect of this assumption is illustrated in the following section, where an expression for turbine efficiency incorporating nozzle exit velocity is developed.

4.1.2 Air Jet-Turbine Blade Interaction

The air jet leaving the inlet nozzle strikes the turbine blade as illustrated in Figure 4.2(a). The jet is of density ρ , travels at velocity V_1 , and has a mass flow rate equal to 1/12 the total turbine flow rate. The turbine wheel of radius R rotates at ω radians/second, resulting in a blade tip velocity of

$$U = \omega R \quad (4.1)$$

Velocity polygons and a close-up view of the turbine blade are shown in Figure 4.2(b). The blade is aligned axially. The sides of the blade slope upward at 37 degrees relative to a line tangent to the wheel minor diameter. The knurled top of the blade is 0.01 inches wide, and the base is 0.05 inches wide. The blade height is 0.015 inches. The inlet velocity polygon consists of three vectors all parallel to the tangential axis: 1) the absolute jet velocity, $\vec{V}_1$, 2) the turbine wheel tip velocity, $\vec{U}$, and 3) the jet velocity relative to the wheel tip, $\vec{W}_1 = \vec{V}_1 - \vec{U}$.

FIGURE 4.2. Air Jet and Turbine Blade Geometry with Velocity Polygons

The exit velocity polygon also consists of the wheel tip velocity, $\vec{U}$, the jet absolute velocity, $\vec{V}_2$, and the jet relative velocity, $\vec{W}_2 = \vec{V}_2 - \vec{U}$. The magnitude and direction of $\vec{U}$ are, of course, constant. Assuming that friction between the blade face and the jet is negligible, the magnitudes of the inlet and exit relative velocities are equal:

$$|\vec{W}_2| = |\vec{W}_1| \quad (4.2)$$

However, the direction of $\vec{W}_2$ (and thus the magnitude and direction of $\vec{V}_2$) is not immediately apparent. A portion of the jet is undoubtedly deflected radially outward and enters the tip clearance area, whose height of 0.008 inches is more than half the blade height. Fluid passing over the blade tips is slowed by drag against the blades and the turbine housing and eventually turns axially, radially inward, and axially again to pass through the exit ports. Part of the jet may also be deflected axially at the outset, particularly since that is the shortest path to the exit ports.

The significance of the direction of $\vec{W}_2$ is revealed by introducing the Euler equation for turbomachines:⁽⁵⁾

$$E_{out} = \frac{\omega}{g} (V_{1\theta} R_1 - V_{2\theta} R_2) \quad (4.3)$$

where E_{out} is the energy transfer per unit mass of fluid, g is the gravitational constant, and $V_{1\theta}$ and $V_{2\theta}$ are the tangential components of the absolute inlet and exit velocities. The expression in parentheses is proportional to the change in angular momentum of the fluid, which in turn is proportional to the torque exerted on the turbine wheel. The rotational speed, ω , is proportional to the distance through which the torque is applied. For the Gamell turbine,

$$R_1 = R_2 = R \quad (4.4)$$

and

$$U = \omega R \quad (4.5)$$

thus

$$E_{out} = \frac{U}{g} (V_{1\theta} - V_{2\theta}) \quad (4.6)$$

Now use the velocity polygons to evaluate $V_{1\theta}$ and $V_{2\theta}$:

$$V_{1\theta} = U + W_{1\theta} \quad (4.7)$$

The vector $\vec{W}_1$ is parallel to the tangential axis, so

$$W_{1\theta} = |\vec{W}_1| \equiv W_1 \quad (4.8)$$

$$V_{1\theta} = U + W_1 \quad (4.9)$$

$$V_{2\theta} = U + W_{2\theta} \quad (4.10)$$

Define the angle β to be the angle through which the jet relative velocity is deflected by striking the blade. The tangential component of the exit relative velocity is then

$$W_{2\theta} = |\vec{W}_2| \cos \beta \equiv W_2 \cos \beta \quad (4.11)$$

$$V_{2\theta} = U + W_2 \cos \beta \quad (4.12)$$

Now combine Equations 4.2, 4.6, 4.9, and 4.12:

$$E_{\text{out}} = \frac{U}{g} \left[(U + W_1) - (U + W_1 \cos \beta) \right] \quad (4.13)$$

$$E_{\text{out}} = \frac{U}{g} W_1 (1 - \cos \beta) \quad (4.14)$$

Equation 4.14 clearly demonstrates the significance of the direction of the exit relative velocity vector, $\vec{W}_2$. If the jet is deflected radially outward, the angle β is 37 degrees; if the jet is deflected axially, β is 90 degrees. An increase in β corresponds to greater energy transfer from the fluid to the blade. Well-designed impulse turbines such as the Pelton wheel deflect the fluid through angles as great as $\beta = 170$ degrees.⁽⁵⁾

In the analysis, the fluid is assumed to exit the blade axially ($\beta = 90$ degrees) for two reasons. First, this highly optimistic assumption leads to the maximum possible energy transfer and thus places an upper bound on turbine efficiency. Second, a portion of the tangential momentum of the fluid passing over the blade tips is transmitted to the turbine wheel as drag. The axial flow assumption eliminates the need to model the exceedingly complex flow in the tip clearance area.

The theoretical power output from the turbine is the product of the mass flow rate and the energy transfer rate per unit mass

$$P_{out} = (\rho g Q) E_{out} \quad (4.15)$$

$$P_{out} = \rho Q U W_1 (1 - \cos \beta) \quad (4.16)$$

The energy in the air entering the turbine is present in the form of kinetic energy and static pressure in excess of atmospheric

$$E_{in} = \frac{V_1^2}{2g} + \frac{P_1}{\rho} \quad (4.17)$$

where P_1 is the jet gage pressure at the nozzle exit. The static pressure, however, exerts no net force on the turbine wheel because it exerts equal force in all directions. To place an upper bound on the turbine efficiency, the jet static pressure is assumed atmospheric, implying that a well-designed nozzle converted all the energy of the incoming air into kinetic energy. The power input to the turbine is

$$P_{in} = (\dot{m})(E_{in}) \quad (4.18)$$

$$P_{in} = \rho Q \frac{V_1^2}{2} \quad (4.19)$$

Neglecting all losses, the turbine efficiency is

$$\eta = \frac{P_{out}}{P_{in}} \quad (4.20)$$

$$\eta = \frac{\rho Q U W_1 (1 - \cos \beta)}{\rho Q V_1^2 / 2} \quad (4.21)$$

Now insert $W_1 = V_1 - U$:

$$\eta = 2 \frac{U}{V_1} \left(1 - \frac{U}{V_1} \right) (1 - \cos \beta) \quad (4.22)$$

Differentiation of Equation 4.22 with respect to the velocity ratio U/V_1 shows that the efficiency is maximized when

$$U/V_1 = 0.5 \quad (4.23)$$

which states that the jet velocity should be twice the turbine blade tip velocity. Inserting this value and $\beta = 90$ degrees into Equation 4.22 yields

$$\eta_{\max} = (2)(.5)(1-.5)(1 - \cos 90^0) \quad (4.24)$$

$$\eta_{\max} = 0.5 \quad (4.25)$$

Thus, according to this analysis, the maximum possible efficiency of a tangential flow turbine with straight axial blades is 50 percent, a direct consequence of turning the fluid only 90 degrees relative to the turbine blade. The discussion of loss mechanisms presented in the following section indicates that significantly lower efficiencies may be expected for the Gamell turbine. Following that discussion, the model predictions are compared with the experimental results described in Section 3.4.

4.1.3 Turbine Loss Mechanisms

A number of loss mechanisms have been identified and well documented in the turbo-machinery literature. Two mechanisms were mentioned in previous sections of this report: bearing losses and losses due to non-optimal velocity ratios. Priest and Roelke, in a text edited by Glassman,⁽²⁾ identify several other loss mechanisms including the following:

- boundary-layer losses due to friction between the working fluid and machine surfaces;
- tip-clearance losses caused by the passage of fluid through the radial clearance gap;
- disk-friction losses due to the circulation of fluid between the rotating disk and stationary housing; and
- partial admission losses due to pumping of air in the inactive regions and to the filling and emptying of blades around the nozzles.

Of these mechanisms, only the velocity-ratio and tip-clearance losses will be treated quantitatively. The bearing losses observed experimentally were highly intermittent and thus difficult to evaluate. The boundary-layer and tip-clearance regions in the Gamell turbine are very nearly coincidental; to account

for both mechanisms might well be double jeopardy. Losses due to disk friction were evaluated for a sample case and were found to be slight. Finally, partial-admission losses were judged likely to be negligible because the Gamell turbine, with twelve inlet nozzles, is nearly a full-admission device.

Turbine efficiency as a function of velocity ratio, calculated via Equation 4.22, is plotted in Figure 4.3 for the case of $\beta = 90$ degrees. The curve is symmetric about a peak value of 50 percent at $U/V_1 = 0.5$. The efficiency is zero at $U/V_1 = 0$ because the turbine wheel is not rotating and therefore delivers no power to the load. At a velocity ratio $U/V_1 = 1.0$, the jet and blade-tip velocities are equal; the jet is not deflected by the blade, and no momentum exchange takes place.

The effect of blade-tip clearance on turbine efficiency is illustrated in Figure 4.4. The solid line and triangular data points were abstracted from Glassman;⁽²⁾ the dotted line is a linear extrapolation performed for the current work. The ordinate of Figure 4.4 is the fraction of the turbine efficiency that would exist if no tip-clearance losses were present. The abscissa

FIGURE 4.3. Turbine Efficiency Versus Velocity Ratio

FIGURE 4.4. Effect of Tip Clearance on Turbine Efficiency

is the tip clearance expressed as a fraction of total passage height. For the Gamell turbine, the clearance fraction CF is

$$CF = (.008)/(.008 + .015) = .348 \quad (4.26)$$

where the tip clearance is .008 inches and the blade height .015 inches.

The tip-clearance efficiency, η_{tc} , corresponding to this clearance fraction is 42 percent. When multiplied by the peak efficiency of 50 percent determined in the previous section (the maximum efficiency attainable with no loss mechanisms whatsoever), the resulting turbine efficiency is 21 percent.

The extrapolation of Figure 4.4 must be used with caution, however. The range of the extrapolation is quite broad; furthermore, the closely-spaced knurled blade tips of the Gamell turbine are quite possibly more adept at extracting momentum from fluid passing over them than conventional impulse-turbine blades. For lack of a more directly applicable model, however, a tip-clearance efficiency of 42 percent will be used in the following section in which the model predictions are compared with experimental results.

4.1.4 Comparison Between Model and Experiment

The model developed in the previous two sections is capable of predicting turbine efficiency as a function of velocity ratio while accounting for tip-clearance losses. The resulting expression for turbine efficiency is

$$\eta = (\eta_{\text{tip clearance}}) (2) \left(\frac{U}{V_1} \right) \left(1 - \frac{U}{V_1} \right) (1 - \cos \beta) \quad (4.27)$$

where the tip clearance efficiency extrapolated via Figure 4.4 is 42 percent; the jet turning angle, β , is 90 degrees; the blade tip velocity, U , is calculated via Equation 4.5; and the jet velocity, V_1 , is assumed sonic and is calculated as

$$V_1 = c = \sqrt{\gamma R_{\text{air}} T_1} \quad (4.28)$$

$$T_1 = T_0 \left(\frac{P_1}{P_0} \right)^{(\gamma-1)/\gamma}$$

Where T_1 is the nozzle exit temperature, $^{\circ}\text{R}$; T_0 is the inlet plenum temperature, typically 533°R ; and P_1/P_0 is the nozzle pressure ratio, 0.528 for choked flow. The resulting air jet velocity is $V_1 = 1030 \text{ ft/sec}$.

The efficiencies predicted by Equation 4.27 are plotted in Figure 4.5. Both rotational speed and blade-tip velocity are indicated on the abscissa.

Also plotted in Figure 4.5 are the peak efficiencies determined experimentally at the four rotational speeds tested. In general, the experimental efficiencies exceed the model values slightly. Bear in mind, however, that peak values were used, and that individual data points are subject to an uncertainty of roughly 15 percent (Appendix D). Recall also the broad extrapolation required to determine the tip-clearance efficiency. In view of these qualifications, the level of agreement between experiment and model is quite satisfactory. Note that the model predicts a decrease in turbine efficiency beyond 40,000 rpm in spite of the assumption of sonic jet velocity. To achieve increased efficiency at higher rotational speeds would require the use of a converging-diverging nozzle.

FIGURE 4.5. Turbine Model and Peak Experimental Efficiencies

4.2 TURBOMACHINERY SCALING RELATIONSHIPS

Only in rare instances is a prototype turbomachine tested at the same conditions under which the finished product is expected to operate. In all other cases, testing is performed over a limited range of presumably benign conditions; dimensionless scaling groups are then used to predict performance under anticipated operating conditions. The derivation and applicability of these groups are well documented.^(2,5,7,8) This section includes descriptions of several frequently-cited dimensionless groups, two of which are chosen for use with the Gamell turbine analysis.

Perhaps the most commonly used turbo-machine scaling group is specific speed, N_s . For compressible-flow turbines, specific speed is often expressed as:⁽⁵⁾

$$N_s = \frac{N \sqrt{P}}{\sqrt{\rho} (gH)^{1.25}} \quad (4.29)$$

This form is dimensionless with N in rad/sec, P in ft-lb_f/sec, ρ in slugs/ft³, and gH in ft²/sec². For pumps, specific speed is frequently written as

$$N_s = \frac{N\sqrt{Q}}{(gH)^{.75}} \quad (4.30)$$

where Q is the volumetric flow rate. Equations 4.29 and 4.30 are related by the expression

$$P = \rho Q(gH) \quad (4.31)$$

which is strictly true only at 100 percent efficiency. Specific speed is properly evaluated at conditions corresponding to peak efficiency. Each class of turbomachines has its peak efficiency over a narrow range of specific speed. Impulse turbines, for instance, perform best at low specific speeds (high head and low flow rate).

A group that excludes the rotational speed is the specific diameter, D_s , expressed as⁽²⁾

$$D_s = \frac{D (gH)^{.25}}{\sqrt{Q}} \quad (4.32)$$

where D is the wheel diameter, ft. The volumetric flow rate in compressible flow machines is evaluated at exit conditions.

A third dimensionless quantity is the specific power, P_s ,⁽⁸⁾

$$P_s = \frac{P}{\rho N^3 D^5} \quad (4.33)$$

which relates power, density, rotational speed, and wheel diameter.

The ratio of the fluid head to the kinetic energy of the fluid running at the rotor tangential speed is expressed as specific head.

$$H_s = gH/(N^2 D^2) \quad (4.34)$$

The flow coefficient or specific capacity is the volume flowrate (always measured at the turbine exhaust) through a machine of unit diameter operating at unit speed. Thus,

$$Q_s = Q/ND^3 \quad (4.35)$$

Two groups of limited applicability are the Reynolds number, Re , and Mach number, M . Reynolds number is important when pumping high viscosity fluids and may be expressed as⁽⁸⁾

$$Re = \rho ND^2/\mu \quad (4.36)$$

where μ is the fluid viscosity. Nearly all turbines operate with low-viscosity fluids and have Reynolds numbers well into the turbulent regime; in such cases, Reynolds number scaling is of secondary importance.

The Mach number form most frequently employed is the blade Mach number, expressed as⁽⁸⁾

$$M = ND/c \quad (4.37)$$

where c is the local speed of sound and the product (ND) is proportional to blade speed. The Mach number is significant only for machines that operate at near-sonic or supersonic velocity.

The use of the dimensionless groups for scaling is strictly correct only if the efficiencies at the two conditions are equal. However, it is well known that larger diameter turbines are more efficient than smaller; Olson⁽⁵⁾ recommends the use of the following expression to circumvent this problem:

$$\frac{1-\eta_{small}}{1-\eta_{large}} = \left(\frac{D_{large}}{D_{small}} \right)^{0.2} \quad (4.38)$$

In the Gamell turbine evaluation, six physical quantities are of consequence: 1) power, P ; 2) head, (gH) ; 3) flow rate, Q ; 4) rotational speed, N ; 5) diameter, D ; and 6) density, ρ . To express these quantities requires the use of four fundamental units: mass, length, time, and temperature. The number of parameters required to perform dimensional analysis is⁽⁷⁾

$$\begin{aligned} (\text{number parameters}) &= (\text{number of physical quantities}) - \\ &(\text{number of units required}) \end{aligned} \quad (4.39)$$

Thus, two dimensionless groups are required to correlate the experimental results. The groups specific speed and specific head were selected for deriving a generalized scaling relationship. By squaring the specific speed equation and ratioing the following is yielded:

$$P_2 = \left(\frac{N_1}{N_2} \right)^2 \left(\frac{\rho_2}{\rho_1} \right) \left(\frac{H_2}{H_1} \right)^{2.5} P_1 \quad (4.40)$$

where the subscript one refers to the base conditions and two to the new conditions. Similarly, the specific head equation is ratioed and rearranged to yield

$$\left(\frac{N_1}{N_2} \right)^2 = \left(\frac{D_2}{D_1} \right)^2 \left(\frac{H_1}{H_2} \right) \quad (4.41)$$

This is then substituted back into equation 4.40 to yield the final scaling relationship:

$$P_2 = P_1 \left(\frac{D_2}{D_1} \right)^2 \left(\frac{\rho_2}{\rho_1} \right) \left(\frac{H_2}{H_1} \right)^{1.5} \frac{\eta_2}{\eta_1} \quad (4.42)$$

where the efficiencies are scaled based on diameter according to Equation 4.38.

Using this scaling relationship, the peak test conditions were scaled to diameters up to 12 inches, keeping all other parameters constant. At the test conditions the pressure reduction^(a) across the turbine was approximately 85%. These scaling results are shown as the curve in Figure 4.6.

Again using the scaling relationships, Figure 4.7 was derived to show the effect of changing working fluids. The output power increases with the increasing molecular weight of the fluid. Here, with a working fluid inlet temperature of 600°F and inlet pressure of 820 psia, Fluorinol-85 shows about a 25% increase in power output above that achievable with air.

$$(a) \text{ pressure reduction} = \frac{\text{Inlet Pressure} - \text{Outlet Pressure}}{\text{Inlet Pressure}}$$

FIGURE 4.6. Turbine Performance Versus Diameter

FIGURE 4.7. Output Power with Organic Working Fluid

5.0 COST ANALYSIS AND MARKET EVALUATION

This section analyzes the manufacturing cost of the bladeless turbine and presents an analysis of the potential of the turbine to compete with other commercial or prototype turbines in similar applications. Candidate applications suggested by JGI include the use of the bladeless turbine in a Rankine cycle to generate shaft power to operate a compressor, pump or generator or as a direct expansion device for hot waste gases. The Rankine cycle applications are targeted for waste heat recovery or for solar thermal applications. Suggested sources of waste heat include engine exhausts from trucks, buses and marine vehicles as well as stationary power plants, and industrial reject heat.

The analysis of the cost of the turbine at various diameters is based on a manufacturing scenario and cost estimate for the three-inch diameter turbine that was provided by JGI. The analysis of the market potential of the turbine is focused on its potential use as an expander in an organic Rankine cycle (ORC). An overview of the current status of ORC development is presented along with an assessment of the potential market for ORC systems in general. The bladeless turbine is compared to other expanders that have been developed for ORC application.

5.1 TURBINE COST ESTIMATE

Cost estimates provided by JGI for the three-inch prototype turbine and the expected manufacturing scenario were used to extrapolate costs to other turbine diameters. These costs were used as a favorable, "best case" assumption. JGI estimates a cost of about \$300 for the three-inch prototype. JGI estimates that the cost of the three-inch turbine, manufactured in quantities of 1000 units/year, is about \$140. At a production rate of 5000 units/year the estimated cost is expected to be about \$70 per turbine. These costs are somewhat lower than the actual selling price would likely be since this cost does not include selling and distribution costs and profit.

These costs are based on fabricating all turbine components, with the exception of the high speed bearings and the O-rings. The O-rings and bearings

are purchased off-the-shelf items and are installed during the fabrication process.

Turbine components will be fabricated from solid stock stainless steel or aluminum depending on compatibility of the material with the working fluid and the cycle temperature. In most cases, stainless steel will be required because aluminum is not typically compatible with the organic working fluids considered for ORC systems.⁽⁹⁾

The total machining time and the cost of each component, as estimated by JGI, are listed in Table 5.1 for a production volume of 1000 units per year. The cost includes direct labor charges, direct materials, and amortized equipment costs. Fabrication would be performed by a skilled worker. About 35% of the turbine cost is labor-related and 65% is materials-related.

The cost of the turbine was scaled to other diameters using the following relationship:

$$\text{Cost}_B = \text{Cost}_A \left(\frac{P_B}{P_A} \right)^X$$

where P is the turbine shaft power.

The exponent X was assumed to be 1.5. Figure 5.1 shows the costs extrapolated to other diameters for both levels of production.

TABLE 5.1. Turbine Machine Time and Cost

<u>Component</u>	<u>Machine Time (minutes)</u>	<u>Cost (\$)</u>
Rotor	30	17.50
Face Plate	60	19.50
Manifold	30	18.60
Distribution Chamber	<u>60</u>	<u>21.30</u>
	180	77.00
High Speed Bearings		60.00
O-Rings		<u>3.00</u>
	TOTAL	\$140.00

FIGURE 5.1. Turbine Cost as a Function of Rotor Diameter

The open-raceway bearings purchased for testing cost about \$50 each. Since these bearings did not demonstrate the reliability needed for long-term, high-speed operation, it is likely that more expensive bearings should be used. In view of the bearing costs, the costs shown in Figure 5.1 are considered optimistic.

5.2 ORGANIC RANKINE CYCLE TECHNOLOGY

This subsection provides a brief overview of ORC system technology. An illustration of an organic Rankine cycle is shown in Figure 5.2. The system consists of the following components:

- a waste heat boiler, which transfers heat from the waste heat source to the working fluid;
- an expander, which converts thermal energy to shaft power;
- a condenser that consists of a water cooled heat exchanger;
- a feed pump; and
- a regenerator.

Basically, the organic Rankine cycle converts heat energy to mechanical energy by evaporating a pressurized working fluid and then expanding the vapor through an expansion device to produce shaft power. Following expansion, the working fluid is condensed and recycled to the boiler.

FIGURE 5.2. Rankine Cycle Flowchart

The temperature-enthalpy diagram corresponding to the ORC schematic is shown in Figure 5.3. The working fluid exits the condenser as a liquid at state 1 and is pressurized by the feed pump to state 2. The liquid is preheated to state 3 in the regenerator by heat exchange with the expander exhaust. The fluid is vaporized and superheated to state 4 in the waste heat boiler. Then, the pressurized vapor is expanded to state 5 producing mechanical energy, (state 5_s represents an ideal, isentropic expansion). The fluid exhausted from the expander is precooled to state 6 in the regenerator prior to entering the condenser.

Over the range of temperatures at which application of ORC systems is considered practical (about 150 to 800°F), ideal ORC efficiencies range from about 12 to 38%.⁽¹⁰⁾ Ideal ORC efficiency is shown as a function of peak cycle temperature in Figure 5.4. For comparison, Carnot efficiency is also plotted versus temperature in the figure. Actual Rankine cycle efficiencies are lower than the ideal efficiencies due to the irreversibility of actual thermodynamic processes and to both mechanical losses and parasitic power requirements. ORC efficiencies of about 10 to 20% have been attained in industrial waste heat recovery applications.⁽⁹⁾

The efficiency of the expander, to a great extent, determines the actual thermal efficiency of the system. Results of demonstrations by a number of

FIGURE 5.3. Rankine Cycle Temperature-Enthalpy Diagram

FIGURE 5.4. Theoretical Rankine Cycle Efficiency as a Function of Temperature

firms indicate that the following component efficiencies are achievable with current technologies:⁽¹¹⁾

- Expander efficiency, 70 to 85%;
- Feed pump efficiency, 70 to 93%; and
- Waste heat boiler, regenerator, condenser, 90%.

The high end of the range for the expander and feed pump correspond to larger systems where phenomena such as turbine clearance losses do not dominate system performance.

Most of the components of an ORC are standard commercial items with the exception of the expander. A number of types of expanders have been proposed for use in ORC systems. These include reciprocating engines, turbines and positive-displacement, rotary devices. Turbines are generally considered most appropriate for large ORC systems, whereas positive-displacement devices have many advantages for smaller units.⁽¹¹⁾

The largest components are the heat exchangers that serve as the waste heat boiler, regenerator and condenser. The heat exchangers tend to dominate system costs, accounting for 30 to 40% of the system costs.^(9,12) The expander accounts for about 20% of the system capital cost.

5.3 ORGANIC RANKINE CYCLE MARKET

This section identifies the major firms involved in ORC development and manufacture, summarizes their efforts in this field, and assesses the market potential for ORC systems.

A number of firms have developed organic Rankine cycle engines for a variety of applications. These applications include bottoming cycles for engines, industrial waste heat recovery, and solar and geothermal applications. Much of the development and demonstration effort in the United States has been funded by the Department of Energy as a part of solar, geothermal or conservation programs.⁽¹¹⁾

A list of some of the most active domestic firms, along with some key parameters are shown in Table 5.2. As shown in the table, the size of the prototype ORC systems have varied from less than 3 kW to as large as 2.5 Megawatts (3 to 3550 hp). In general, both the expander efficiency and the overall ORC efficiency increase with size. However, it is also evident that high expander efficiencies are achievable at the smaller sizes. Most notable is the G.E. rotary vane expander with an efficiency of 84 to 85% at three horsepower.

A number of foreign firms have been active in ORC development and manufacture. Japanese government and industry, driven by dependence on energy imports, have been involved in ORC research and testing for more than ten years. Their focus has been on ORC development for waste heat recovery, solar and geothermal applications.⁽¹³⁾ Sofretes, a firm in France, is now producing a limited number of ORC units for solar water pumping.⁽¹¹⁾ Ormat, in Israel, has been manufacturing smaller ORC engines (1-kW) for more than 10 years.⁽¹¹⁾ In both of these latter two cases, the ORC systems are marketed primarily for remote applications and for developing countries.

TABLE 5.2. Organic Rankine Cycle Demonstration Projects

Developer	Power kW	Expander Type	Expander Efficiency, %	ORC Efficiency %	Application
AiResearch	10	radial inflow turbine	80	10-25	solar & waste heat
Barber-Nichols	1) 3-5 2) >75	radial inflow turbine	1) 60-65 2) 80	6-10	solar A/C
General Elec.	1) 2,3 2) 7 3) 30	rotary vane expander	84-85	1) 8-10 2) 14-15 3) 16.5	1) Solar A/C 2) Solar A/C 3) diesel truck
Mechanical Technology, Inc.	1000-2500	radial inflow turbine	>70	12-15	waste heat and OTEC
Sundstrand	750	axial flow turbine	75-76	20	waste heat, indus- trial & diesel
Thermo- Electron	1) 35 2) 75 3) 450	axial flow turbine	1) 72 2) 76 3) 81	1) 18 2) 19-20 3) 20	1) diesel truck 2) diesel truck 3) diesel generator

None of the domestic or foreign ORC systems can be considered to be fully commercialized. Cost reductions due to learning and to economies of scale that are associated with large volume production have not been realized due to low demand for the product. Figure 5.5 presents current and projected costs of ORC systems at 1982 price levels. Costs were obtained from the domestic developers of ORC systems at current production levels and for 10,000 units per year.

Current costs reflect the fact that a significant portion of the cost is for development engineering rather than fabrication. The projected costs, at a production rate of 10,000 units per year, are lower for a number of reasons. First, moving from producing one-of-a-kind to standardized components will lower costs by amortizing the engineering and development costs over a larger number of units. In the same manner, fabrication costs will be reduced as equipment and overhead costs are spread over a larger number of units. Lastly, costs will be reduced as labor hours decrease along a learning curve.

In a study for the Department of Energy, Resource Planning Associates (RPA) identified a total technical market potential for ORC application in the

FIGURE 5.5. ORC System Costs

industrial sector of 3220.5 MW (129,946 BPDE^(a)).⁽¹⁵⁾ Using industry-specific values for expected ORC utilization, cost of electricity, ORC installed cost, and required rate of return on investment, RPA estimated the total economic market potential at 64.4% of the technical potential or 2,072.4 MW (83,621 BPDE).⁽¹⁵⁾

Although the technical and market potential of ORC systems appear to be very promising, a number of barriers to market penetration are evident. These barriers are summarized in Table 5.3. Industry is reluctant to invest in equipment for which reliability and operating cost have not been fully established. Some industries are skeptical about dealing with high-speed, precision equipment and toxic or flammable working fluids. Industries that already have operating experience with this type of equipment and chemicals do not consider this a barrier.

Some industries are already adopting waste heat recovery technologies that utilize waste heat in the same range at which ORC systems could be applied. For example, feedstock preheating in the chemical industry uses waste heat at temperatures of 150 to 300°F.⁽¹⁴⁾ In the primary metals industries, ORC systems would have to compete with steam generation, and preheating of combustion air and charge materials. For ORC adoption, the industry must have a need

(a) Barrels per day equivalent

TABLE 5.3. Barriers to ORC Market Penetration^(14,15)

- Lack of operating history
 - system reliability
 - operating costs
 - toxic and flammable working fluids
- Competing technologies in certain temperature ranges
- Special material requirements
- Interest in reducing energy input rather than recovering waste energy
- New systems favored over retrofits because of lower costs
- Limited space available for retrofits

for on-site power generation that does not compete with alternative uses for the waste heat.

At higher temperatures, and due to constituents of some waste heat streams, special materials will be required for the waste heat boiler. Because this is one of the major cost components of an ORC, a material change could significantly impact the total cost.

Competing with all waste heat recovery technologies is the growing interest in reducing energy usage. If more efficient process equipment is available or more efficient operating techniques, industry may prefer to implement these to reduce the amount of waste heat rather than try to recapture it.

Another potential barrier is the fact that new systems are less expensive than retrofitted systems. The market potential for new systems, however, is significantly smaller than that for retrofits since investment in new plants and equipment represents only a very small fraction of the existing stock. In some cases, retrofitting may be prohibited not only by the cost, but also by the space available. This is thought to be particularly true for the older, congested plants in the primary metals industry.⁽¹⁵⁾

5.4 MARKET POTENTIAL OF BLADELESS TURBINE

This section discusses the expected market potential of the bladeless turbine in organic Rankine cycle applications and analyzes the potential of the specific applications proposed by JGI. These applications are heat recovery from the exhaust of a gasoline internal combustion engine to supercharge a rotary screw air compressor and heat recovery from the exhaust of an internal combustion engine to turbo charge the engine itself.

In order to evaluate the potential of the bladeless turbine in an ORC system, it was first necessary to estimate the overall cost of the ORC. The turbine cost was assumed to be about 20% of the capital cost of the ORC.⁽¹²⁾ This is a highly favorable assumption in view of two considerations. First, because of the simplicity of the turbine it is less expensive than other turbines and would therefore likely represent a smaller percentage of the total ORC cost than other systems. Secondly, due to the low efficiency of the expander, the associated heat exchangers and pumps would have to be larger than

those systems with more efficient expanders in order to produce a given output. Thus, the 20% assumption is used as a "best case" scenario. Other assumptions used to estimate the installed cost of an ORC system using a Gamell expander include the following:

- marketing, administration and engineering 6% of total capital⁽⁹⁾
- profit 6% of total capital⁽⁹⁾
- installation 150% of total capital⁽¹²⁾

The installed ORC system cost was divided by the shaft power produced by the turbine to yield \$/kW figures for diameters up to 12-inches. The turbine shaft power at each diameter for a turbine with a Fluorinol-85 working fluid was presented earlier in Figure 4.8. At an annual production rate of 1000 units per year, the unit costs averaged about \$110/kW at a diameter of five inches. At a production rate of 5000 units/year, this average unit cost was \$60/kW. The current costs of ORC systems utilizing the 5-inch bladeless turbine would be about \$250/kW. Comparing these figures with those presented earlier in Figure 5.5 indicates that the Gamell turbine ORC may have a cost advantage over some of the other ORC systems at production volumes of 5000 units per year. Caution must be used, however, in comparing these figures since they have not all been derived on the same basis. The current costs presented in Figure 5.5 are probably the most accurate since they are based on actual experience. The projected costs in Figure 5.5 and those for the Gamell turbine are highly uncertain and depend to a large extent on an expanding market.

Of the barriers to market penetration presented earlier in Table 5.3, two will likely impact the Gamell system more than other prototype systems. The first of these is its lack of operating history. Because of the high rotational speeds experienced by the turbine, the reliability question becomes an important issue. Testing at PNL indicated that the bearings currently proposed for the prototype are inadequate for high speed operation. Also the shaft failure experienced after 50 hours of testing may be indicative of problems that could be encountered during actual operations. The other impediment that may affect acceptance of the Gamell turbine is the limited space available for retrofits. At a given capacity, the Gamell system would have to be larger than other proposed systems due to its lower efficiency.

The specific applications for the bladeless turbine, proposed by JGI are shown in Figures 5.6 and 5.7. Figure 5.6 shows an organic Rankine cycle recovering heat from the exhaust of an internal combustion engine. The bladeless turbine provides shaft power to operate a compressor which turbocharges the engine. The expected load factor (percent of year ORC is operating) is about 40 percent.

Figure 5.7 also shows the organic Rankine cycle recovering heat from the exhaust of an internal combustion engine. This application is intended for on-site power generation where the power is used to operate compressors, such as in mining operations. In this case, the bladeless turbine generates shaft power to operate a compressor which "pre-compresses" air delivered to a rotary screw air compressor. The expected load factor for this ORC is about 25%.

The operating temperatures, pressures and flow rates are the same in both cases. The working fluid is Fluorinol-85 and the peak cycle temperature is 600°F at a pressure of 820 psia. The diameter of the turbine in this application is 4.88 inches.

FIGURE 5.6. Rankine Cycle Turbocharging System

FIGURE 5.7. Heat Recovery to Supercharge Air Compressor

Using the efficiency scaling relationship presented in Section 4.0, the peak turbine efficiency for this application was calculated to be about 30%. The required effectiveness of the vapor generator must be greater than 93% in order for the 30 hp to be produced. Once the parasitic pumping power is accounted for, the net output of the cycle is about 27 hp. If a vapor generator effectiveness of one, corresponding to an infinite heat exchange area, is assumed, the turbine shaft power generated is 32 hp, with the net cycle horsepower slightly less than 30. In both cases, the relatively large heat transfer areas required would result in relatively high costs for the system. The size of the vapor generator may preclude the use of this system in mobile applications.

The overall cycle efficiency with a heat exchanger effectiveness of one is about 19%. As the effectiveness of the vapor generator is dropped to more realistic values, the overall cycle efficiency will drop as well. At a net cycle output of 27 hp (effectiveness equal to 93.8%) the overall cycle efficiency is dropped to 17.6%.

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APPENDIX A
ROTAMETER CALIBRATION

APPENDIX A

ROTAMETER CALIBRATION

Results of the calibration of two Schutte-Koerting model 61-06-692-M rotameters are presented herein. The rotameters are identified by U.S. Government property numbers L544217 and L544218. The rated capacity of each rotameter is 1.2 to 11.6 standard cubic feet of air per minute (SCFM).

TEST FACILITY AND OPERATING PROCEDURE

The calibration was performed in the facility illustrated schematically in Figure 1. Water from a process water line was filtered and its temperature measured using a Type J thermocouple and a Fluke digital temperature readout. Flow rates were controlled by means of a throttling valve downstream of the rotameter. The discharge water was collected in a tank mounted on a 500 pound Fairbanks-Morse scale over a period of time measured by a Durgin-Browne electronic timer. The volumetric flow rate, gallons per minute (GPM), was calculated by dividing the weight of the water collected by the product of the elapsed time and fluid density. Conversion from GPM to SCFM was done in accordance with Instrument Society of America practice: GPM multiplied by 4.15 yields SCFM (1).

The thermocouple, temperature readout, timer, and scale are calibrated traceable to NBS standards by the Hanford Engineering Development Laboratory. Copies of the applicable documentation are included in Addenda 1-5.

RESULTS

Calibration results are presented in Figures 2 and 3 in terms of flow rate, SCFM, versus rotameter scale reading, SCFM. Data from the two calibrations are tabulated in Addendum 5. Results of linear regression curve fits of the data sets are also plotted in Figures 2 and 3. The correlation coefficients of the regression analyses for Figures 2 and 3 are 0.99929 and 0.99926 respectively, which indicate a high degree of linearity.

FIGURE 2 CALIBRATION CURVE OF SCHUTTE-KOERTING ROTAMETER L544218

FIGURE 3 CALIBRATION CURVE OF SCHUTTE-KOERTING ROTAMETER L544217

DATA UNCERTAINTY

An uncertainty analysis using the error and uncertainty methods of Schenck (2) was performed and is illustrated in Addendum F. Uncertainty values were calculated for the average results of data points A-20 through A-24 and B-20 through B-27. The uncertainty in the volumetric flow of each rotameter is 1.01%. Furthermore, based on data points A5, A20 through A24 and B5, B20 through B27, the volumetric flow rates of L544218 and L544217 are repeatable to within $\pm 1.04\%$ and $\pm 1.70\%$ respectively. Uncertainty and repeatability summaries are presented in Tables 1 and 2.

d

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TABLE 1. Summary of Uncertainty per Averaged Data A20 through A24 and B20 through B27

<u>Parameter</u>	<u>Value</u>	<u>Uncertainty Interval</u>	<u>Percent</u>
Temperature	47 ⁰ F	+ 1 ⁰ F	+ 2.13
Time	194.66 sec	+ 0.25 sec	+ 0.13
Weight	50 lbf	+ 0.5 lbf	+ 1.00
Flow rate (A20-A24)	0.256 lbm/sec	+ 0.00258 lbm/sec	+ 1.01
Flow rate (B20-B27)	0.257 lbm/sec	+ 0.00259 lbm/sec	+ 1.01

TABLE 2. Summary of Repeatability for Data A5, A20 through A24 and B5, B20 through B27

<u>Parameter</u>	<u>Average Value</u>	<u>High Value -Low Value</u>	<u>Percent</u>
Flow rate (A5, A20-A24)	7.66 SCFM	0.08	1.04
Flow rate (B5, B20-B27)	7.66 SCFM	1.30	1.70

ADDENDUM 1

Rotameter Calibration Instrumentation List

<u>Item</u>	<u>Identifying Code (e.g., Serial Number)</u>	<u>Calibration Date</u>	<u>Expiration Date</u>	<u>Calibration Reference Number</u>
Digital timer Durgin & Browne LT BX10 A	7807 (sn)	3-12-81	3-12-82	HEDL 374-18- 02-002
500 lb scale Fairbanks-Morse	USA-HEW 457-698	3-6-81	3-6-82	HEDL 374-66- 01-001
Thermocouple Type J	#57	3-20-81	3-20-82	HEDL 121821
Fluke Digital Thermometer	0790011 (sn)	1-20-82	1-20-83	HEDL 374-79- 06-001

ADDENDUM 3

[illegible]

INSERT DATE 3-20-81		Hanford Engineering Development Laboratory <small>Washington Hanford Engineering & Technology A Subsidiary of Washington Electric Corporation P. O. Box 1478, Richland, WA 99352</small>		PHYSICAL AND ELECTRICAL STANDARDS LAB REPORT				REFERENCE NUMBER 121821					
02 INSTRUMENT NAME, IFCR/MODEL, SUB CATEGORY THERMOCOUPLES 2 EA TYPE J		05 STANDARDS CODE NUMBER 18		01 HISTORY		17 SERVING CODE KE3600		22 CUSTOMER CODE D7H57		23 WORK ORDER NO. A78570			
03 CUSTODIAN/ADDRESS HANFORD RD 1890-1000A PNL		04 SENDER		07 SERIAL 00 NEW		19 CUSTOMER/ APPROVAL 29. 1. WIRE 8. UHI 9. OTHER 10. TEST 2. LAG 9. HNH A. 004 11. VITRO 3. PUL 7. DCH 4. ANNO 8. JAJ		20 WORK S		21 TECH NO			
06 EQUIPMENT SPECIFICATIONS A N S I 5		09 ROOM NO.		10 INST. RLDG.		11 RECALL CYCLE		21 SERV. CATEG. 4 78		22 DATE RECEIVED 03 20			
07 LAB CODE		12 DATE CALIB.		13 DATE DUE		14 RECALL STATUS 1. FROM TODAY 2. FIXED 3. SURFING		15 TOL. HISTORY		16 TOL. REC. 1. IN 2. OUT 3. NA			
08 EQUIPMENT USED C02-55-01-004, C02-49-02-005 C02-55-01-002, C02-49-02-004 C02-72-02-017, C02-79-02-014 5.04 5.6 6.848 5.12 5.7 6.849 7.05 6.0 6.850 7.11 6.7 6.850 1.66 3.0 6.866 1.66 3.7 6.866		09 PROCEDURE NO. R011004H in a bath type T wire 5.6 5.6 149.87 5.7 149.89 5.6 174.83 5.7 174.89		10 CALIBRATION DATA / REMARKS 1.50 1.50 1.80 1.80		11 TRAINING HOURS 1. 6 2. 1 3. 1 4. 1 5. 1 6. 1 7. 1 8. 1 9. 1 10. 1		12 MATERIALS 0. 1		13 FIXED PRICE 70 TOTAL 227		14 DATE SHIPPED 03 2	
Calibration Traceable To The National Bureau Of Standards Or Nationally Recognized Standards													
APPROVED BY: [Signature]													

INSERT	DATE	Hanford Engineering Development Laboratory		Westinghouse Hanford Company A subsidiary of Westinghouse Electric Corporation P. O. Box 1970, Richland, WA 99352		PHYSICAL AND ELECTRICAL STANDARDS LAB REPORT				REFERENCE NUMBER	
MODIFY	12-1-81									140280	
02 INSTRUMENT NAME, MFGR/MODEL, SUB CATEGORY		STANDARDS CODE NUMBER		01 HISTORY		SERVICING CODE		CUSTOMER CODE		WORK ORDER NO.	
THE MONITOR DIGITL		374-75-16-011		H		M1110		VE2160		F71525	
03 FLUORE 2166A		07 SERIAL		08 NEW		19 CUSTOMER/ APPROVAL		20 WORK			
04		71111				29 1. WNC 5. UNI 9. OTHER U. U.S. TEST 2. LAB 6. HEN A. DOE V. VITRO 3. PNL 7. BCS 4. ARHCO 8. JAJ		80 L-L P-IN			
05 CUSTODIAN/ADDRESS		09 ROOM NO.		10 INST. BLDG.		11 RECALL CYCLE		21 SERV. DEPT. CATEG.		DATE RECEIVED	
H3 KLE RD				100		360		31 4 32 70 33 34		01/19	
06 100-1000A-1L		12 SUB CATEG.		13 DATE CALIB.		DATE DUE		14 RECALL STATUS		15 TOL. HISTORY	
				01/20/82		01/20/83		1 1 FROM TODAY 2 FIXED 3 SUSPEND		1	
EQUIPMENT SPECIFICATIONS		16		17		18		19		20	
400-011111-111111											
EQUIPMENT USED		PROCEDURE NO.									
002-49-61-014		002-73-62-114		002-49-62-115							
CALIBRATION DATA/REMARKS											
all points within $\pm 1\%$ over specified range											
Calibration Traceable To The National Bureau Of Standards Or Nationally Recognized Standards											
APPROVED BY:											
D. Binstaff											

ADDENDUM 6

CALIBRATION DATA

A. Meter L544218

Run no.	Weight (lb)	Time (sec)	Temp (°F)	ρ (lb/ft ³)	Q (SCFM)	Rotameter Reading
1	100	269.02	55	62.39	11.08	11.3 ₊ .1
2	100	271.03	55	62.39	11.00	11.0 ₊ .25
3	100	305.28	54	62.39	9.79	10.0 ₊ .25
4	100	344.92	52	62.40	8.67	9.0 ₊ .25
5	100	387.19	50	62.41	7.72	8.0*
6	100	446.14	49	62.41	6.68	7.0
7	100	447.65	48	62.41	6.68	7.0
8	100	527.78	48	62.41	5.64	6.0
9	100	637.20	48	62.41	4.69	5.0
10	100	804.12	48	62.41	3.69	4.0
11	100	1095.68	48	62.41	2.74	3.0
12	100	1650.39	49	62.41	1.83	2.0
13	50	1502.39	51	62.41	1.00	1.2
14	50	812.78	51	62.41	1.83	2.0
15	50	540.85	51	62.41	2.74	3.0
16	50	401.19	51	62.41	3.74	4.0
17	50	317.71	51	62.41	4.69	5.0
18	50	261.92	51	62.41	5.69	6.0
19	50	223.21	51	62.41	6.68	7.0
20	50	195.25	50	62.41	7.64	8.0 ₊ .25
21	50	195.54	50	62.41	7.64	8.0 ₊ .25
22	50	195.09	50	62.41	7.64	8.0 ₊ .25
23	50	195.10	50	62.41	7.64	8.0 ₊ .25
24	50	195.80	49	62.41	7.64	8.0 ₊ .25
25	50	173.88	48	62.41	8.59	9.0 ₊ .25
26	50	155.12	48	62.41	9.63	10.0 ₊ .25
27	50	138.41	48	62.41	10.79	11.0 ₊ .25
28	50	134.09	47	62.42	11.12	11.4*
29	100	268.73	47	62.42	11.12	11.4

* Steady reading.

ADDENDUM 6

CALIBRATION DATA

B. Meter L544217

<u>Run no.</u>	<u>Weight (lb)</u>	<u>Time (sec)</u>	<u>Temp (°F)</u>	<u>ρ (lb/ft³)</u>	<u>Q (SCFM)</u>	<u>Rotameter Reading</u>
1	100	265.45	55	62.39	11.25	11.4
2	100	275.62	50	62.41	10.83	11.0
3	100	308.25	49	62.41	9.67	10.0
4	100	344.86	48	62.41	8.67	9.0
5	100	389.39	47	62.42	7.68	8.0
6	100	449.88	47	62.42	6.64	7.0
7	100	531.21	47	62.42	5.60	6.0
8	100	633.67	47	62.42	4.69	5.0
9	100	791.26	47	62.42	3.78	4.0
10	50	526.01	48	62.41	2.82	3.0
11	50	799.32	48	62.41	1.87	2.0
12	50	1369.46	49	62.41	1.08	1.2
13	50	794.06	50	62.41	1.87	2.0
14	50	793.54	50	62.41	1.87	2.0
15	50	533.74	49	62.41	2.78	3.0
16	50	391.10	49	62.41	3.82	4.0
17	50	312.86	48	62.41	4.77	5.0
18	50	262.13	48	62.41	5.69	6.0
19	50	223.34	48	62.41	6.68	7.0
20	50	194.37	47	62.42	7.68	8.0
21	50	195.93	47	62.42	7.59	8.0
22	50	194.42	47	62.42	7.68	8.0
23	50	193.75	47	62.42	7.72	8.0
24	50	196.40	46	62.42	7.59	8.0
25	50	192.94	46	62.42	7.72	8.0
26	50	194.23	46	62.42	7.68	8.0
27	50	195.26	46	62.42	7.64	8.0
28	55	190.10	46	62.42	8.63	9.0
29	50	153.98	46	62.42	9.67	10.0
30	50	138.19	45	62.42	10.79	11.0
31	50	134.11	45	62.42	11.12	11.4
32	50	133.58	45	62.42	11.16	11.4
33	50	173.54	45	62.42	8.59	9.0

ADDENDUM 7

UNCERTAINTY ANALYSIS

Consider a dependent variable y which is a continuous and differentiable function of n independent variables x_n . The uncertainty interval of y , ω_y , is given by:⁽²⁾

$$\omega_y = \left\{ \left(\frac{\partial y}{\partial x_1} \right)^2 x_1^2 + \left(\frac{\partial y}{\partial x_2} \right)^2 \omega_{x_2}^2 + \dots + \left(\frac{\partial y}{\partial x_n} \right)^2 \omega_{x_n}^2 \right\}^{1/2} \quad (7.1)$$

Partial derivatives in Equation 7.1 are determined by stating the functional dependence of y on the independent variables and then performing the indicated differentiation. Estimates of uncertainty intervals originate in a variety of sources. A typical example is to use one half of the minor gradation of an etched scale.

Mass flow rate is a function of the mass collected and the time of collection, hence

$$\dot{m} = \frac{m}{\Delta t} \quad (7.2)$$

Differentiating Equation 7.2 and arranging the terms as in Equation 7.1 yields:

$$\omega_{\dot{m}} = \left[\left(\frac{\partial \dot{m}}{\partial \Delta t} \right)^2 \omega_{\Delta t}^2 + \left(\frac{\partial \dot{m}}{\partial m} \right)^2 \omega_m^2 \right]^{1/2} \quad (7.3)$$

For this study the volume flow rate uncertainty equals the mass flow uncertainty since it can be shown that the uncertainty in water density is negligible for the temperature uncertainty encountered.

The mass flow rate uncertainty interval, $\omega_{\dot{m}}$, was calculated for the average of points A20 through A24 for rotameter L544218 and points B20 through B27 for rotameter L544217. The results are summarized in Table 3.

TABLE 3. Summary of Uncertainty Interval Calculations for Average of Points B20 through B27 and Average of Points A20 through A24

Meter L44218
A20 through A24

<u>Parameter</u>	<u>Value</u>	<u>Uncertainty Interval</u>	<u>Percent Error, %</u>
$\dot{m}$	50 lbm	$\pm 0.5 \text{ lbm}$	1.00
t	195.36s	$\pm 0.25\text{s}$	0.13
m	0.256 lbm/s	$\pm 0.00258 \text{ lbm/sec}$	1.01

Meter L44217
B20 through B27

m	50 lbm	$\pm 0.5 \text{ lbm}$	1.00
t	194.66 s	$\pm 0.25 \text{ s}$	0.13
$\dot{m}$	0.257 lbm/s	$\pm 0.00259 \text{ lbm/sec}$	1.01

APPENDIX B

TEST STAND CALIBRATION DOCUMENTS

APPENDIX B
TEST STAND CALIBRATION DOCUMENTS

Test Stand Instrumentation List

<u>Item</u>	<u>Identifying Code (e.g., Serial Number)</u>	<u>Calibration Date</u>	<u>Expiration Date</u>	<u>Calibration Reference Number</u>
Schutte & Koerting model 61-06-692- M rotameters (2)	L544217 U.S. Gov't. L544218 U.S. Gov't.	1-28-82 1-27-82	1-28-83 1-27-83	See App. A See App. A
Thermocouples, (2) Type J	#683, #668	3-08-82	3-08-83	HEDL 145575
Validyne pres- sure transducer, P305D		3-22-82	3-22-83	HEDL 145790
Heise pressure gage		10-01-81	10-01-82	HEDL 124436 HEDL 512-31-04-003
Pioneer Photo- tach 0-100,000 rpm	PI 182 408	2-16-82	2-16-83	HEDL 131543
Load cell, Inter- face minibeam MB-5-4	A16940	3-22-82	3-22-83	HEDL 145791
Keithley 191 Digital multi- meter	22895	9-10-81	9-30-82	Manufacturer calibration
Fluke Digital Thermometer	0790011	1-20-82	1-20-83	HEDL 374-79-06-001
DC Power Supplies (2)	L544795 U.S. Gov't.	not applicable		
Power Design, Inc. 4005	L545663 U.S. Gov't.			

INSTRUMENT NAME, MFG/ MODEL, SUB CATEGORY		STANDARDS CODE NUMBER		01 HISTORY		SERVICING CODE		CUSTOMER CODE		WORK ORDER NO.	
THERMOCOUPLES 2EA						V112160		V112160		F71535	
03		07 SERIAL		08 NEW		19 CUSTOMER/ APPROVAL		20 WORK SITE			
04						29. 1. WHC 5. UNI 9. OTHER U.U.S. TEST 2. LAB 6. MEH A. DOC V. VITRO 3. PNL 7. BCS 4. ARHCO 8. JAJ		30 L-LAB P-INPLACE			
05 CUSTODIAN/ADDRESS		09 ROOM NO.		10 INST. BLDG.		11 RECALL CYCLE		21 SERV. CATEG.		22 DATE RECEIVED	
HINKLE KD				189D				4 78		02 24	
06 189D-100DA PNL		12 SUB CATEG.		13 DATE CALIB.		DATE DUE		14 RECALL STATUS		15 TOL. HISTORY	
		02		03/08/82				1. FROM TODAY 2. FIXED 3. SUSPEND		1. IN 2. OUT 3. NA	
EQUIPMENT SPECIFICATIONS		EQUIPMENT USED		PROCEDURE NO.		TRAINING HOURS		CALIBRATION HOURS		ACCEPTANCE HOURS	
A1151 27d grade 3 spec		002-55-01-004, 002-79-00-004 002-49-02-001, 002-12-07-001 002-77-02-004, 002-49-02-003		P1111111111111111				1. 4.0		2. 1.0	
CALIBRATION DATA/REMARKS		4683		4683		46.68		3. 1.0		4. 1.0	
Std temp °F		T1 TC		T2 TC		Std temp °F		T1 TC		T2 TC	
32		1.2		1.2		1.2		1.2		1.2	
50		1.25		1.25		1.25		1.25		1.25	
75		1.25		1.25		1.25		1.25		1.25	
100		1.25		1.25		1.25		1.25		1.25	
125		1.25		1.25		1.25		1.25		1.25	
Calibration Traceable To The National Bureau Of Standards Or Nationally Recognized Standards		APPROVED BY:						18 FORCED NEXT CALIB.		DATE SHIPPED	
								75		78	
								03		08	

5D-3

11 REFERENCE NUMBER 16
145790

INSERT MODIFY		DATE 2-17-82	Hanford Engineering Development Laboratory <small>Westinghouse Hanford Company A subsidiary of Westinghouse Electric Corporation P O Box 1870 Richland Wa 99352</small>	PHYSICAL AND ELECTRICAL STANDARDS LAB REPORT													
02 INSTRUMENT NAME, MFGR/MODEL, SUB CATEGORY X-DUCER PRESSURE VALDYNE P305D		STANDARDS CODE NUMBER 10		01 HISTORY		SERVICING CODE Vm460		CUSTOMER CODE DTH57		WORK ORDER NO. 23 28 B17961							
03 04		07 SERIAL		08 HEW		19 CUSTOMER/ APPROVAL 29. 1.WHC 5.UNI 9.OTHER U.U.S.TEST 2.LAB 6.HEH A.DOE V.VITRO 3.PNL 7.BCS 4.ARMCO 8.JAJ		20 WORK SITE 30 L-LAB P-INPLACE		31							
05 CUSTODIAN/ADDRESS HENKLE KD		SENDER		09 ROOM NO.		10 INST. BLDG. 189D		11 RECALL CYCLE		21 SERV. DEPT. CATEG. 5 80		DATE RECEIVED 02 02		JOB COUNT 38 MORE THAN ONE		TECH NO. 40 42 030	
06 189D-100DA PNL		12 SUB CATEG. 02		13 DATE CALIB. 13-22-82		DATE DUE		14 RECALL STATUS 1 FROM TODAY 2 FIXED 3 SUSPEND		15 TOL. HISTORY		TOL.REC. 1.IN 2 OUT 3.NA 43 44 45		SPLIT 46 49			
EQUIPMENT SPECIFICATIONS												TRAINING HOURS					
EQUIPMENT USED (12-81-04-016)												CALIBRATION HOURS 1.					
002-45-05-011 002-14-01-015												ACCEPTANCE HOURS 2.					
002-14-01-015 002-14-01-015												REPAIR HOURS 3. 1.0					
002-14-01-015 002-14-01-015												FABRICATE HOURS 4. 1.0					
002-14-01-015 002-14-01-015												OTHER HOURS 5.					
002-14-01-015 002-14-01-015												MATERIALS 6.					
002-14-01-015 002-14-01-015												FIXED PRICE 70		TOTAL		71 74	
002-14-01-015 002-14-01-015												18 FORCED NEXT CALIB.		DATE SHIPPED		75 78	
Calibration Traceable To The National Bureau Of Standards Or Nationally Recognized Standards												03 02					
APPROVED BY:																	

REC'D
STOLAB
30

B.3

54-7310-052 (5-81)

DATE 11-1-81		Manufacturer Hartford Engineering Development Laboratory		PHYSICAL AND ELECTRICAL STANDARDS LAB REPORT								11. REFERENCE NUMBER 124436	
INSTRUMENT NAME, MFG/MODEL, SUB CATEGORY GAGE PRESSURE HEISE 0-2000 PSI				STANDARD CODE NUMBER 51		U1 HISTORY 11		SERVICING CODE V-160		CUSTOMER CODE 5D-4		WORK ORDER NO. 5D-4	
CUSTODIAN ADDRESS JTV				SENDER P.W.		07 SERIAL		08 NEW		19 CUSTOMER/ APPROVAL		20 WORK SITE	
09 ROOM NO.				10 INST. BLDG		11 RECALL CYCLE		21 SER. CATEG.		DATE RECEIVED		JOB COUNT	
12 LAB CODE				13 DATE CALIB		DATE DUE		14 RECALL STATUS		15 TOL HISTORY		TOL REC	
16 SPECIFICATIONS				17		18		19		20		21	
PROCEDURE NO.				22		23		24		25		26	
CALIBRATION DATA / REMARKS				27		28		29		30		31	
800				199.9		1750		1750.0		1750.0		1750.0	
300				500.8		2000		1750.0		1750.0		1750.0	
750				750.1		0		0		0		0	
1000				1000.1									
1250				1250.2									
1500				1500.4									
TOTAL				71		74		75		78		78	

[illegible]

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J.L. ACKERMAN

Keithley Instruments, Inc.
28775 Aurora Road
Cleveland, Ohio 44139
(216) 248-0400 Telex: 98-5469

CERTIFICATE OF COMPLIANCE
AND CALIBRATION

MODEL 191 SERIAL NO. 22895 DATE September 10, 1981

This notification serves to certify that the unit described above has been inspected and tested in accordance with specifications published by Keithley Instruments, Inc.

The accuracy and calibration of this instrument are traceable to the National Bureau of Standards through equipment which is calibrated at planned intervals by comparison to certified standards maintained in the Laboratories of Keithley Instruments, Inc.

The Calibration procedures of this company are in compliance with MIL-C-45662A.

R. Schroeder
Calibration Technician

Edward T. Kifer
Edward T. Kifer
Quality Assurance Manager

ETK:dr

INSERT	DATE 12-1-81	Hanford Engineering Development Laboratory	Westinghouse Hanford Company A subsidiary of Westinghouse Electric Corporation P.O. Box 1970 Richland WA 99352	PHYSICAL AND ELECTRICAL STANDARDS LAB REPORT				11 REFERENCE NUMBER 140280	
MODIFY				STANDARDS CODE NUMBER 374-72-4-011		01 HISTORY H	SERVICING CODE VITRO	CUSTOMER CODE VE2160	WORK ORDER NO. F71535
02 INSTRUMENT NAME, MFGR/MODEL, SUB CATEGORY THE MONITOR DIGITL				07 SERIAL 70511		08 NEW	19 CUSTOMER/ APPROVAL		20 WORK SITE
03 FLING 2166A							29 1. WNC 5. UNI 9. OTHER U.S. TEST 2. LAB 6. HEM A. DOE V. VITRO 3. PNL 7. ECB 4. ARNCD 8. JAJ		30 L-LAB P-INPLX
04				09 ROOM NO.		10 INST. BLDG.	11 RECALL CYCLE	21 SERV. DEPT.	22 DATE RECEIVED
05 CUSTODIAN/ADDRESS HICKLE RD				SENDER				31 4	32 70
06 102-1000A-11L				12 SUB CATEG.		13 DATE CALIB.	DATE DUE	14 RECALL STATUS	15 TOL. HISTORY
						01/20/82	01/21/83	1 FROM TODAY 2. FIXED 3. SUSPEND	1 IN 2 OUT 3. NA
EQUIPMENT SPECIFICATIONS				EQUIPMENT USED					
				PROCEDURE NO.					
				CALIBRATION DATA/REMARKS					
				all points within $\pm 1\%$ over specified range					
				TRAINING HOURS					
				CALIBRATION HOURS 1. 2.5					
				ACCEPTANCE HOURS 2. 0.3					
				REPAIR HOURS 3. 0.0					
				FABRICATE HOURS 4. 0.0					
				OTHER HOURS 5. 0.3					
				MATERIALS 6. 0.0					
				FIXED PRICE 70					
				TOTAL 98					
				18 FORCED NEXT CALIB					
				DATE SHIPPED					
				78 11 2.					
Calibration Traceable To The National Bureau Of Standards Or Nationally Recognized Standards									
APPROVED BY: <i>D. Benthoff</i>									

APPENDIX C

DATA SHEETS

APPENDIX C

DATA SHEETS

DATA SHEET LEGEND

Roman numerals specify the class of data:

- I Dynamic zeroes
- II Performance data
- III Preoperational test stand response tests.

Upper case letters denote the test stand configuration:

- A Two open turbine bearings and shaft #1,
- B cleaned and reassembled as in configuration A,
- C two sealed turbine bearings and shaft #1,
- D two sealed turbine bearings and shaft #2,
- E one open turbine bearing, one sealed turbine bearing and shaft #2

Note: Performance data was not taken with configuration D.

Static zero - load cell response when turbine occupied its position
in the test stand, rotor stationary

Dynamic zero - air brake wheel and shroud removed, rotor spinning

Single asterisk, *, denotes a no load condition. The air brake wheel and shroud were in place but there was no air flow to the brake.

Double asterisk, **, denotes a maximum speed test performed under the no load condition.

I. DYNAMIC ZEROES

Data Set	Air Temperature		Excitation Voltage (V DC)	Pressure		Absolute Pressure	
	Rotameter	Turbine		Output Voltages		Rotameter	Turbine
	Inlet (°F)	Inlet (°F)		Rotameter Inlet (V DC)	Turbine Inlet (V DC)	Rotameter Inlet (psia)	Turbine Inlet (psia)
A1a.							
b.	73	73	12.43	4.02	.283	115.0	21.6
c.	73	73	12.43	4.02	.283	115.0	21.6
d.	72	72	12.43	3.98	.292	114.0	21.8
e.	72	72	12.43	3.98	.305	114.0	22.1
f.	72	72	12.43	3.96	.304	113.5	22.1
g.	73	72	12.43	4.02	.310	115.0	22.3
A2a.							
b.	72	72	12.37	4.03	.305	115.3	22.1
c.	72	72	12.37	4.06	.313	116.0	22.3
d.	72	72	12.37	4.06	.321	116.0	22.5
e.	72	72	12.37	4.06	.319	116.0	22.5
f.							
A3a.							
b.	72	72	12.37	4.02	.309	115.0	22.2
c.	72	72	12.36	3.97	.329	113.8	22.7
d.	72	72	12.36	4.02	.314	115.0	22.4
e.	72	72	12.36	3.96	.322	113.5	22.6
f.							
A4a.							
b.	74	73	12.36	3.95	.625	113.3	30.1
c.	73	73	12.36	4.01	.596	114.8	29.4
d.	73	73	12.36	4.03	.598	115.3	29.5
e.	73	73	12.36	3.95	.586	113.3	29.2
f.							
A5a.							
b.	73	73	12.36	4.04	.608	115.5	29.7
c.	73	73	12.36	4.02	.602	115.0	29.6
A6a.							
b.	73	73	12.36	3.99	.595	114.3	29.4
c.	73	73	12.36	3.91	.583	112.3	29.1
d.	73	73	12.36	3.90	.583	112.0	29.1
e.							

I. DYNAMIC ZEROES

Data Set	Flow Rate		Load Cell Response				Speed (rpm)
	Volume (SCFM ₀)	Mass (lbm/sec)	Static Zero (V DC)	Excitation Voltage (V DC)	Output Voltage (mV)	Force (lbf)	
A1a.			.446	8.45			
b.	1.0	.0028		8.45	.475	.0936	10000
c.	1.0	.0028		8.45	.464	.0914	10000
d.	1.0	.0028		8.45	.467	.0920	10000
e.	1.0	.0028		8.46	.464	.0914	10000
f.	1.0	.0028		8.46	.465	.0916	10000
g.	1.0	.0028		8.46	.470	.0926	10000
A2a.			.492	8.45			
b.	1.0	.0028		8.45	.481	.0949	10000
c.	1.0	.0028		8.46	.472	.0930	10000
d.	1.0	.0028		8.46	.482	.0950	10000
e.	1.0	.0028		8.46	.470	.0926	10000
f.			.483	8.46			
A3a.			.547	8.46			
b.	1.0	.0028		8.46	.472	.0930	10000
c.	1.0	.0028		8.46	.478	.0942	10000
d.	1.0	.0028		8.46	.482	.0950	10000
e.	1.0	.0028		8.46	.480	.0946	10000
f.			.496	8.46			
A4a.			.501	8.46			
b.	1.6	.0048		8.46	.452	.0890	20000
c.	1.5	.0045		8.46	.455	.0896	20000
d.	1.5	.0045		8.46	.460	.0906	20000
e.	1.5	.0045		8.46	.450	.0887	20000
f.			.483	8.46			
A5a.			.550	8.46			
b.	1.5	.0045		8.46	.455	.0896	20000
c.	1.5	.0045		8.46	.453	.0892	20000
A6a.			.466	8.46			
b.	1.5	.0045		8.46	.464	.0914	20000
c.	1.5	.0045		8.46	.459	.0904	20000
d.	1.5	.0045		8.46	.462	.0910	20000
e.			.458	8.46			

I. DYNAMIC ZEROES

Data Set	Air Temperature		Excitation Voltage (V DC)	Pressure		Absolute Pressure	
	Rotameter	Turbine		Output Voltages		Rotameter	Turbine
	Inlet (°F)	Inlet (°F)		Rotameter Inlet (V DC)	Turbine Inlet (V DC)	Rotameter Inlet (psia)	Turbine Inlet (psia)
A7a.							
b.	73	73	12.36	3.95	.977	113.3	38.9
c.	73	73	12.36	3.94	1.095	113.0	41.9
d.	73	73	12.36	3.92	1.106	112.5	42.2
e.	73	73	12.36	3.88	1.108	111.5	42.2
f.	73	73	12.36	3.84	1.099	110.5	42.0
g.							
A8a.							
b.	74	74	12.36	3.85	1.582	110.8	54.1
c.	74	74	12.36	3.78	1.650	109.0	55.8
d.	74	74	12.36	3.79	1.68	109.3	56.5
e.	74	74	12.36	3.74	1.66	108.0	56.0
f.							
B1a.							
b.	75	75	12.45	4.03	.1801	115.1	18.9
c.	75	75	12.45	3.99	.1804	114.1	18.9
d.	75	75	12.45	4.07	.1857	116.1	19.0
e.	75	75	12.45	4.03	.1825	115.1	18.9
B2a.	75	75	12.45	4.04	.501	115.4	26.9
b.	75	75	12.45	3.95	.4913	113.1	26.7
c.	75	75	12.45	3.96	.4914	113.4	26.7
d.	75	75	12.45	3.96	.4892	113.4	26.6
B3a.	75	75	12.45	3.98	.9787	113.9	38.8
b.	75	75	12.45	3.98	.9809	113.9	38.9
c.	75	75	12.45	3.98	.9770	113.9	38.8
d.	75	75	12.45	3.89	.9737	111.6	38.7
B4a.	75	75	12.45	3.90	1.633	111.9	55.2
C1a.							
b.	73	73	12.58	4.07	.1182	116.1	17.3
c.	73	73	12.58	4.09	.1209	116.6	17.4
d.	73	73	12.58	4.08	.1178	116.3	17.3
e.	73	73	12.58	4.04	.1327	115.3	17.7

I. DYNAMIC ZEROES

Data Set	Flow Rate		Load Cell				Speed (rpm)
	Volume (SCFM ₀)	Mass (lbm/sec)	Static Zero (V DC)	Excitation Voltage (V DC)	Output Voltage (mV)	Force (lbf)	
A7a.			.550	8.46			
b.	2.1	.0065		8.46	.495	.0975	30000
c.	2.3	.0072		8.46	.580	.1143	30000
d.	2.3	.0072		8.46	.560	.1103	30000
e.	2.3	.0071		8.46	.555	.1093	30000
f.	2.3	.0071		8.46	.545	.1074	30000
g.			.485	8.46			
A8a.			.507	8.46			
b.	3.0	.0095		8.46	.566	.112	40000
c.	3.1	.0097		8.46	.540	.106	40000
d.	3.3	.0104		8.46	.520	.102	40000
e.	3.3	.0103		8.46	.505	.100	40000
f.			.481	8.46			
B1a.			.422	8.47			
b.	<1	<.0028		8.47	.436	.0858	10000
c.	<1	<.0028		8.47	.429	.0844	10000
d.	<1	<.0028		8.47	.433	.0852	10000
e.	<1	<.0028		8.47	.430	.0846	10000
B2a.	1.4	.0042		8.47	.430	.0846	20000
b.	1.4	.0041		8.47	.435	.0856	20000
c.	1.4	.0041		8.47	.430	.0846	20000
d.	1.4	.0041		8.47	.430	.0846	20000
B3a.	2.1	.0065		8.47	.440	.0866	30000
b.	2.1	.0065		8.47	.450	.0885	30000
c.	2.1	.0065		8.47	.450	.0885	30000
d.	2.1	.0065		8.47	.460	.0905	30000
B4a.	3.1	.0098		8.47	.460	.0905	40000
C1a.			.447	8.47			
b.	<1	<.0028		8.47	.470	.0925	10000
c.	<1	<.0028		8.47	.470	.0925	10000
d.	<1	<.0028		8.47	.475	.0935	10000
e.	<1	<.0028		8.47	.470	.0925	10000

I. DYNAMIC ZEROES

Data Set	Air Temperature		Excitation Voltage (V DC)	Pressure		Absolute Pressure	
	Rotameter Inlet (°F)	Turbine Inlet (°F)		Output Voltages		Rotameter Inlet (psia)	Turbine Inlet (psia)
				Rotameter Inlet (V DC)	Turbine Inlet (V DC)		
C2a.	73	73	12.58	3.97	.4544	113.6	25.7
b.	72	72	12.58	3.97	.4555	113.6	25.7
c.	73	73	12.58	4.00	.4596	114.3	25.8
d.	72	72	12.58	4.06	.4688	115.8	26.1
C3a.	72	72	12.58	3.97	.8543	113.6	35.7
b.	72	72	12.58	3.92	.8450	112.3	35.5
c.	72	72	12.58	3.92	.8469	112.3	35.5
d.	72	72	12.58	3.99	.864	114.1	35.9
C4a.	73	73	12.58	3.84	1.457	110.3	50.8
b.	73	73	12.58	3.83	1.457	110.1	50.8
c.	73	73	12.58	3.87	1.468	111.1	51.0
d.	73	73	12.58	3.85	1.462	110.6	50.9
C5a.	73	73	12.58	3.71	2.322	107.1	72.4
b.	73	73	12.58	3.74	2.357	107.8	73.3
c.	73	73	12.58	3.75	2.357	108.1	73.3
d.	73	73	12.58	3.77	2.371	108.6	73.6
				Relative Pressure		Absolute Pressure	
				Rotameter Inlet (psig)	Turbine Inlet (psig)	Rotameter Inlet (psia)	Turbine Inlet (psia)
D1a.	73	73		100.4	-1.3	118.5	16.8
b.	73	73		99.7	-1.2	117.8	16.9
D2a.	73	73		97.4	5.4	115.5	23.5
b.	73	73		97.0	5.3	115.1	23.4
D3a.	72	72		97.2	14.8	115.3	32.9
b.	72	72		96.5	14.7	114.6	32.8
D4a.	72	72		93.5	22.5	111.6	40.6
b.	72	72		95.1	22.8	113.2	40.9

I. DYNAMIC ZEROES

Set	Flow Rate		Static Zero (V DC)	Load Cell Response			Speed (rpm)
	Volume (SCFM ₀)	Mass (lbm/sec)		Excitation Voltage (V DC)	Output Voltage (mV)	Force (lbf)	
C2a.	1.3	.0038		8.47	.475	.0935	20000
b.	1.3	.0038		8.47	.475	.0935	20000
c.	1.3	.0038		8.47	.480	.0945	20000
d.	1.3	.0038		8.47	.475	.0935	20000
C3a.	1.9	.0058		8.47	.470	.0925	30000
b.	1.9	.0058		8.47	.473	.0931	30000
c.	1.9	.0058		8.47	.470	.0925	30000
d.	1.9	.0059		8.47	.485	.0954	30000
C4a.	2.9	.0091		8.47	.490	.0964	40000
b.	2.9	.0091		8.47	.480	.0945	40000
c.	2.9	.0091		8.47	.490	.0964	40000
d.	2.9	.0091		8.47	.490	.0964	40000
C5a.	4.3	.0136		8.47	.505	.0994	50000
b.	4.3	.0136		8.47	.515	.1013	50000
c.	4.3	.0137		8.47	.505	.0994	50000
d.	4.3	.0137		8.47	.510	.1004	50000
D1a.	<1	<.0028		8.47	.505	.0994	10000
b.	<1	<.0028		8.47	.505	.0994	10000
D2a.	1.5	.0045		8.47	.540	.1063	20000
b.	1.5	.0045		8.47	.550	.1082	20000
D3a.	2.3	.0073		8.47	.600	.1181	30000
b.	2.3	.0072		8.47	.600	.1181	30000
D4a.	3.0	.0095		8.47	.580	.1141	40000
b.	3.0	.0096		8.47	.585	.1151	40000

I. DYNAMIC ZEROES

Data Set	Air Temperature		Pressure			
	Rotameter Inlet (°F)	Turbine Inlet (°F)	Relative Pressure		Absolute Pressure	
			Rotameter Inlet (psig)	Turbine Inlet (psig)	Rotameter Inlet (psia)	Turbine Inlet (psia)
E1a.	73	73	98.3	-1.2	116.4	16.9
b.	73	73	96.6	-1.3	114.7	16.8
E2a.	73	73	94.8	4.8	112.9	22.9
b.	73	73	95.0	5.3	113.1	23.4
c.	73	73	96.5	5.3	114.6	23.4
E3a.	73	73	93.5	13.8	111.6	31.9
b.	72	72	96.5	14.8	114.6	32.9
E4a	72	72	95.1	22.0	113.2	40.1

I. DYNAMIC ZEROES

Set	Flow Rate		Load Cell Response				Speed (rpm)
	Volume (SCFM _o)	Mass (lbm/sec)	Static Zero (V DC)	Excitation Voltage (V DC)	Output Voltage (mV)	Force (lbf)	
E1a.	1	.0028		8.47	.580	.1141	10000
b.	1	.0028		8.47	.590	.1161	10000
2a.	1.4	.0041		8.47	.600	.1181	20000
b.	1.5	.0045		8.47	.605	.1190	20000
c.	1.5	.0045		8.47	.600	.1181	20000
3a.	2.3	.0071		8.47	.640	.1259	30000
b.	2.3	.0072		8.47	.660	.1299	30000
4a.	3.0	.0096		8.47	.630	.1240	40000

I. DYNAMIC ZEROES

Summary		
Data Set	Speed (rpm)	Dynamic Zero (lbf) avg.
A1,2,3	10000	.093
A4,5,6	20000	.090
A7	30000	.108
A8	40000	.105
B1	10000	.085
B2	20000	.085
B3	30000	.089
B4	40000	.091
C1	10000	.092
C2	20000	.094
C3	30000	.093
C4	40000	.096
C5	50000	.100
D1	10000	.099
D2	20000	.107
D3	30000	.118
D4	40000	.115
E1	10000	.115
E2	20000	.118
E3	30000	.128
E4	40000	.124

II. PERFORMANCE DATA

Data Set	Air Temperature		Pressure				
	Rotameter Inlet (°F)	Turbine Inlet (°F)	Output Voltages				
			Transducer Excitation Voltage (V DC)	Rotameter Inlet (V)	Turbine Inlet (V)	Rotameter Inlet (psia)	Turbine Inlet (psia)
A1a.							
b*	73	72	12.19	3.96	.620	113.3	29.8
c.	74	72	12.19	3.76	.822	108.3	34.8
d.	74	73	12.19	3.58	1.21	103.8	44.5
e.	73	73	12.19	3.65	1.57	104.8	53.5
f.	73	73	12.19	3.63	2.02	105.0	64.8
g.	73	73	12.19	3.65	2.31	105.5	72.0
h.	73	73	12.19	3.55	2.76	103.0	83.3
i.	73	73	12.19	3.49	3.31	101.5	97.0
A2a.*	74	73	12.19	3.83	1.28	110.0	46.3
b.	74	73	12.19	3.60	1.59	104.3	54.0
c.	74	73	12.19	3.57	2.01	103.5	64.5
d.	73	73	12.19	3.54	2.40	102.8	74.3
e.	73	73	12.19	3.48	2.76	101.3	83.3
f.	73	73	12.19	3.50	3.17	101.8	93.5
g.							
A3a.							
b.*	73	73	12.80	3.57	1.88	103.7	61.4
c.	73	73	12.80	3.60	2.00	104.4	64.4
d.	73	73	12.80	3.60	2.37	104.4	73.7
e.	73	73	12.80	3.60	2.78	104.4	83.9
f.	73	73	12.80	3.53	3.22	102.7	94.9
g.	74	74	12.80	3.64	3.64	105.4	105.4
A4a.*	73	73	12.80	3.60	3.17	104.4	93.7
b.	73	73	12.80	3.54	3.54	102.9	102.9
c.							

II. PERFORMANCE DATA

Data Set	Turbine Inlet Pressure		Flow Rate		Load Cell		
	Friction Factor	Corrected Inlet Pressure (psia)	Volume (SCFM ₀)	Mass (lbm/sec)	Static Zero	Excitation Voltage (V DC)	Output Voltage (mV)
A1a.					.552	8.43	
b.*	.0233	28.8	1.6	.00483		8.43	.525
c.	.0223	33.7	1.9	.00572		8.44	.558
d.	.0209	43.0	2.5	.00755		8.44	.606
e.	.0195	51.5	3.2	.00988		8.44	.666
f.	.0185	62.2	3.9	.01219		8.42	.754
g.	.0180	69.1	4.4	.01386		8.44	.818
h.	.0173	80.0	5.1	.01597		8.44	.915
i.	.0169	93.5	5.7	.01779		8.44	.932
A2a.*	.0205	44.6	2.6	.00811		8.44	.563
b.	.0196	52.0	3.2	.00985		8.44	.609
c.	.0187	62.1	3.8	.01177		8.44	.685
d.	.0179	71.4	4.5	.01401		8.44	.737
e.	.0174	80.0	5.1	.01584		8.44	.812
f.	.0168	89.8	5.8	.01813		8.44	.887
g.					.541	8.44	
A3a.					.484	8.45	
b.*	.0188	59.1	3.7	.01146		8.45	.563
c.	.0187	62.0	3.8	.01183		8.45	.600
d.	.0181	71.0	4.3	.01347		8.45	.630, .660
e.	.0173	80.6	5.1	.01608		8.45	.730
f.	.0168	91.4	5.7	.01789		8.45	.760
g.	.0165	101.7	6.1	.01943		8.45	.790
A4a.*	.0169	90.1	5.6	.01772		8.45	.564
b.	.0165	89.0	6.2	.01954		8.45	.640
c.					.535	8.45	

II. PERFORMANCE DATA

Data Set	Gross Force (lbf)	Net Force (lbf)	Torque (ft-lbf)	Speed (rpm)	Turbine Performance		
					Measured Output (hp)	Theoretical Input (hp)	Efficiency (%)
A1a.							
b.*	.104	.0108	.005	10000	.0098	.158	6.2
c.	.110	.017	.008	10000	.0155	.225	6.9
d.	.120	.027	.013	10000	.0241	.371	6.5
e.	.132	.039	.019	10000	.0385	.611	6.3
f.	.149	.056	.027	10000	.0508	.758	6.7
g.	.162	.069	.033	10000	.0619	.910	6.8
h.	.181	.088	.042	10000	.0792	1.13	7.0
i.	.184	.091	.043	10000	.0822	1.35	6.1
A2a.*	.111	.021	.010	20000	.0766	.815	9.4
b.	.120	.030	.014	20000	.0548	.554	9.9
c.	.135	.045	.021	20000	.0819	.731	11.2
d.	.146	.056	.027	20000	.1003	.946	10.6
e.	.160	.070	.033	20000	.1270	1.11	11.4
f.	.175	.085	.040	20000	.1540	1.34	11.5
g.							
A3a.							
b.*	.111	.002	.001	30000	.0054	.692	.78
c.	.118	.010	.005	30000	.0279	.734	3.8
d.	.124	.016	.008	30000	.0442	.902	4.9
e.	.130	.022	.010	30000	.0602	.899	6.7
f.	.144	.036	.017	30000	.0976	1.13	8.6
g.	.150	.042	.020	30000	.1136	1.34	8.5
h.	.156	.048	.023	30000	.1296	1.51	8.6
A4a.*	.011	.006	.003	40000	.0224	1.32	1.7
b.	.126	.021	.010	40000	.0766	1.45	5.3
c.							

II. PERFORMANCE DATA

Set	Air Temperature		Pressure				
	Rotameter Inlet (°F)	Turbine Inlet (°F)	Transducer Excitation Voltage (V DC)	Output Voltages		Rotameter Inlet (psia)	Turbine Inlet (psia)
				Rotameter Inlet (V)	Turbine Inlet (V)		
A5a.							
b.*	73	73	12.50	3.97	.532	113.7	27.7
c.	73	73	12.50	3.88	.790	111.4	34.2
d.	73	73	12.50	3.59	1.18	104.2	43.9
e.	73	73	12.50	3.57	1.61	103.7	54.7
f.	73	73	12.50	3.62	2.02	104.9	64.9
g.	73	73	12.50	3.56	2.42	103.4	74.9
h.	73	73	12.50	3.53	2.77	102.7	83.7
i.	74	74	12.50	3.45	3.14	100.7	92.9
A6a.							
b.*	73	73	12.50	3.86	1.09	110.9	41.7
c.	73	73	12.50	3.72	1.65	107.4	55.7
d.	73	73	12.50	3.73	2.05	107.7	65.7
e.	74	74	12.50	3.55	2.36	103.2	73.4
f.	74	74	12.50	3.70	2.90	106.9	86.9
g.	73	73	12.50	3.50	3.20	101.9	94.4
h.	74	74	12.50	3.53	3.52	102.7	102.4

II. PERFORMANCE DATA

Data Set	Turbine Inlet Pressure		Flow Rate		Load Cell		
	Friction Factor	Corrected Inlet Pressure (psia)	Volume (SCFM ₀)	Mass (lbm/sec)	Static Zero	Excitation Voltage (V DC)	Output Voltage (mV)
A5a.					.568	8.47	
b.*	.0242	26.7	1.4	.00416		8.47	.535
c.	.0227	33.2	1.8	.00547		8.47	.565
d.	.0209	42.4	2.5	.00757		8.47	.611
e.	.0196	52.7	3.2	.00983		8.47	.674
f.	.0187	62.6	3.8	.01186		8.47	.750
g.	.0180	72.2	4.4	.01373		8.47	.805
h.	.0172	80.3	5.2	.01627		8.47	.880
i.	.0170	89.5	5.6	.01738		8.47	.900
A6a.					.568	8.47	
b.*	.0212	40.3	2.3	.00714		8.47	.555
c.	.0195	53.7	3.2	.01001		8.47	.610
d.	.0185	63.1	3.9	.01235		8.47	.670
e.	.0181	72.8	4.3	.01337		8.47	.715
f.	.0173	83.7	5.1	.01626		8.47	.790
g.	.0169	90.9	5.7	.01783		8.47	.825
h.	.0166	98.7	6.1	.01917		8.47	.865

II. PERFORMANCE DATA

Data Set	Gross Force (lbf)	Net Force (lbf)	Torque (ft-lbf)	Speed (rpm)	Turbine Performance		
					Measured Output (hp)	Theoretical Input (hp)	Efficiency (%)
A5a.							
b.*	.105	.012	.006	10000	.0108	.123	8.8
c.	.111	.018	.009	10000	.0163	.212	7.7
d.	.120	.027	.013	10000	.0244	.364	6.7
e.	.133	.040	.019	10000	.0361	.555	6.5
f.	.148	.055	.026	10000	.0497	.742	6.7
g.	.158	.065	.031	10000	.0587	.917	6.4
h.	.173	.080	.038	10000	.0723	1.15	6.3
i.	.177	.084	.040	10000	.0759	1.29	5.9
A6a.							
b.*	.109	.019	.009	20000	.0343	.033	10.3
c.	.120	.030	.014	20000	.0542	.571	9.5
d.	.132	.042	.020	20000	.0759	.774	9.8
e.	.141	.051	.024	20000	.0922	.904	10.2
f.	.155	.065	.031	20000	.1175	1.18	10.0
g.	.162	.072	.034	20000	.1301	1.33	9.8
h.	.170	.080	.038	20000	.1446	1.48	9.8

II. PERFORMANCE DATA

Data Set	Air Temperature		Pressure				
	Rotameter Inlet (°F)	Turbine Inlet (°F)	Output Voltages				
			Transducer Excitation Voltage (V DC)	Rotameter Inlet (V)	Turbine Inlet (V)	Rotameter Inlet (psia)	Turbine Inlet (psia)
B1a.							
b.*	73	73	12.69	4.00	.270	114.4	21.1
c.*	72	72	12.69	4.06	.247	115.9	20.5
d.	73	72	12.69	3.90	.418	111.9	24.8
e.	72	72	12.69	3.97	.414	113.6	24.7
f.	72	72	12.69	3.86	.810	110.9	34.6
g.	72	72	12.69	3.73	.777	107.6	33.8
h.	72	72	12.69	3.57	1.18	103.6	43.9
i.	72	72	12.69	3.60	1.19	104.4	44.2
j.	72	72	12.69	3.60	1.17	104.4	43.6
k.	72	72	12.69	3.55	1.60	103.1	54.3
l.	72	72	12.69	3.55	1.97	103.1	63.6
m.	72	72	12.69	3.54	1.97	102.9	63.5
n.	72	72	12.69	3.49	2.36	101.6	73.3
o.	73	73	12.69	3.50	2.37	101.9	73.6
p.	73	73	12.69	3.60	2.90	104.4	86.9
q.	73	73	12.69	3.60	2.94	104.4	87.9
r.	73	73	12.69	3.60	3.27	104.4	96.0
s.	73	73	12.69	3.59	3.28	104.1	94.8
B2a.*	73	73	12.69	3.92	.889	112.4	36.6
b.*	73	73	12.69	3.91	.865	112.1	36.0
c.	72	72	12.69	3.76	1.22	108.4	44.8
d.	72	72	12.69	3.73	1.21	107.6	44.6
e.	72	72	12.69	3.58	1.60	103.9	54.2
f.	73	73	12.69	3.53	1.58	102.6	53.8
g.	73	73	12.69	3.59	2.01	104.1	64.6
h.	73	73	12.68	3.64	2.04	105.4	65.4
i.	73	73	12.68	3.51	2.35	102.1	73.1
j.	73	73	12.68	3.53	2.37	102.6	73.5
k.	73	73	12.68	3.48	2.80	101.4	84.2
l.	73	73	12.68	3.51	2.83	102.1	85.0
m.	73	73	12.68	3.52	3.14	102.4	92.8
n.	73	73	12.68	3.44	3.14	100.4	92.8

II. PERFORMANCE DATA

Data Set	Turbine Inlet Pressure		Flow Rate		Load Cell		
	Friction Factor	Corrected Inlet Pressure (psia)	Volume (SCFM ₀)	Mass (lbm/sec)	Static Zero	Excitation Voltage (V DC)	Output Voltage (mV)
B1a.					.429	8.46	
b.*	.0277	20.7	0.9	.0025		8.46	.460
c.*	.0276	20.1	0.9	.0025		8.46	.468
d.	.0248	24.1	1.3	.0038		8.46	.485
e.	.0248	24.0	1.3	.0038		8.46	.485
f.	.0223	33.5	1.9	.0058		8.46	.550
g.	.0224	32.6	1.9	.0057		8.46	.545
h.	.0209	42.4	2.5	.0075		8.46	.640
i.	.0207	42.5	2.6	.0079		8.46	.605
j.	.0194	41.0	3.3	.0102		8.46	.655
k.	.0196	52.3	3.2	.0098		8.46	.685
l.	.0187	61.2	3.8	.0117		8.46	.785
m.	.0187	61.1	3.8	.0117		8.46	.770
n.	.0179	70.5	4.5	.0139		8.46	.880
o.	.0180	70.9	4.4	.0136		8.46	.880
p.	.0173	83.7	5.1	.0160		8.46	.938
q.	.0173	84.7	5.1	.0160		8.46	.935
r.	.0169	92.5	5.6	.0177		8.46	1.030
s.	.0169	91.3	5.6	.0176		8.46	1.045
B2a.*	.0220	35.4	2.0	.0061		8.46	.485
b.*	.0220	34.7	2.0	.0061		8.46	.494
c.	.0208	43.2	2.5	.0077		8.46	.520
d.	.0208	43.0	2.5	.0077		8.46	.540
e.	.0196	52.2	3.2	.0098		8.46	.580
f.	.0194	51.7	3.3	.0101		8.46	.590
g.	.0187	62.2	3.8	.0118		8.46	.680
h.	.0187	63.0	3.8	.0119		8.46	.690
i.	.0179	70.2	4.5	.0139		8.46	.720
j.	.0179	70.6	4.5	.0140		8.46	.745
k.	.0173	80.9	5.2	.0161		8.46	.810
l.	.0173	81.7	5.2	.0162		8.46	.845
m.	.0168	89.1	5.8	.0181		8.46	.935
n.	.0168	89.1	5.8	.0180		8.46	.930

II. PERFORMANCE DATA

Data Set	Gross Force (lbf)	Net Force (lbf)	Torque (ft-lbf)	Speed (rpm)	Turbine Performance		
					Measured Output (hp)	Theoretical Input (hp)	Efficiency (%)
B1a.							
b.*	.091	.006	.003	10000	.0054	.045	12.0
c.*	.092	.007	.003	10000	.0063	.041	15.2
d.	.096	.011	.005	10000	.0099	.0943	10.5
e.	.096	.011	.005	10000	.0099	.0934	10.6
f.	.108	.023	.011	10000	.0208	.226	9.2
g.	.107	.022	.010	10000	.0199	.216	9.2
h.	.126	.041	.019	10000	.0370	.359	10.3
i.	.119	.034	.016	10000	.0307	.379	8.1
j.	.129	.044	.021	10000	.0398	.480	8.3
k.	.135	.050	.024	10000	.0452	.545	8.3
l.	.155	.070	.033	10000	.0633	.719	8.8
m.	.152	.067	.032	10000	.0605	.720	8.4
n.	.173	.088	.042	10000	.0795	.914	8.7
o.	.173	.088	.042	10000	.0795	.903	8.8
p.	.185	.100	.047	10000	.0904	1.14	7.9
q.	.184	.099	.047	10000	.0895	1.15	7.8
r.	.203	.118	.056	10000	.1066	1.32	8.1
s.	.206	.121	.057	10000	.1093	1.30	8.4
B2a.*	.096	.011	.005	20000	.0199	.252	7.9
b.*	.097	.012	.006	20000	.0217	.247	8.8
c.	.102	.017	.008	20000	.0307	.374	8.2
d.	.106	.021	.010	20000	.0380	.376	10.1
e.	.114	.029	.014	20000	.0524	.546	9.6
f.	.116	.031	.015	20000	.0560	.560	10.0
g.	.134	.049	.023	20000	.0886	.732	12.1
h.	.136	.051	.024	20000	.0922	.744	12.4
i.	.142	.057	.027	20000	.1030	.920	11.2
j.	.147	.062	.029	20000	.1120	.926	12.1
k.	.160	.075	.036	20000	.1355	1.14	11.9
l.	.166	.081	.038	20000	.1464	1.15	12.7
m.	.184	.099	.047	20000	.1789	1.34	13.4
n.	.183	.098	.047	20000	.1771	1.32	13.4

II. PERFORMANCE DATA

Data Set	Air Temperature		Pressure				
	Rotameter Inlet (°F)	Turbine Inlet (°F)	Output Voltages		Turbine Inlet (V)	Rotameter Inlet (psia)	Turbine Inlet (psia)
			Transducer Excitation Voltage (V DC)	Rotameter Inlet (V)			
B3a.*	73	73	12.68	3.81	1.74	109.6	57.7
b.*	73	73	12.68	3.80	1.72	109.4	57.4
c.	73	73	12.68	3.60	2.00	104.4	64.4
d.	73	73	12.68	3.68	2.05	106.4	65.5
e.	73	73	12.68	3.62	2.43	104.9	75.1
f.	73	73	12.68	3.59	2.42	104.1	74.8
g.	73	73	12.68	3.55	2.79	103.1	84.1
h.	73	73	12.68	3.53	2.76	102.6	83.4
i.	73	73	12.68	3.55	3.21	103.1	94.5
j.	73	73	12.68	3.65	3.31	105.6	97.0
B4a.*	73	73	12.68	3.67	2.78	106.1	83.8
b.*	73	73	12.68	3.69	2.78	106.6	83.9
c.	73	73	12.68	2.51	3.15	102.1	93.2
d.	73	73	12.68	3.62	3.19	104.9	94.1
e.**	73	73	12.68	3.57	3.54	103.6	102.9
f.**	73	73	12.68	3.66	3.62	105.9	104.9
g.							

II. PERFORMANCE DATA

Data Set	Turbine Inlet Pressure		Flow Rate		Load Cell		
	Friction Factor	Corrected Inlet Pressure (psia)	Volume (SCFM ₀)	Mass (lbm/sec)	Static Zero	Excitation Voltage (V DC)	Output Voltage (mV)
B3a.*	.0193	55.6	3.3	.0104		8.46	.565
b.*	.0193	55.3	3.3	.0104		8.46	.560
c.	.0187	62.0	3.8	.0118		8.46	.610
d.	.0186	63.1	3.8	.0119		8.46	.610
e.	.0180	72.4	4.4	.0138		8.46	.725
f.	.0180	72.0	4.4	.0137		8.46	.710
g.	.0173	80.9	5.1	.0159		8.46	.725
h.	.0173	80.2	5.1	.0159		8.46	.740
i.	.0169	91.0	5.6	.0176		8.46	.790
j.	.0169	93.5	5.6	.0178		8.46	.810
B4a.*	.0175	80.8	4.9	.0155		8.46	.610
b.*	.0174	80.8	4.9	.0155		8.46	.605
c.	.0168	89.4	5.8	.0181		8.46	.690
d.	.0168	90.4	5.7	.0180		8.46	.670
e.**			6.2	.0195		8.46	.625
f.**			6.2	.0195		8.46	.640
g.					.464	8.46	

II. PERFORMANCE DATA

Data Set	Gross Force (lbf)	Net Force (lbf)	Torque (ft-lbf)	Speed (rpm)	Turbine Performance		
					Measured Output (hp)	Theoretical Input (hp)	Efficiency (%)
B3a.*	.111	.022	.010	30000	.0596	.602	9.9
b.*	.110	.021	.010	30000	.0569	.605	9.4
c.	.120	.031	.015	30000	.0840	.730	11.5
d.	.120	.031	.015	30000	.0840	.743	11.3
e.	.143	.054	.026	30000	.1464	.927	15.8
f.	.140	.051	.024	30000	.1383	.916	15.1
g.	.143	.054	.026	30000	.1464	1.13	13.0
h.	.146	.057	.027	30000	.1545	1.12	13.8
i.	.156	.067	.032	30000	.1816	1.31	13.9
j.	.160	.071	.034	30000	.1925	1.34	14.4
B4a.*	.120	.029	.014	40000	.1048	1.09	9.6
b.*	.119	.028	.013	40000	.1012	1.09	9.3
c.	.136	.045	.021	40000	.1627	1.33	12.2
d.	.132	.041	.019	40000	.1428	1.29	11.1
e.**	.123			48000			
f.**	.126			45000			
g.							

II. PERFORMANCE DATA

Data Set	Air Temperature		Pressure				
	Rotameter Inlet (°F)	Turbine Inlet (°F)	Output Voltages		Turbine Inlet (V)	Rotameter Inlet (psia)	Turbine Inlet (psia)
			Transducer Excitation Voltage (V DC)	Rotameter Inlet (V)			
C1a.							
b.*	74	74	12.58	4.05	.281	115.6	21.4
c.*	73	73	12.58	4.00	.274	114.3	21.2
d.	73	73	12.58	3.86	.821	110.8	34.9
e.	73	73	12.58	3.73	.802	107.6	34.4
f.	73	73	12.58	3.60	1.19	104.3	44.1
g.	73	73	12.58	3.60	1.19	104.3	44.1
h.	73	73	12.58	3.59	1.56	104.1	53.3
i.	73	73	12.58	3.59	1.57	104.1	53.6
j.	73	73	12.58	3.63	2.04	105.1	65.3
k.	73	73	12.58	3.50	1.99	101.8	64.1
l.	73	73	12.58	3.53	2.41	102.6	74.6
m.	73	73	12.58	3.69	2.51	106.6	77.1
n.	73	73	12.58	3.66	2.90	105.8	86.8
o.	73	73	12.58	3.46	2.78	100.8	83.8
p.	73	73	12.58	3.60	3.25	104.3	95.6
q.	73	73	12.58	3.57	3.19	103.6	94.1
C2a.*	73	73	12.58	3.99	.807	114.1	34.5
b.*	73	73	12.58	3.92	.792	112.3	34.1
c.	73	73	12.58	3.74	1.18	107.8	43.8
d.	73	73	12.58	3.76	1.19	108.3	44.1
e.	73	73	12.58	3.62	1.60	104.8	54.3
f.	73	73	12.58	3.67	1.62	106.1	54.8
g.	73	73	12.58	3.57	2.01	103.6	64.6
h.	73	73	12.58	3.60	2.03	104.3	65.1
i.	73	73	12.58	3.53	2.39	102.6	74.1
j.	73	73	12.58	3.57	2.42	103.6	74.8
k.	73	73	12.58	3.54	2.82	102.8	84.8
l.	73	73	12.58	3.55	2.82	103.1	84.8
m.	73	73	12.58	3.66	3.21	105.8	94.6
n.	73	73	12.58	3.45	3.07	100.6	91.1
o.							

II. PERFORMANCE DATA

Data Set	Turbine Inlet Pressure		Flow Rate		Load Cell		
	Friction Factor	Corrected Inlet Pressure (psia)	Volume (SCFM ₀)	Mass (lbm/sec)	Static Zero	Excitation Voltage (V DC)	Output Voltage (mV)
C1a.					.513	8.47	
b.*	.0277	21.0	.9	.00246		8.47	.473
c.*	.0277	20.8	.9	.00245		8.47	.475
d.	.0224	33.8	1.9	.00577		8.47	.505
e.	.0224	33.3	1.9	.00569		8.47	.490
f.	.0209	42.6	2.5	.00755		8.47	.495
g.	.0209	42.6	2.5	.00755		8.47	.495
h.	.0196	51.3	3.2	.00983		8.47	.505
i.	.0197	51.7	3.1	.00950		8.47	.505
j.	.0186	62.8	3.9	.0122		8.47	.525
k.	.0186	61.6	3.9	.0120		8.47	.520
l.	.0180	71.9	4.4	.0136		8.47	.545
m.	.0179	74.4	4.4	.0139		8.47	.540
n.	.0174	83.7	5.0	.0158		8.47	.555
o.	.0174	80.6	5.1	.0158		8.47	.565
p.	.0169	92.1	5.6	.0177		8.47	.580
q.	.1069	90.6	5.6	.0176		8.47	.590
C2a.*	.0226	33.4	1.8	.00552		8.47	.465
b.*	.0227	33.1	1.8	.00547		8.47	.465
c.	.0208	42.2	2.5	.00768		8.47	.480
d.	.0208	42.6	2.5	.00770		8.47	.490
e.	.0196	52.3	3.2	.00986		8.47	.500
f.	.0195	52.8	3.2	.00992		8.47	.490
g.	.0187	62.2	3.8	.0118		8.47	.495
h.	.0187	62.7	3.8	.0118		8.47	.495
i.	.0179	71.2	4.5	.0140		8.47	.495
j.	.0180	72.1	4.4	.0137		8.47	.495
k.	.0174	81.7	5.0	.0156		8.47	.500
l.	.0174	81.7	5.0	.0156		8.47	.500
m.	.0170	91.2	5.5	.0175		8.47	.510
n.	.0170	87.6	5.6	.0173		8.47	.505
o.					.539	8.47	

II. PERFORMANCE DATA

Data Set	Gross Force (lbf)	Net Force (lbf)	Torque (ft-lbf)	Speed (rpm)	Turbine Performance		
					Measured Output (hp)	Theoretical Input (hp)	Efficiency (%)
C1a.							
b.*	.0931	.008	.004	10000	.0073	.00115	15.8
c.*	.0935	.009	.004	10000	.0077	.00132	17.1
d.	.0994	.014	.007	10000	.0130	.228	5.7
e.	.0964	.014	.007	10000	.0103	.219	4.7
f.	.0974	.012	.006	10000	.0112	.365	3.1
g.	.0974	.012	.006	10000	.0112	.365	3.1
h.	.0994	.014	.007	10000	.0130	.542	2.4
i.	.0994	.014	.007	10000	.0130	.529	2.5
j.	.103	.018	.009	10000	.0163	.758	2.2
k.	.102	.017	.008	10000	.0154	.738	2.1
l.	.107	.022	.010	10000	.0199	.910	2.2
m.	.106	.021	.010	10000	.0190	.949	2.0
n.	.109	.024	.011	10000	.0190	1.12	1.7
o.	.111	.026	.012	10000	.0235	1.12	2.1
p.	.114	.029	.014	10000	.0262	1.32	2.0
q.	.116	.031	.015	10000	.0280	1.33	2.1
C2a.*	.0915	.007	.003	20000	.0118	.215	5.5
b.*	.0915	.007	.003	20000	.0118	.210	5.6
c.	.0945	.010	.005	20000	.0172	.370	4.7
d.	.0964	.011	.005	20000	.0206	.375	5.5
e.	.0984	.013	.006	20000	.0242	.550	4.4
f.	.0964	.011	.005	20000	.0206	.558	3.7
g.	.0974	.012	.006	20000	.0224	.723	3.1
h.	.0974	.012	.006	20000	.0224	.735	3.1
i.	.0974	.012	.006	20000	.0224	.934	2.4
j.	.0974	.012	.006	20000	.0224	.915	2.5
k.	.0984	.013	.006	20000	.0242	1.10	2.2
l.	.0984	.013	.006	20000	.0242	1.10	2.2
m.	.100	.015	.007	20000	.0271	1.30	2.1
n.	.099	.014	.007	20000	.0271	1.29	2.1

II. PERFORMANCE DATA

Data Set	Air Temperature		Pressure				
	Rotameter Inlet (°F)	Turbine Inlet (°F)	Gage Zero (psig)	Relative Pressure		Absolute Pressure	
				Rotameter Inlet (psig)	Turbine Inlet (psig)	Rotameter Inlet (psia)	Turbine Inlet (psia)
E1a.			-3.6				
b.*	73	73		99.2	4.1	117.3	22.2
c.*	73	73		97.7	4.1	115.8	22.2
d.	73	73		96.4	6.6	114.5	24.7
e.	73	73		96.5	6.7	114.6	24.8
f.	73	73		89.7	16.3	107.8	34.4
g.	73	73		88.0	15.8	106.1	33.9
h.	73	73		86.7	26.0	104.8	44.1
i.	73	73		87.8	26.5	105.9	44.6
j.	73	73		86.5	35.6	104.6	53.7
k.	73	73		86.5	35.8	104.6	53.9
l.	73	73		90.0	47.9	108.1	66.0
m.	73	73		85.1	46.8	103.2	64.9
n.	73	73		83.8	56.4	101.9	74.5
o.	73	73		88.0	56.2	106.1	74.3

II. PERFORMANCE DATA

Data Set	Turbine Inlet Pressure		Flow Rate		Load Cell		
	Friction Factor	Corrected Inlet Pressure (psia)	Volume (SCFM ₀)	Mass (lbm/sec)	Static Zero	Excitation Voltage (V DC)	Output Voltage (mV)
E1a.					.610	8.47	
b.*	.0247	21.3	1.3	.00386		8.47	.620
c.*	.0247	21.4	1.3	.00384		8.47	.610
d.	.0234	23.5	1.6	.00484		8.47	.635
e.	.0234	23.6	1.6	.00484		8.47	.640
f.	.0208	32.4	2.5	.00769		8.47	.730
g.	.0208	31.9	2.5	.00763		8.47	.730
h.	.0192	41.3	3.4	.01052		8.47	.850
i.	.0192	41.8	3.4	.01058		8.47	.850
j.	.0181	50.0	4.3	.01345		8.47	.980
k.	.0182	50.4	4.2	.01312		8.47	.980
l.	.0172	61.7	5.1	.01633		8.47	1.140
m.	.0173	60.7	5.1	.01596		8.47	1.120
n.	.0166	69.5	6.0	.01876		8.47	1.230
o.	.0166	69.3	5.9	.01881		8.47	1.240

II. PERFORMANCE DATA

Data Set	Gross Force (lbf)	Net Force (lbf)	Torque (ft-lbf)	Speed (rpm)	Turbine Performance		
					Measured Output (hp)	Theoretical Input (hp)	Efficiency (%)
E1a.							
b.*	.122	.007	.003	10000	.006	.070	8.6
c.*	.120	.005	.002	10000	.005	.081	6.2
d.	.125	.010	.005	10000	.009	.114	7.9
e.	.126	.011	.005	10000	.010	.116	8.6
f.	.144	.029	.014	10000	.026	.286	9.1
g.	.144	.029	.014	10000	.026	.280	9.3
h.	.167	.052	.025	10000	.047	.495	9.5
i.	.167	.052	.025	10000	.047	.500	9.4
j.	.193	.078	.037	10000	.070	.722	9.7
k.	.193	.078	.037	10000	.070	.707	9.9
l.	.224	.109	.052	10000	.098	1.00	9.8
m.	.220	.105	.050	10000	.095	.969	9.8
n.	.242	.127	.060	10000	.115	1.22	9.4
o.	.244	.129	.061	10000	.117	1.23	9.5

II. PERFORMANCE DATA

Data Set	Air Temperature		Gage Zero (psig)	Pressure			
	Rotameter Inlet (°F)	Turbine Inlet (°F)		Relative Pressure		Absolute Pressure	
				Rotameter Inlet (psig)	Turbine Inlet (psig)	Rotameter Inlet (psia)	Turbine Inlet (psia)
E2a.			-3.9				
b.*	72	72		97.4	11.0	115.8	29.4
c.*	72	72		97.4	10.8	115.8	29.2
d.	72	72		90.7	16.3	109.1	34.7
e.	72	72		89.3	16.0	107.7	34.4
f.	72	72		88.3	26.2	106.7	44.6
g.	72	72		88.1	26.2	106.5	44.6
E3a.			-3.2				
b.	72	72		91.0	34.7	108.7	52.4
c.	72	72		91.0	35.2	108.7	52.9
d.	72	72		85.0	46.6	102.7	64.3
e.	72	72		88.8	46.5	106.5	64.2
f.	72	72		88.0	53.8	105.7	71.5
g.	72	72		89.7	56.0	107.4	73.7
h.	72	72		83.2	70.6	100.9	88.3
i.	73	73		85.6	66.9	103.3	84.6
j.	73	73		86.2	76.1	103.9	93.8
k.	73	73		83.0	77.0	100.7	94.7
E4a.*	73	73		91.7	26.6	109.4	
b.*	73	73		93.8	27.0	111.5	
c.	73	73		88.9	37.5	106.6	55.2
d.	73	73		88.1	37.1	105.8	54.8
e.	73	73		87.3	49.0	105.0	66.7
f.	73	73		89.3	51.0	107.0	68.7
g.	73	73		85.0	58.2	102.7	75.9
h.	73	73		85.8	58.1	103.5	75.8
i.	73	73		84.5	68.8	102.2	86.5
j.	73	73		85.3	70.9	103.0	88.6
k.	73	73		84.2	78.0	101.9	95.7
l.	73	73		88.8	79.7	106.5	97.4

II. PERFORMANCE DATA

Data Set	Turbine Inlet Pressure		Flow Rate		Load Cell		
	Friction Factor	Corrected Inlet Pressure (psia)	Volume (SCFM ₀)	Mass (lbm/sec)	Static Zero	Excitation Voltage (V DC)	Outlet Voltage (mV)
E2a.					.605	8.46	
b.*	.0220	27.8	2.0	.0062		8.46	.625
c.*	.0220	27.6	2.0	.0062		8.46	.625
d.	.0210	32.7	2.5	.0077		8.47	.670
e.	.0210	32.4	2.5	.0077		8.47	.670
f.	.0192	41.8	3.4	.0106		8.47	.780
g.	.0192	41.8	3.4	.0106		8.47	.775
E3a.					.699	8.47	
b.	.0181	48.7	4.2	.0134		8.47	.890
c.	.0181	49.2	4.2	.0134		8.47	.885
d.	.0172	59.7	5.3	.0166		8.47	1.020
e.	.0172	59.6	5.2	.0165		8.47	1.040
f.	.0167	66.4	5.8	.0184		8.47	1.150
g.	.0167	68.7	5.8	.0186		8.47	1.160
h.	.0161	82.8	6.9	.0215		8.47	1.360
i.	.0159	78.4	7.1	.0224		8.47	1.380
j.	.0156	87.2	7.7	.0244		8.47	1.470
k.	.0157	88.7	7.5	.0234		8.47	1.440
E4a.*	.0191	41.4	3.4	.0107		8.47	.760
b.*	.0191	41.7	3.4	.0108		8.47	.760
c.	.0181	51.5	4.3	.0136		8.47	.870
d.	.0181	51.1	4.3	.0135		8.47	.875
e.	.0171	62.2	5.3	.0167		8.47	.995
f.	.0171	64.2	5.3	.0169		8.47	1.01
g.	.0166	70.8	6.1	.0191		8.47	1.07
h.	.0165	70.6	6.1	.0192		8.47	1.07
i.	.0160	80.6	7.0	.0220		8.47	1.18
j.	.0160	82.8	7.0	.0221		8.47	1.19
k.	.0156	89.3	7.7	.0242		8.47	1.25
l.	.0155	90.9	7.7	.0248		8.47	1.28

II. PERFORMANCE DATA

Data Set	Gross Force (lbf)	Net Force (lbf)	Torque (ft-lbf)	Speed (rpm)	Turbine Performance		
					Measured Output (hp)	Theoretical Input (hp)	Efficiency (%)
E2a.							
b.*	.123	.005	.0024	20000	.009	.192	4.7
c.*	.123	.005	.0024	20000	.009	.188	4.8
d.	.132	.014	.0066	20000	.025	.287	8.7
e.	.132	.014	.0066	20000	.025	.281	8.9
f.	.154	.036	.017	20000	.065	.500	13.0
g.	.153	.035	.017	20000	.063	.500	12.6
E3a.							
b.	.174	.057	.0271	20000	.103	.706	14.6
c.	.174	.056	.0266	20000	.101	.711	14.2
d.	.201	.083	.0394	20000	.150	.993	15.1
e.	.205	.087	.0413	20000	.157	.994	15.8
f.	.226	.108	.0513	20000	.195	1.17	16.6
g.	.228	.110	.0522	20000	.199	1.21	16.5
h.	.268	.150	.0712	20000	.271	1.52	17.8
i.	.272	.154	.0731	20000	.278	1.55	17.9
j.	.289	.171	.0812	20000	.309	1.78	17.4
k.	.283	.165	.0783	20000	.298	1.71	17.4
E4a.*	.150	.022	.0104	30000	.060	.509	11.8
b.*	.150	.022	.0104	30000	.060	.517	11.6
c.	.171	.043	.0204	30000	.117	.750	15.6
d.	.172	.044	.0209	30000	.119	.739	16.1
e.	.196	.068	.0323	30000	.184	1.03	17.9
f.	.199	.071	.0337	30000	.193	1.06	18.2
g.	.211	.083	.0394	30000	.225	1.26	17.8
h.	.211	.083	.0394	30000	.225	1.26	17.8
i.	.232	.104	.0494	30000	.282	1.54	18.3
j.	.234	.106	.0503	30000	.287	1.57	18.3
k.	.246	.118	.0560	30000	.320	1.78	18.0
l.	.252	.124	.0589	30000	.336	1.83	18.4

II. PERFORMANCE DATA

Data Set	Air Temperature		Gage Zero (psig)	Pressure			
	Rotameter Inlet (°F)	Turbine Inlet (°F)		Relative Pressure		Absolute Pressure	
				Rotameter Inlet (psig)	Turbine Inlet (psig)	Rotameter Inlet (psia)	Turbine Inlet (psia)
E5a.*	73	73		89.6	50.4	107.3	68.1
b.*	73	73		88.2	50.7	105.9	68.4
c.	73	73		86.7	54.8	104.4	72.5
d.	73	73		86.8	54.9	104.5	72.6
e.	74	74		85.8	68.6	103.5	86.3
f.	74	74		87.2	70.4	104.9	88.1
g.	74	74		90.1	80.4	107.8	98.1
h.	74	74		85.0	78.7	102.7	96.4
E6a.			-3.7				
b.*	72	72		98.5	3.3	116.6	21.4
c.*	72	72		96.9	3.2	115.0	21.3
d.	72	72		87.6	15.5	105.7	33.6
e.	72	72		87.6	15.6	105.7	33.7
f.	72	72		87.3	25.1	105.4	43.2
g.	72	72		89.2	26.0	107.3	44.1
h.	72	72		89.3	37.2	107.4	55.3
i.	72	72		86.2	35.2	104.3	53.5
j.	72	72		89.5	47.7	107.6	65.8
k.	72	72		89.9	47.6	108.0	65.7
l.	72	72		84.7	54.9	102.8	73.0
m.	72	72		84.7	56.8	102.8	74.9
E7a.							
b.*	74	74		97.2	10.3	115.3	28.4
c.*	74	74		96.4	10.2	114.5	28.3
d.	73	73		94.7	16.1	112.8	34.2
e.	73	73		91.2	15.8	109.3	33.9
f.	73	73		90.7	15.7	108.8	33.8
g.	73	73		90.6	15.7	108.7	33.8
h.	73	73		88.5	26.3	106.6	44.4
i.	73	73		89.3	26.5	107.4	44.6
j.	73	73		90.1	37.6	108.2	55.7
k.	73	73		86.4	35.2	104.5	53.3
l.	73	73		86.9	46.0	105.0	64.1
m.	73	73		86.0	45.7	104.1	63.8

II. PERFORMANCE DATA

Data Set	Turbine Inlet Pressure		Flow Rate		Load Cell		
	Friction Factor	Corrected Inlet Pressure (psia)	Volume (SCFM ₀)	Mass (lbm/sec)	Static Zero	Excitation Voltage (V DC)	Output Voltage (mV)
E5a.*	.0170	63.4	5.4	.0172		8.47	.905
b.*	.0170	63.8	5.4	.0171		8.47	.910
c.	.0167	67.6	5.8	.0183		8.47	.955
d.	.0167	67.7	5.8	.0183		8.47	.960
e.	.0160	80.5	6.9	.0218		8.47	1.08
f.	.0160	82.4	6.9	.0219		8.47	1.10
g.	.0155	91.4	7.8	.0252		8.47	1.21
h.	.0156	89.9	7.8	.0246		8.47	1.19
i.					.564	8.47	
E6a.					.558	8.46	
b.*	.025	20.6	1.3	.0039		8.46	.565
c.*	.025	20.5	1.3	.0038		8.47	.565
d.	.021	31.7	2.5	.0076		8.47	.690
e.	.021	31.8	2.5	.0076		8.47	.690
f.	.019	40.4	3.4	.0106		8.47	.810
g.	.019	41.3	3.4	.0106		8.47	.820
h.	.018	51.9	4.2	.0133		8.47	.960
i.	.018	49.8	4.2	.0131		8.47	.950
j.	.017	61.4	5.2	.0166		8.47	1.12
k.	.017	61.2	5.2	.0167		8.47	1.12
l.	.017	67.9	6.0	.0188		8.47	1.22
m.	.017	69.9	6.0	.0188		8.47	1.23
E7a.					.627	8.45	
b.*	.0223	27.0	1.9	.006		8.45	.625
c.*	.0223	26.9	1.9	.006		8.45	.630
d.	.021	32.3	2.4	.0075		8.45	.685
e.	.021	32.1	2.4	.0074		8.45	.680
f.	.021	32.0	2.4	.0074		8.45	.680
g.	.021	32.0	2.4	.0074		8.45	.670
h.	.0192	41.6	3.4	.0106		8.45	.775
i.	.0192	41.8	3.4	.0106		8.45	.790
j.	.0181	52.3	4.2	.0133		8.45	.920
k.	.0182	49.8	4.2	.0131		8.45	.910
l.	.0174	60.0	5.0	.0158		8.45	1.02
m.	.0174	59.7	5.0	.0157		8.45	1.01

II. PERFORMANCE DATA

Data Set	Gross Force (lbf)	Net Force (lbf)	Torque (ft-lbf)	Speed (rpm)	Turbine Performance		
					Measured Power Output (hp)	Theoretical Input (hp)	Efficiency (%)
E5a.*	.178	.054	.026	40000	.195	1.07	18.2
b.*	.180	.055	.026	40000	.199	1.07	18.6
c.	.188	.064	.030	40000	.231	1.18	19.6
d.	.189	.065	.031	40000	.235	1.18	19.9
e.	.213	.089	.042	40000	.322	1.53	21.0
f.	.217	.093	.044	40000	.336	1.56	21.6
g.	.238	.114	.054	40000	.412	1.88	22.0
h.	.234	.110	.052	40000	.398	1.82	21.9
E6a.							
b.*	.111	-.004	-.002	10000	-.004	.0741	-5.4
c.*	.111	-.005	-.002	10000	-.004	.0727	-5.5
d.	.136	.021	.010	10000	.019	.275	6.9
e.	.136	.021	.010	10000	.019	.275	6.9
f.	.160	.045	.021	10000	.041	.488	8.4
g.	.161	.046	.022	10000	.042	.506	8.3
h.	.189	.074	.035	10000	.067	.736	9.1
i.	.187	.072	.034	10000	.065	.707	9.2
j.	.220	.105	.050	10000	.095	1.02	9.3
k.	.220	.105	.050	10000	.095	1.02	9.3
l.	.240	.125	.059	10000	.113	1.22	9.3
m.	.242	.127	.060	10000	.115	1.24	9.3
E7a.							
b.*	.123	.005	.002	20000	.009	.173	5.2
c.*	.124	.006	.003	20000	.011	.175	6.3
d.	.135	.017	.008	20000	.031	.282	11.0
e.	.134	.016	.008	20000	.029	.274	10.6
f.	.134	.016	.008	20000	.029	.271	10.7
g.	.132	.014	.007	20000	.025	.269	9.3
h.	.153	.035	.017	20000	.063	.500	12.6
i.	.156	.038	.018	20000	.069	.507	13.6
j.	.182	.064	.030	20000	.116	.744	15.6
k.	.180	.062	.029	20000	.112	.704	15.9
l.	.201	.083	.039	20000	.150	.955	15.7
m.	.199	.081	.038	20000	.146	.942	15.5

II. PERFORMANCE DATA

Data Set	Air Temperature		Gage Zero (psig)	Pressure			
	Rotameter Inlet (°F)	Turbine Inlet (°F)		Relative Pressure		Absolute Pressure	
				Rotameter Inlet (psig)	Turbine Inlet (psig)	Rotameter Inlet (psia)	Turbine Inlet (psia)
E7n.	73	73		85.6	55.6	103.7	73.7
o.	73	73		88.3	58.7	106.4	76.8
p.	73	73		88.0	68.0	106.1	86.1
q.	73	73		88.5	67.1	106.6	85.2
r.	73	73		87.0	77.0	105.1	95.1
s.	73	73		87.5	76.9	105.6	95.0
t.	73	73		82.7	81.0	100.8	99.1
u.	73	73		81.0	80.6	99.1	98.7
v.							
E8a.*	73	72		92.6	31.8	110.7	49.9
b.*	72	72		91.4	31.2	109.5	49.3
c.	72	72		87.5	34.5	105.6	52.6
d.	72	72		91.8	36.8	109.9	54.9
e.	72	72		78.4	45.0	96.5	63.1
f.	72	72		86.6	48.2	104.7	66.3
g.	72	72		86.2	57.0	104.3	75.1
h.	72	72		87.1	57.3	105.2	75.4
i.	72	72		84.9	65.8	103.0	83.9
j.	72	72		89.7	68.8	107.8	86.9
k.	73	73		88.0	78.3	106.1	96.4
l.	73	73		83.1	76.3	101.2	94.4
m.	73	73		87.6	84.5	105.7	102.6
n.	73	73		83.3	82.7	101.4	100.8
E9a.*	73	73		89.7	50.2	107.8	68.3
b.*	73	73		89.4	50.6	107.5	68.7
c.	73	73		86.5	55.5	104.6	73.6
d.	73	73		87.8	55.7	105.9	73.8
e.	73	73		86.5	67.2	104.6	85.3
f.	73	73		85.8	66.2	103.9	84.3
g.	73	73		85.5	75.4	103.6	93.5
h.	73	73		90.5	77.2	108.6	95.3
i.	73	73		87.1	86.8	105.2	104.8
j.	73	73		84.4	83.6	102.5	101.7

II. PERFORMANCE DATA

Data Set	Turbine Inlet Pressure		Flow Rate		Load Cell		
	Pressure Friction Factor	Corrected Inlet Pressure (psia)	Volume (SCFM ₀)	Mass (lbm/sec)	Static Zero	Excitation Voltage (V DC)	Output Voltage (mV)
E7n.	.0167	68.8	5.9	.0186		8.45	1.19
o.	.0167	72.1	5.8	.0185		8.45	1.18
p.	.0161	80.6	6.7	.0214		8.45	1.32
q.	.0161	79.6	6.7	.0215		8.45	1.34
r.	.0157	88.9	7.5	.0239		8.45	1.49
s.	.0156	88.6	7.6	.0243		8.45	1.48
t.	.0154	92.3	8.2	.0257		8.45	1.52
u.	.0155	92.1	8.1	.0252		8.45	1.53
v.					.581	8.45	
E8a.*	.0185	46.7	3.8	.0122		8.46	.785
b.*	.0186	46.1	3.8	.0121		8.46	.780
c.	.0183	49.2	4.1	.0129		8.46	.820
d.	.0181	51.4	4.2	.0135		8.46	.840
e.	.0174	58.9	5.2	.0157		8.46	.940
f.	.0170	61.7	5.4	.0171		8.46	.960
g.	.0167	70.3	5.9	.0187		8.46	1.06
h.	.0167	70.7	5.8	.0184		8.46	1.06
i.	.0160	78.1	6.9	.0218		8.46	1.15
j.	.0160	81.2	6.8	.0220		8.46	1.18
k.	.0156	90.2	7.5	.0241		8.46	1.28
l.	.0157	88.2	7.6	.0238		8.46	1.27
m.	.0152	95.6	8.3	.0266		8.46	1.35
n.	.0154	94.1	8.2	.0258		8.46	1.32
9a.*	.0171	63.9	5.3	.0170		8.46	.910
b.*	.0171	64.3	5.3	.0169		8.46	.910
c.	.0167	68.6	5.9	.0187		8.46	.960
d.	.0167	69.0	5.8	.0185		8.46	.970
e.	.0161	79.6	6.8	.0216		8.46	1.07
f.	.0161	78.6	6.8	.0215		8.46	1.07
g.	.0157	87.3	7.5	.0238		8.46	1.15
h.	.0156	88.9	7.5	.0243		8.46	1.16
i.	.0153	98.1	8.2	.0262		8.46	1.26
j.	.0153	94.8	8.3	.0262		8.46	1.24

II. PERFORMANCE DATA

Data Set	Gross Force (lbf)	Net Force (lbf)	Torque (ft-lbf)	Speed (rpm)	Turbine Performance		
					Power Output (hp)	Input (hp)	Efficiency (%)
E7n.	.235	.117	.056	20000	.211	1.21	17.5
o.	.233	.115	.055	20000	.208	1.23	16.9
p.	.260	.142	.067	20000	.257	1.50	17.1
q.	.264	.146	.069	20000	.264	1.50	17.6
r.	.294	.176	.084	20000	.318	1.76	18.1
s.	.292	.174	.083	20000	.315	1.78	17.7
t.	.300	.182	.086	20000	.329	1.91	17.2
u.	.302	.184	.087	20000	.333	1.87	17.8
v.							
F8a.*	.155	.027	.013	30000	.073	.624	11.7
b.*	.154	.026	.012	30000	.071	.623	11.4
c.	.162	.034	.016	30000	.092	.687	13.4
d.	.166	.038	.018	30000	.103	.741	13.9
e.	.185	.057	.027	30000	.155	.945	16.4
f.	.189	.061	.029	30000	.165	1.04	15.8
g.	.209	.081	.038	30000	.220	1.23	17.9
h.	.209	.081	.038	30000	.220	1.22	18.1
i.	.227	.099	.047	30000	.268	1.51	17.8
j.	.233	.105	.050	30000	.285	1.55	18.4
k.	.252	.124	.059	30000	.336	1.78	18.9
l.	.250	.122	.058	30000	.331	1.74	19.0
m.	.266	.138	.065	30000	.374	2.01	18.6
n.	.260	.132	.063	30000	.358	1.94	18.5
E9a.*	.179	.055	.026	40000	.200	1.07	18.7
b.*	.179	.055	.026	40000	.200	1.07	18.7
c.	.189	.065	.031	40000	.235	1.21	19.4
d.	.191	.067	.032	40000	.242	1.20	20.1
e.	.211	.087	.041	40000	.315	1.51	20.9
f.	.211	.087	.041	40000	.315	1.50	21.0
g.	.227	.103	.049	40000	.372	1.73	21.5
h.	.229	.105	.050	40000	.380	1.78	21.3
i.	.248	.124	.059	40000	.448	1.99	22.5
j.	.244	.120	.057	40000	.434	1.97	22.0

II. PERFORMANCE DATA

Data Set	Air Temperature		Pressure				
	Rotameter Inlet (°F)	Turbine Inlet (°F)	Gage Zero (psig)	Relative Pressure		Absolute Pressure	
				Rotameter Inlet (psig)	Turbine Inlet (psig)	Rotameter Inlet (psia)	Turbine Inlet (psia)
E10a.			-3.4				
b.*	73	73		99.3	.5	117.0	18.2
c.*	73	73		99.5	.4	117.2	18.2
d.*	73	73		100.3	.5	118.0	18.2
e.*	73	73		98.8	.4	116.5	18.1
f.	73	73		93.8	6.2	111.5	23.9
g.	73	73		93.8	6.1	111.5	23.8
h.	73	73		94.2	6.1	111.9	23.8
i.	72	72		93.2	5.6	110.9	23.3
j.	72	72		89.0	15.1	106.7	32.8
k.	72	72		88.4	15.0	106.1	32.7
l.	72	72		88.5	15.0	106.2	32.7
m.	73	73		89.4	15.2	107.1	32.9
n.	72	72		91.2	27.6	108.9	45.3
o.	73	73		88.1	26.6	105.8	44.3
p.	72	72		90.2	28.0	107.9	45.7
q.	72	72		88.5	26.9	106.2	44.6
r.	72	72		86.7	33.6	104.4	51.3
s.	72	72		86.1	34.2	103.8	51.9
t.	73	73		88.7	36.2	106.4	53.9
u.	72	72		86.9	34.4	104.6	52.1
v.	73	73		85.8	46.5	103.5	64.2
w.	72	72		88.2	49.5	105.9	67.2
x.	73	73		84.1	46.8	101.8	64.0
y.	73	73		86.2	49.3	103.9	67.0
z.	73	73		86.1	56.3	103.8	74.0
aa.	73	73		87.1	58.2	104.8	75.9
bb.	73	73		85.8	56.4	103.5	74.1
cc.	73	73		88.2	58.4	105.9	76.1
dd.	73	73		86.7	64.9	104.4	82.6
ee.	73	73		86.1	65.0	103.8	82.7
ff.	73	73		83.1	62.0	100.8	79.7
gg.	73	73		86.4	65.1	104.1	82.8

II. PERFORMANCE DATA

Data Set	Turbine Inlet Pressure		Flow Rate		Load Cell		
	Friction Factor	Turbine Pressure (psia)	Volume (SCFM ₀)	Mass (lbm/sec)	Static Zero	Excitation Voltage (V DC)	Output Voltage (mV)
E10a.					.692	8.46	
b.*	.028	17.8	0.9	.002		8.46	.635
c.*	.028	17.7	0.9	.002		8.46	.630
d.*	.028	17.8	0.9	.0025		8.47	.635
e.*	.028	17.7	0.9	.0025		8.47	.630
f.	.024	22.9	1.55	.0046		8.47	.690
g.	.024	22.8	1.5	.0044		8.47	.680
h.	.0236	22.8	1.55	.0046		8.47	.690
i.	.0239	22.3	1.5	.0044		8.47	.690
j.	.0211	31.0	2.4	.0073		8.47	.785
k.	.0211	30.9	2.4	.0073		8.47	.785
l.	.0211	30.9	2.4	.0073		8.47	.790
m.	.0211	31.1	2.4	.0073		8.47	.795
n.	.0191	42.5	3.4	.0107		8.47	.960
o.	.0192	41.5	3.4	.0106		8.47	.940
p.	.0192	43.0	3.4	.0107		8.47	.950
q.	.0192	41.8	3.4	.0106		8.47	.950
r.	.0182	47.7	4.2	.0131		8.47	1.02
s.	.0183	48.5	4.1	.0128		8.47	1.01
t.	.0183	50.6	4.1	.0129		8.47	1.00
u.	.0184	48.9	4.0	.0125		8.47	1.02
v.	.0172	59.8	5.2	.0163		8.47	1.17
w.	.0173	63.1	5.1	.0162		8.47	1.19
x.	.0172	59.5	5.3	.0165		8.47	1.19
y.	.0173	62.9	5.1	.0160		8.47	1.22
z.	.0165	68.8	6.1	.0192		8.47	1.35
aa.	.0165	70.6	6.2	.0197		8.47	1.37
bb.	.0165	68.9	6.1	.0192		8.47	1.36
cc.	.0166	71.1	6.0	.0191		8.47	1.34
dd.	.0161	76.9	6.7	.0213		8.47	1.45
ee.	.0161	77.1	6.7	.0212		8.47	1.46
ff.	.0163	74.2	6.6	.0206		8.47	1.42
gg.	.0161	77.2	6.7	.0212		8.47	1.47

II. PERFORMANCE DATA

Data Set	Gross Force (lbf)	Net Force (lbf)	Torque (ft-lbf)	Speed (rpm)	Turbine Performance		
					Power Output (hp)	Input (hp)	Efficiency (%)
E10a.							
b.*	.125	.010	.005	10000	.009	.026	34.0
c.*	.124	.009	.004	10000	.008	.025	31.4
d.*	.125	.010	.005	10000	.009	.027	33.5
e.*	.124	.009	.004	10000	.008	.026	31.1
f.	.136	.021	.010	10000	.019	.104	18.2
g.	.134	.019	.009	10000	.017	.099	17.1
h.	.136	.021	.010	10000	.019	.103	18.4
i.	.136	.021	.010	10000	.019	.095	20.0
j.	.155	.039	.019	10000	.035	.259	13.5
k.	.155	.039	.019	10000	.035	.257	13.6
l.	.155	.040	.019	10000	.036	.259	13.9
m.	.156	.041	.019	10000	.037	.262	14.1
n.	.189	.074	.035	10000	.067	.519	12.9
o.	.185	.070	.033	10000	.063	.500	12.6
p.	.187	.072	.034	10000	.065	.520	12.5
q.	.187	.072	.034	10000	.065	.504	12.9
r.	.201	.086	.041	10000	.078	.690	11.3
s.	.199	.084	.040	10000	.076	.679	11.2
t.	.197	.082	.039	10000	.074	.705	10.5
u.	.201	.086	.041	10000	.078	.667	11.7
v.	.230	.115	.055	10000	.104	.990	10.5
w.	.234	.119	.056	10000	.108	1.01	10.7
x.	.234	.119	.056	10000	.108	1.00	10.8
y.	.240	.125	.059	10000	.113	1.00	11.3
z.	.266	.151	.072	10000	.136	1.25	10.9
aa.	.270	.155	.074	10000	.140	.963	10.8
bb.	.268	.153	.073	10000	.138	1.26	11.0
cc.	.264	.149	.071	10000	.135	1.27	10.6
dd.	.285	.170	.081	10000	.154	1.47	10.5
ee.	.287	.172	.082	10000	.155	1.46	10.6
ff.	.279	.164	.078	10000	.148	1.40	10.6
gg.	.289	.174	.083	10000	.157	1.47	10.7

II. PERFORMANCE DATA

Data Set	Air Temperature		Pressure				
	Rotameter Inlet (°F)	Turbine Inlet (°F)	Gage Zero (psig)	Relative Pressure		Absolute Pressure	
				Rotameter Inlet (psig)	Turbine Inlet (psig)	Rotameter Inlet (psia)	Turbine Inlet (psia)
El1a.*	73	73		96.0	10.4	113.7	28.1
b.*	73	73		96.0	10.2	113.7	27.9
c.*	73	73		96.8	10.3	114.5	28.0
d.*	73	73		96.0	10.2	113.7	27.9
e.	73	73		92.7	15.8	110.4	33.5
f.	73	73		93.2	15.8	110.9	33.5
g.	73	73		93.0	15.7	110.7	33.4
h.	73	73		91.5	15.4	109.2	33.1
i.	73	73		89.1	26.3	106.8	44.0
j.	73	73		91.8	27.7	109.5	45.4
k.	73	73		89.3	26.3	107.0	44.0
l.	73	73		87.4	25.8	105.1	43.5
m.	73	73		88.2	37.1	105.9	54.8
n.	73	73		90.0	38.1	107.7	55.8
o.	73	73		89.9	37.8	107.6	55.5
p.	73	73		88.3	37.1	106.0	54.8
q.	73	73		88.2	44.5	105.9	62.2
r.	73	73		89.5	47.5	107.2	65.2
s.	73	73		90.5	46.8	108.2	64.5
t.	73	73		86.6	45.0	104.3	62.7
u.	73	73		88.0	57.5	105.7	75.2
v.	73	73		85.6	54.7	103.3	72.4
w.	73	73		88.6	57.6	106.2	75.3
x.	73	73		86.2	54.6	103.9	72.3
y.	74	74		88.5	65.4	106.2	83.1
z.	74	74		84.3	63.6	102.0	81.3
aa.	74	74		85.5	62.7	103.2	80.4
bb.	74	74		89.2	65.7	106.9	83.4
cc.	74	74		87.1	76.6	104.8	94.3
dd.	74	74		83.9	73.3	101.6	91.0
ee.	74	74		83.6	73.5	101.3	91.2
ff.	74	74		86.0	75.6	103.7	93.3
gg.	75	75		86.8	83.4	104.5	101.1
hh.	75	75		86.4	83.0	104.1	100.7
ii.	75	75		80.4	80.8	98.1	98.5
jj.	75	75		82.2	81.2	99.9	98.9

II. PERFORMANCE DATA

Data Set	Turbine Inlet Pressure		Flow Rate		Load Cell		
	Friction Factor	Corrected Inlet Pressure (psia)	Volume (SCFM ₀)	Mass (lbm/sec)	Static Zero	Excitation Voltage (V DC)	Output Voltage (mV)
El1a.*	.0223	26.7	1.9	.0058		8.47	.640
b.*	.0223	26.5	1.9	.0058		8.47	.640
c.*	.0223	26.6	1.9	.0059		8.47	.640
d.*	.0223	26.5	1.9	.0058		8.47	.650
e.	.021	31.6	2.4	.0074		8.47	.690
f.	.021	31.6	2.4	.0075		8.47	.700
g.	.021	31.2	2.4	.0074		8.47	.700
h.	.021	31.2	2.4	.0074		8.47	.690
i.	.019	41.2	3.4	.0106		8.47	.810
j.	.019	42.6	3.4	.0107		8.47	.830
k.	.019	41.2	3.4	.0106		8.47	.810
l.	.019	40.7	3.4	.0105		8.47	.810
m.	.018	51.4	4.2	.0132		8.47	.940
n.	.018	52.4	4.2	.0133		8.47	.940
o.	.018	52.1	4.2	.0133		8.47	.950
p.	.018	51.4	4.2	.0132		8.47	.930
q.	.017	57.9	5.0	.0158		8.47	1.05
r.	.017	61.1	5.0	.0159		8.47	1.06
s.	.017	60.3	5.0	.0160		8.47	1.07
t.	.017	58.5	5.0	.0157		8.47	1.05
u.	.017	70.5	5.8	.0184		8.47	1.23
v.	.017	67.4	5.9	.0185		8.47	1.21
w.	.017	70.5	5.8	.0185		8.47	1.24
x.	.017	67.3	5.9	.0186		8.47	1.19
y.	.016	77.5	6.6	.0211		8.47	1.35
z.	.016	75.7	6.7	.0210		8.47	1.33
aa.	.016	76.6	6.7	.0211		8.47	1.34
bb.	.016	77.8	6.6	.0212		8.47	1.35
cc.	.016	88.1	7.5	.0239		8.47	1.51
dd.	.016	84.4	7.7	.0242		8.47	1.49
ee.	.016	84.8	7.6	.0238		8.47	1.48
ff.	.016	86.7	7.7	.0244		8.47	1.51
gg.	.015	94.5	8.0	.0255		8.47	1.59
hh.	.015	93.9	8.1	.0257		8.47	1.59
ii.	.016	91.8	8.2	.0253		8.47	1.54
jj.	.015	92.1	8.2	.0255		8.47	1.55

II. PERFORMANCE DATA

Data Set	Gross Force (lbf)	Net Force (lbf)	Torque (ft-lbf)	Speed (rpm)	Turbine Performance		
					Power Output (hp)	Power Input (hp)	Efficiency (%)
E11a.*	.126	.008	.004	20000	.0145	.173	8.4
b.*	.126	.008	.004	20000	.0145	.171	8.5
c.*	.126	.008	.004	20000	.0145	.171	8.4
d.*	.128	.010	.005	20000	.0181	.171	10.6
e.	.136	.018	.009	20000	.033	.277	11.9
f.	.138	.020	.009	20000	.036	.271	13.3
g.	.138	.020	.009	20000	.036	.271	13.3
h.	.136	.018	.009	20000	.033	.270	12.2
i.	.159	.041	.019	20000	.074	.500	14.8
j.	.163	.045	.021	20000	.081	.519	15.6
k.	.159	.041	.019	20000	.074	.500	14.8
l.	.159	.041	.019	20000	.074	.490	15.1
m.	.185	.067	.032	20000	.121	.729	16.6
n.	.185	.067	.032	20000	.121	.742	16.3
o.	.187	.069	.033	20000	.125	.744	16.8
p.	.183	.065	.031	20000	.117	.727	16.1
q.	.207	.089	.042	20000	.161	.942	17.1
r.	.209	.091	.043	20000	.164	.970	16.9
s.	.211	.093	.044	20000	.168	.971	17.3
t.	.207	.089	.042	20000	.161	1.00	16.1
u.	.242	.124	.059	20000	.224	1.22	18.4
v.	.238	.120	.057	20000	.217	1.20	18.1
w.	.244	.126	.060	20000	.228	1.23	18.6
x.	.234	.116	.055	20000	.210	1.20	17.5
y.	.266	.148	.070	20000	.267	1.46	18.3
z.	.262	.144	.068	20000	.260	1.44	18.1
aa.	.264	.146	.069	20000	.264	1.44	18.4
bb.	.266	.148	.070	20000	.267	1.47	18.2
cc.	.297	.179	.085	20000	.323	1.75	18.5
dd.	.293	.175	.083	20000	.316	1.74	18.2
ee.	.291	.173	.082	20000	.313	1.72	18.2
ff.	.297	.179	.085	20000	.323	1.78	18.2
gg.	.313	.195	.093	20000	.352	1.92	18.3
hh.	.313	.195	.093	20000	.352	1.93	18.3
ii.	.303	.185	.088	20000	.334	1.89	17.7
jj.	.305	.187	.089	20000	.338	1.91	17.7

II. PERFORMANCE DATA

Data Set	Air Temperature		Gage Zero (psig)	Pressure			
	Rotameter Inlet (°F)	Turbine Inlet (°F)		Relative Pressure		Absolute Pressure	
				Rotameter Inlet (psig)	Turbine Inlet (psig)	Rotameter Inlet (psia)	Turbine Inlet (psia)
E12a.*	74	74		93.8	25.9	111.5	43.6
b.*	74	74		95.3	26.1	113.0	43.8
c.*	74	74		94.1	26.0	111.8	43.7
d.*	74	74		95.6	26.3	113.3	44.0
e.	74	74		89.1	35.5	106.8	53.2
f.	74	74		88.9	35.6	106.6	53.3
g.	74	74		89.0	35.7	106.7	53.4
h.	74	74		91.1	37.5	108.8	55.2
i.	74	74		89.4	46.2	107.1	63.9
j.	74	74		88.5	46.1	106.2	63.8
k.	74	74		91.0	49.1	108.7	66.8
l.	74	74		90.8	48.9	108.5	66.6
m.	74	74		86.5	56.6	104.2	74.3
n.	74	74		90.7	57.7	108.4	75.4
o.	74	74		89.6	59.4	107.3	77.1
p.	74	74		87.1	58.0	104.8	75.7
q.	74	74		88.6	70.0	106.3	87.7
r.	74	74		87.6	69.5	105.3	87.2
s.	74	74		85.3	67.2	103.0	84.9
t.	74	74		84.8	66.8	102.5	84.5
u.	74	74		87.8	78.3	105.5	96.0
v.	74	74		83.7	75.5	101.4	93.2
w.	74	74		88.1	78.6	105.8	96.3
x.	74	74		83.6	75.7	101.3	93.4
y.	74	74		86.5	86.0	104.2	103.7
x.	74	74		84.9	82.0	102.6	99.7
aa.	74	74		89.1	84.5	106.8	102.2
bb.	74	74		89.1	84.4	106.8	102.1
E13a.*	74	74		92.3	63.0	110.0	80.7
b.*	74	74		89.5	62.0	107.2	79.7
c.*	74	74		91.1	61.5	108.8	79.2
d.*	74	74		89.9	59.3	107.6	77.0
e.	74	74		89.4	67.2	107.1	84.9
f.	73	73		88.9	67.0	106.6	84.7
g.							

II. PERFORMANCE DATA

Data Set	Turbine Inlet Pressure		Flow Rate		Load Cell		
	Friction Factor	Corrected Inlet Pressure (psia)	Volume (SCFM ₀)	Mass (lbm/sec)	Static Zero	Excitation Voltage (V DC)	Output Voltage (mV)
E12a.	.019	41.0	3.2	.0102		8.47	.770
b.	.019	41.0	3.3	.0106		8.47	.780
c.	.019	41.1	3.2	.0102		8.47	.770
d.	.019	41.2	3.3	.0106		8.47	.760
e.	.0183	49.8	4.1	.0129		8.47	.860
f.	.0183	49.9	4.1	.0129		8.47	.870
g.	.0183	50.0	4.1	.0129		8.47	.860
h.	.0181	51.7	4.2	.0133		8.47	.880
i.	.0172	59.3	5.2	.0166		8.47	.990
j.	.0172	59.3	5.2	.0165		8.47	.980
k.	.0172	62.5	5.1	.0164		8.47	1.01
l.	.0172	62.3	5.1	.0163		8.47	1.00
m.	.0166	69.2	6.0	.0189		8.47	1.09
n.	.0165	70.2	6.0	.0193		8.47	1.09
o.	.0166	72.1	6.0	.0192		8.47	1.10
p.	.0166	70.7	6.0	.0190		8.47	1.09
q.	.0161	82.3	6.7	.0214		8.47	1.21
r.	.0161	81.8	6.7	.0213		8.47	1.18
s.	.0161	79.3	6.8	.0214		8.47	1.19
t.	.0161	78.9	6.8	.0214		8.47	1.18
u.	.0156	89.7	7.6	.0243		8.47	1.28
v.	.0157	86.9	7.6	.0238		8.47	1.27
w.	.0157	90.1	7.5	.0240		8.47	1.30
x.	.0157	87.3	7.5	.0235		8.47	1.27
y.	.0153	97.0	8.2	.0261		8.47	1.25
z.	.0154	92.8	8.2	.0259		8.47	1.25
aa.	.0153	95.4	8.1	.0261		8.47	1.29
bb.	.0153	95.1	8.2	.0264		8.47	1.32
E13a.*	.0163	75.3	6.3	.0205		8.47	.930
b.*	.0164	74.5	6.2	.0199		8.47	.910
c.*	.0164	74.1	6.1	.0197		8.47	.900
d.*	.0165	72.0	6.0	.0192		8.47	.900
e.	.0161	79.2	6.7	.0215		8.47	.930
f.	.0161	79.1	6.7	.0215		8.47	.930
g.					.595	8.47	

II. PERFORMANCE DATA

Data Set	Gross Force (lbf)	Net Force (lbf)	Torque (ft-lbf)	Speed (rpm)	Turbine Performance		
					Power Output (hp)	Power Input (hp)	Efficiency (%)
E12a.*	.152	.034	.016	30000	.092	.477	19.3
b.*	.153	.035	.017	30000	.095	.497	19.1
c.*	.152	.034	.016	30000	.092	.479	19.2
d.*	.150	.032	.015	30000	.087	.500	17.4
e.	.169	.041	.019	30000	.111	.698	15.9
f.	.171	.043	.020	30000	.116	.699	16.6
g.	.169	.041	.019	30000	.111	.703	15.8
h.	.173	.045	.021	30000	.122	.744	16.4
i.	.195	.067	.032	30000	.182	1.01	18.1
j.	.193	.065	.031	30000	.176	.994	17.7
k.	.199	.071	.034	30000	.193	1.02	18.9
l.	.197	.069	.033	30000	.187	1.02	18.4
m.	.214	.086	.041	30000	.233	1.24	18.8
n.	.214	.086	.041	30000	.233	1.27	18.3
o.	.216	.088	.042	30000	.239	1.29	18.5
p.	.214	.086	.041	30000	.233	1.26	18.5
q.	.238	.110	.052	30000	.298	1.53	19.5
r.	.232	.104	.049	30000	.282	1.52	18.6
s.	.234	.106	.050	30000	.287	1.50	19.2
t.	.232	.104	.049	30000	.282	1.49	18.9
u.	.252	.124	.059	30000	.336	1.80	18.7
v.	.250	.122	.058	30000	.331	1.74	19.0
w.	.256	.128	.061	30000	.347	1.78	19.5
x.	.250	.122	.058	30000	.331	1.72	19.3
y.	.246	.118	.056	30000	.320	1.99	16.1
z.	.246	.118	.056	30000	.320	1.94	16.5
aa.	.254	.126	.060	30000	.342	1.98	17.3
bb.	.260	.132	.063	30000	.358	2.00	17.9
E13a.*	.183	.059	.028	40000	.213	1.40	15.2
b.*	.179	.055	.026	40000	.199	1.35	14.7
c.*	.177	.053	.025	40000	.192	1.34	14.3
d.*	.177	.053	.025	40000	.192	1.29	14.9
e.	.183	.059	.028	40000	.213	1.50	14.2
f.	.183	.059	.028	40000	.213	1.50	14.2

III. PREOPERATIONAL TEST - TURBINE ROTOR STATIONARY, INCREMENTAL LOAD TEST

Data Set	Load Cell				Measured Load (lbf)	Approximate Applied Load (lbf)
	Static Zero (mV)	Excitation (V DC)	Gross Output (mV)	Net Output (mV)		
A1a.	.620	8.92				
b.		8.92	1.225	.605	.113	.101
c.		8.92	1.487	.867	.148	.162
d.		8.90	2.835	2.215	.415	.367
A2a.	.607	8.43				
b.		8.43	1.116	.509	.101	.101
c.		8.43	1.389	.782	.155	.148
d.		8.43	2.075	1.468	.290	.249
e.		8.44	3.134	2.527	.499	.467
A3a.	.548	8.43				
b.		8.43	1.299	.751	.149	.148
c.		8.43	1.754	1.206	.238	.249
d.		8.43	2.890	2.342	.463	.467
A4a.	.550	8.46				
b.	.536	8.46				
c.		8.46	.549	.013	.003	.003
d.		8.46	.544	.008	.001	.006
e.		8.46	.551	.015	.003	.009
f.		8.46	.579	.043	.009	.012
g.		8.46	.595	.059	.012	.015
h.		8.46	.609	.073	.014	.019
i.		8.46	.637	.101	.020	.022
j.		8.46	.643	.107	.021	.025
k.		8.46	.637	.101	.020	.022
l.		8.46	.611	.075	.015	.019
m.		8.46	.595	.059	.012	.015
n.		8.46	.604	.068	.013	.012
o.		8.46	.586	.050	.010	.009
p.		8.46	.551	.015	.003	.006
q.		8.46	.538	.002	.0004	.003

III. PREOPERATIONAL TEST - TURBINE ROTOR STATIONARY, INCREMENTAL LOAD TEST

Data Set	Load Cell		Gross Output Voltage (mV)	Measured Load (lbf)	Incremental Change (lbf)	Approximate Applied Incremental Load (lbf)
	Static Zero (mV)	Excitation (V DC)				
A5a.	.527	8.46		.104		
b.		8.46	.539	.106	.002	.003
c.		8.46	.560	.110	.004	.003
d.		8.46	.574	.113	.003	.003
e.		8.46	.588	.116	.003	.003
f.		8.46	.597	.118	.002	.003
g.		8.46	.618	.122	.004	.003
h.		8.46	.632	.125	.003	.003
i.		8.46	.653	.129	.004	.003
j.		8.46	.643	.127	-.002	-.003
k.		8.46	.620	.122	-.005	-.003
l.		8.46	.603	.119	-.003	-.003
m.		8.46	.610	.120	-.001	-.003
n.		8.46	.585	.115	-.005	-.003
o.		8.46	.567	.112	-.003	-.003
p.		8.46	.551	.109	-.003	-.003
q.	.528	8.46		.104	-.005	-.003
r.	.528	8.46		.104	0	0
s.		8.46	.551	.109	.005	.003
t.		8.46	.537	.106	-.003	-.003
u.		8.46	.552	.109	.003	.003
v.		8.46	.531	.105	-.004	-.003
A6a.	1.635	8.46		.322		
b.		8.46	1.650	.325	.003	.003
c.		8.46	1.667	.328	.003	.003
d.		8.46	1.684	.332	.004	.003
e.		8.46	1.701	.335	.003	.003
f.		8.46	1.714	.338	.003	.003
g.		8.46	1.729	.341	.003	.003
h.		8.46	1.742	.343	.002	.003
i.		8.46	1.757	.346	.003	.003
j.		8.46	1.740	.343	-.003	-.003
k.		8.46	1.725	.340	-.003	-.003
l.		8.46	1.710	.337	-.003	-.003
m.		8.46	1.695	.334	-.003	-.003
n.		8.46	1.678	.331	-.003	-.003
o.		8.46	1.662	.327	-.004	-.003
p.		8.46	1.646	.324	-.003	-.003
q.	1.630	8.46		.321	-.003	-.003

III. PREOPERATIONAL TEST - TURBINE ROTATING, INCREMENTAL LOAD TEST

Data Set	Turbine Pressure			Load Cell		Gross Output (mV)
	Excitation (V DC)	Transducer Output (V DC)	Net Pressure (psia)	Static Zero (mV)	Excitation (V DC)	
A7a.	12.5	3.2	94.4	.537	8.46	
b.	12.5				8.46	.835
c.	12.5				8.46	.900
d.	12.5				8.46	.970
e.	12.5				8.46	.900
f.	12.5				8.46	.835
A8a.	12.5	3.3	96.9	1.673	8.47	
b.	12.5				8.47	1.938
A9a.				1.527	8.45	
b.	12.5	1.18	43.9		8.46	1.62
c.	12.5				8.46	1.93
d.	12.5				8.46	1.95
e.	12.5				8.46	1.97
f.	12.5				8.46	1.99
g.	12.5				8.46	2.02
h.	12.5				8.46	2.05
i.	12.5				8.46	2.02
j.	12.5				8.46	1.98
k.	12.5				8.46	1.95
l.	12.5				8.46	1.91

III. PREOPERATIONAL TEST - TURBINE ROTATING, INCREMENTAL LOAD TEST

<u>Data Set</u>	<u>Measured Load (lbf)</u>	<u>Incremental Change (lbf)</u>	<u>Applied Incremental Load (lbf)</u>	<u>Speed (rpm)</u>
A7a.				
b.	.164			20000
c.	.177	.013	.012	20000
d.	.191	.014	.012	20000
e.	.177	.014	-.012	20000
f.	.164	.013	-.012	20000
A8a.	.381			20000
b.	.381			20000
A9a.				
b.	.319			20000
c.	.380			20000
d.	.384	.004	.006	20000
e.	.388	.004	.003	20000
f.	.392	.004	.003	20000
g.	.398	.004	.006	20000
h.	.404	.006	.006	20000
i.	.398	-.006	-.006	20000
j.	.390	-.008	-.006	20000
k.	.384	-.006	-.006	20000
l.	.376	-.008	-.006	20000

III. PREOPERATIONAL TEST - TURBINE ROTATING, INCREMENTAL LOAD TEST

Data Set	Load Cell		Gross Output Voltage (mV)	Measured Load (lbf)	Incremental Change (lbf)	Approximate Incremental Load (lbf)
	Static Zero (mV)	Excitation (V DC)				
A10a.	.481	8.47				
b.		8.47	.430	.085		
c.		8.47	.445	.088	.003	.003
d.		8.47	.460	.091	.003	.003
e.		8.47	.475	.093	.002	.003
f.		8.47	.485	.095	.002	.003
g.		8.47	.505	.099	.004	.003
h.		8.47	.525	.103	.004	.003
i.		8.47	.540	.106	.003	.003
j.		8.47	.550	.108	.002	.003
k.		8.47	.425	.084	.024	-.025
A11a.		8.47	1.54	.303		
b.		8.47	1.56	.307	.004	.003
c.		8.47	1.59	.313	.006	.006
d.		8.47	1.63	.321	.008	.009
e.		8.47	1.67	.329	.008	.003

APPENDIX D

TURBINE EFFICIENCY UNCERTAINTY ANALYSIS

APPENDIX D

TURBINE EFFICIENCY UNCERTAINTY ANALYSIS

To provide an indication of the uncertainty of the turbine efficiency data presented in Appendix C an uncertainty analysis based on a general method described by Schenck⁽¹⁾ was performed and is outlined herein.

Data uncertainty is commonly expressed in terms of a percent error which is based on an uncertainty interval, ω , and the dependent variable value, y . Percent error is expressed as

$$\% \text{ Error} = \left(\frac{\omega y}{y} \right) \times 100\% \quad (\text{D.1})$$

The uncertainty interval for a dependent variable which is a continuous and differentiable function of n independent variables, x_n is given by⁽¹⁾

$$\omega_y = \left\{ \left(\frac{\partial y}{\partial x_1} \right)^2 \omega_{x_1}^2 + \left(\frac{\partial y}{\partial x_2} \right)^2 \omega_{x_2}^2 + \dots + \left(\frac{\partial y}{\partial x_n} \right)^2 \omega_{x_n}^2 \right\}^{1/2} \quad (\text{D.2})$$

The partial derivatives in Equation D.2 are determined by first stating the functional dependence of y on the independent variables x_n and then performing the indicated differentiation. Uncertainty interval estimations originate in a variety of sources. For example, let x_1 represent the tachometer reading in rpm; ω_{x_1} may then be represented by a percentage of the full scale deflection. Consider also the uncertainty interval of the load cell measurement, x_2 ; ω_{x_2} may be represented by one-half the least significant digit of a steady voltmeter display reading or a percentage of the average of a fluctuating reading.

Turbine efficiency, η , is a function of inlet outlet pressures, P_i and P_o ; inlet temperature, T_i ; mass flow rate, $\dot{m}$; rotational speed, N ; force, F ; moment arm length, L . The functional dependence is given by

$$\eta = \frac{NF L \frac{2\pi}{60}}{\dot{m} R \frac{\gamma}{\gamma-1} T_i \left[1 - \left(\frac{P_o}{P_i} \right)^{\gamma/\gamma-1} \right]} \quad (\text{D.3})$$

where R is the gas constant and γ is the ratio of specific heats.

Differentiating Equation D.3, assuming the turbine discharge pressure, P_o , to be invariant and arranging the terms as Equation D.2 yields:

$$\omega_\eta = \left\{ \left(\frac{\partial \eta}{\partial N} \right)^2 \omega_N^2 + \left(\frac{\partial \eta}{\partial F} \right)^2 \omega_F^2 + \left(\frac{\partial \eta}{\partial l} \right)^2 \omega_l^2 + \left(\frac{\partial \eta}{\partial T_i} \right)^2 \omega_{T_i}^2 + \left(\frac{\partial \eta}{\partial P_i} \right)^2 \omega_{P_i}^2 \right\}^{1/2} \quad (D.4)$$

The uncertainty intervals of speed, force, moment arm length, temperature and pressure are based on instrument readability and measurement fluctuation, Table D.1.

TABLE D.1. Uncertainty Intervals

<u>Variable</u>	<u>Uncertainty Interval,</u>
Speed, N	$\pm 0.5\%$ full scale value (± 200 rpm)
Force, F	$\pm 10\%$ data point
Moment arm length, l	± 0.005 in., $\pm 4.17 \times 10^{-4}$ ft
Temperature, T_i	$\pm 1^\circ\text{F}$, $\pm 1^\circ\text{R}$
Pressure, P_i	$\pm 5\%$ reading of pressure transducer or $\pm 0.2\%$ full scale value (± 4 psig) of Heise gage

Mass flow rate uncertainty interval is obtained by stating the functional dependence of mass flow rate on its appropriate independent variables and differentiating in accordance with Equation D.2 as follows:

$$\dot{m} = Q \frac{P_r}{RT_r} \quad (D.5)$$

$$\omega_{\dot{m}} = \left\{ \left(\frac{\partial \dot{m}}{\partial Q} \right)^2 \omega_Q^2 + \left(\frac{\partial \dot{m}}{\partial P_r} \right)^2 \omega_{P_r}^2 + \left(\frac{\partial \dot{m}}{\partial T_r} \right)^2 \omega_{T_r}^2 \right\}^{1/2} \quad (D.6)$$

where Q is the volumetric flowrate.

The volume flow rate uncertainty interval is 0.05 ft³/min and the uncertainty intervals of rotameter pressure and temperature are taken from Table D.1.

Results of a complete analysis of data point E 10.f are summarized in Tables D.2, D 3, and D.4. The parameters of E10.f are among the lowest obtained during testing; hence the percentage errors tabulated are likely to be among the highest for the entire testing sequence.

TABLE D.2 Parameters of Data Point E10.f

Parameter	Value
Temperatures	
rotameter, T _r	533R
turbine inlet, T _i	533R
Pressures	
rotameter, P _r	111.5 psia
turbine inlet, P _i	22.9 psia
discharge, P _o	14.4 psia
Volume flow rate, Q (corrected to calibration)	.491 cfm
Mass flow rate, $\dot{m}$	0.0046 lbm/sec
Speed, N	10,000 rpm
Force, F	0.021 lbf
Moment arm length, l	0.475 ft
Efficiency, η	.182

TABLE D.3. Summary of Mass Flow Rate Uncertainty Calculation for E10.f

<u>Parameter</u>	<u>Value</u>	<u>Uncertainty Interval, ω</u>	<u>Percent Error, %</u>
Q	.491 ft ³ /min	0.05 ft ³ /min	10.2
T _r	533 R	1 R	0.2
P _r	111.5 psia	4 psi	3.6
$\dot{m}$	0.0046 lbm/sec	4.99 (10 ⁻⁴) lbm/sec	10.8

TABLE D.4. Summary of Uncertainty Interval Calculations for Data Point E10.f

<u>Parameter</u>	<u>Value</u>	<u>Uncertainty Interval, ω</u>	<u>Percent Error, %</u>
N	10,000 rpm	200 rpm	2
F	0.021 lbf	0.0021 lbf	10
l	0.475 ft	4.17 (10 ⁻⁴) ft	0.09
$\dot{m}$	0.0046 lbm/sec	4.99 (10 ⁻⁴) lbm/sec	10.8
T _i	533 R	1 R	0.2
P _i	22.9 psia	4 psi	17.5
η	.182	.0284	15.6

References

1. Schenck, H. 1979. Theories of Engineering Experimentation. Third Edition. Hemisphere Publishing Corporation, Washington, D.C.

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