

INDUSTRIAL GAS TURBINE PERFORMANCE MEASUREMENT

by

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The author was born on the outskirts of Toronto, and attended Public and High School there. During his teens, his main hobby interest was building and flying rubber-powered model aircraft. He attended the University of Manitoba in Winnipeg for 4 years, graduating with a Bachelor of Science degree in Electrical Engineering.

After graduation, he joined the Crown company, Turbo Research Limited, and later moved to Malton to join the Gas Turbine Division of A. V. Roe Canada Limited. Since then he has been continuously in the employ of what is now known as Hawker Siddeley Canada Ltd., Orenda Division, where he holds the position Turbine Design Manager.

ABSTRACT

An accurate assessment of the performance of an industrial gas turbine, in service, has been the goal of many test programs initiated by users and manufacturers alike. The rewards of such programs often are quite skimpy because of the difficulty of obtaining an accurate measurement of some of the basic engine parameters necessary for this assessment, power output being one of the more obstinate parameters.

The author has been indirectly involved, during the past two and a half years, in devising and carrying out factory and field performance tests which had, as a primary purpose, a lessening of the uncertainties involved in measuring and interpreting these parameters. Where possible, direct measurement of power output, turbine inlet temperature and air mass flow was utilized for comparison with other, less direct methods of measurement.

Those installations in which electrical power is the end product are the most compliant, since the electrical generator driven by the turbine provides an accurate determination of power output. On the other hand, a load consisting of a pump or compressor presents a much less accurately determined picture of power output. It is those installations having the latter type of loading device which have been the subject of the investigations reported in this paper.

INTRODUCTION

In-house and field performance tests on several industrial gas turbines have been carried out during the

*International Congress on Combustion Engines. Congress International des Machines a Combustion.

past three years by the author's company, for the purpose of establishing actual performance.

Earlier tests had yielded a good deal of conflicting evidence, both in the form of performance data which did not agree with the predicted performance, and in the measured values of the important parameters not satisfying the basic laws of thermodynamics.

More recent tests have been specifically planned to utilize instrumentation which would provide more consistent and creditable data readings, and to establish a firm basis of knowledge of the performance of the main engine components.

Three separate series of tests, which have been carried out recently, are dealt with in this paper to illustrate the learning process that goes with devising meaningful tests, and to highlight the successes of the effort.

These tests were carried out under standard procedures and rules similar to those defined under the ASME Power Test Code 22 or the CIMAC* code of acceptance requirements.

They were performed on normal production units which were being prepared for delivery or for on-site acceptance.

Where doubt existed as to our ability to measure accurately any of the important parameters, an attempt was made to provide an alternative, even if indirect, method of determining their values. The resulting experience has allowed us, now, to specify with confidence, the instrumentation for any future tests.

PERFORMANCE TEST PROGRAMS

The three test programs are dealt with here individually.

Field Load Test

This involved the field acceptance testing of one of our OT-390 units, one which comprises an aircraft engine derived gas generator and a matched expansion power turbine. It presented our first opportunity to measure the installed output directly, and care was taken with the design and installation of the instruments used in measuring the parameters important to the determination of power output and heat rate. Of particular concern in measurement were:

1. *Output Torque*

The torque measuring device available for this test functioned by sensing distortion of the magnetic flux induced in the torque shaft when the shaft is subjected to

torsional strain. This device is pictured in Figure 1. Normally the relationship between flux change and shaft strain was shown by static rig tests to be linear. However, in-plant turbine load tests had revealed unexpected inconsistencies, necessitating an experimental test program on the static torque calibration rig illustrated in Figure 2 to determine those factors which might influence its calibration.

These static rig experiments revealed the following relationships:

- (a) Temperature of shaft and magnetic system—little affect up to the maximum temperature to which the system would be subjected.

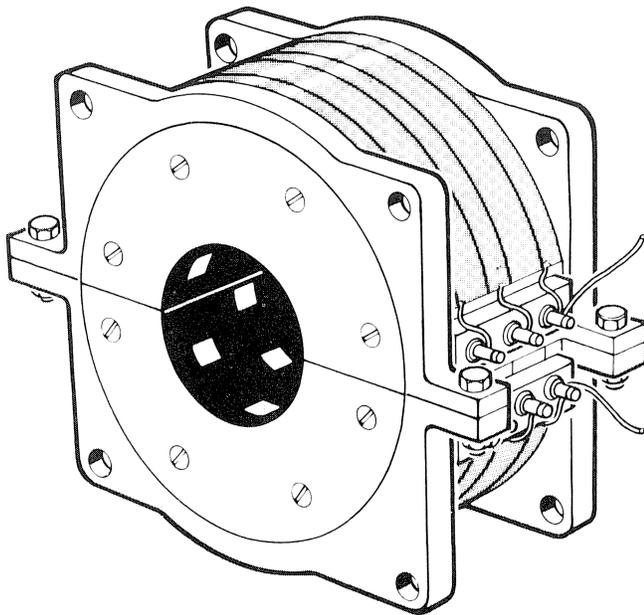


Figure 1. Magnetic Flux Type Torquemeter.

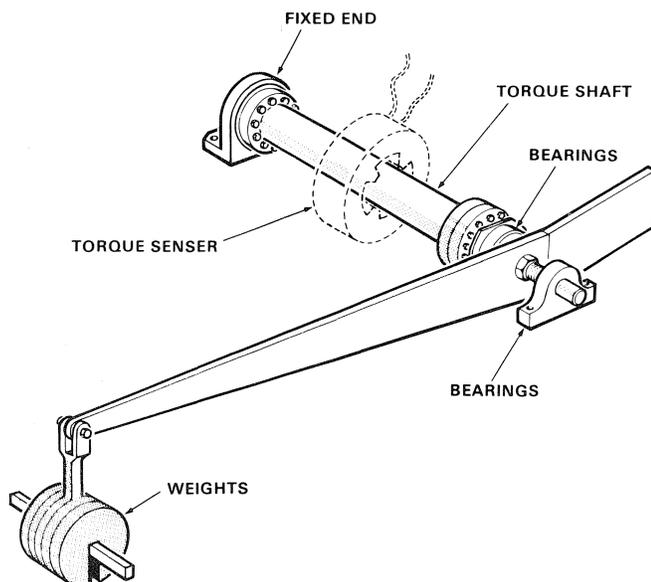


Figure 2. Torquemeter Static Calibration Rig.

- (b) Oil mist in the air gap—little affect.
- (c) Eccentricity of the magnetic system relative to the shaft—minor deviation from ideal calibration.
- (d) Angular displacement of the shaft relative to the magnetic system—major deviation from the ideal calibration, see Fig. 3.

It would appear then that the thermal expansion of the turbine supports was causing a displacement of the turbine output shaft, relative to the shaft of the loading device, resulting in a movement of the coupling torque shaft relative to the magnetic system. A redesign to the support housing was instituted, fitting it with pivot points co-planar with the toothed couplings of the torque shaft so that the axis of the magnetic system, in its mounting on the housing, would remain concentric and in angular alignment with the torque shaft. This is illustrated in the sketch in Figure 4.

This redesigned support housing with the magnetic sensing head was assembled to the static calibration rig to allow static load tests to be carried out. The results shown in graphical form in Figure 5, indicate an acceptable consistency of calibration. The variation did not exceed 75 lb. ft., less than 0.5%, at full load.

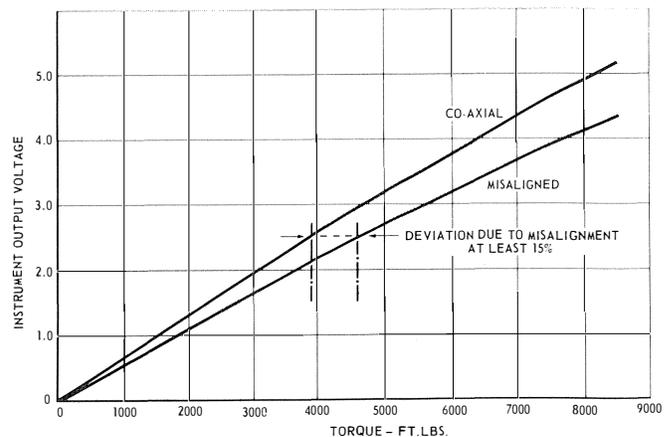


Figure 3. Instrument Error Due to Misalignment—Torquemeter.

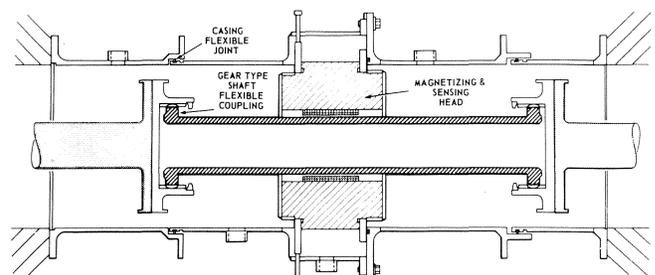


Figure 4. Torquemeter Centering Arrangement.

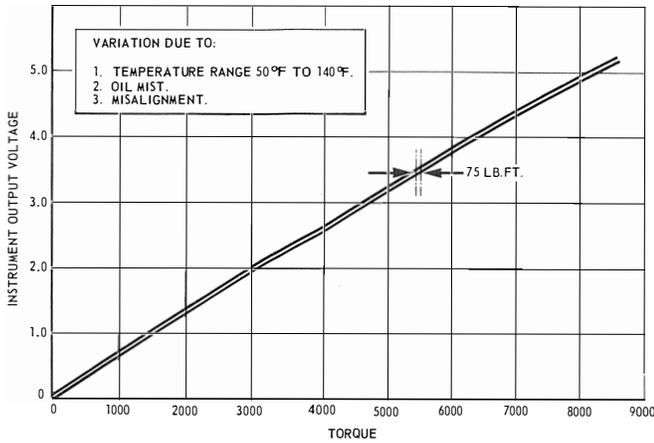


Figure 5. Static Calibration Curve—Torquemeter.

This field test of the OT-390 turbine installation provided the first opportunity to try out the redesigned torque-meter mounting system and consequently some apprehension attended its use.

2. Turbine inlet temperature

As a company, we have never had too much faith in the ability to make an accurate measurement of mean turbine inlet temperature, even with multiple rakes. However for these tests we did fit a single 3 point stagnation thermocouple rake in the outlet from each of the six combustion chambers to provide a rough guide as to temperature. During the course of this program we had continuously to guard against the tendency creeping in to accept these single unit readings as being a real mean outlet temperature, a tendency which resulted from the consistency of readings obtained.

3. Power turbine exhaust temperature

In this turbine unit, power turbine exhaust gas temperature is the parameter which is used to limit power turbine output. Exhaust gas temperature is normally sensed through six thermocouple units, three of which are installed in each of the paired exhaust duct elbows. These are conventionally designed stagnation type thermocouples of Chromel alumel. These thermocouples were felt to be reliable, but data from earlier tests had indicated a very significant incompatibility with the temperature data derived elsewhere. The statement was made that readings

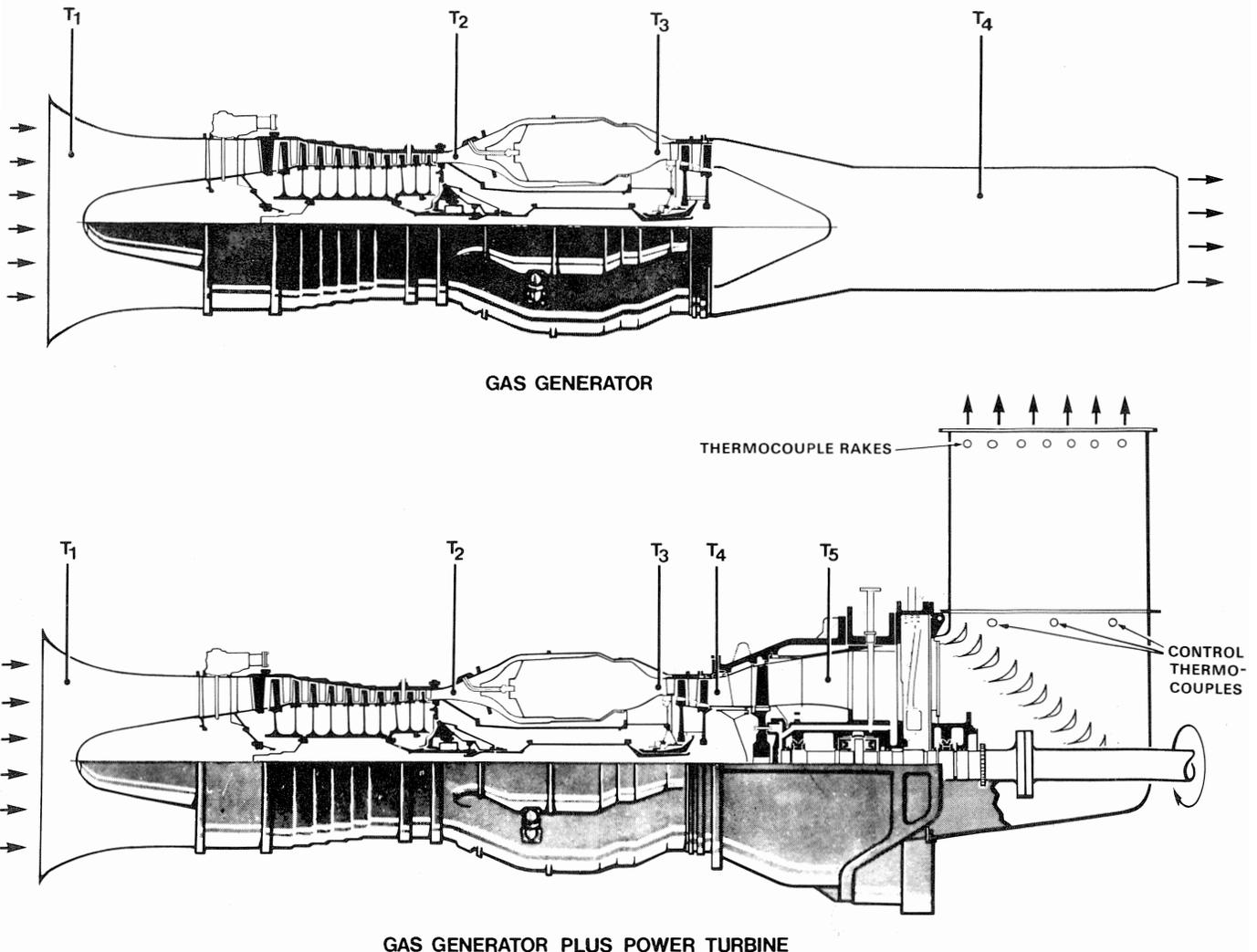


Figure 6. Points of Temperature Measurement.

of exhaust temperature were lower than they should be by about 60° C. when relating them to the measurements previously made of turbine inlet temperature and inter-turbine temperature. The last was particularly disconcerting since the inter-turbine temperature readings obtained on the earlier power turbine test corresponded reasonably well with the gas generator outlet temperature (jet pipe temperature) obtained from the cell tests of the gas generator. The points of temperature measurement are illustrated in Figure 6.

This situation did not rest well with us, since we could see no reason why the exhaust gas temperature measurements should be inaccurate, unless of course the temperature profile was badly distorted. To resolve this question a set of 14 five point thermocouple rakes was made up to obtain a complete grid of temperature readings across the exhaust duct. On test the individual and the average readings obtained from these rakes were compared with simultaneous readings obtained from the six normal exhaust temperature control thermocouples. The differences were found to be almost negligible, not exceeding about 10 centigrade degrees under any of the test conditions. This rather conclusive evidence threw the suspicion back onto the readings of turbine inlet and interturbine temperature.

Experience since that time has confirmed the validity and the reliability of readings obtained from the normal exhaust duct "control" thermocouples, and we now have sufficient confidence in their accuracy that we need not check them with the multiple 5 point rakes.

The primary purpose of this field test was to establish that the design rating of 9280 H.P. NEMA is achieved within the limiting gas generator speed and power turbine outlet temperature.

Test data from all of the instrumentation were recorded over a range of gas generator speeds varying from 6500 rpm to 7525 rpm and power turbine speeds varying from 6100 rpm to 7500 rpm. The test results were reduced to standard conditions (80° F @ 1000 ft.) for presentation in graphical form.

Figure 7 shows the relationship between gas generator speed and power output as measured by the torque-meter. Also shown for comparison is the shape of the curve as derived from heat balance calculations. And finally is included a plot of power output based on the calculations of power absorbed by the gas booster driven by the turbine. This latter information was obtained from the customer who had monitored the throughput of the gas booster during the test and calculated power absorbed on the basis of measured gas flow and pressure rise, and previously determined performance characteristic.

Transferring the test gas generator speeds to the Graph of Gas Generator Speed versus Power Turbine Exhaust Temperature, Figure 8, shows the exhaust gas temperature at which the machine must be run to produce the NEMA turbine output rating of 9280 HP, depending on which method of measuring power output is selected.

The result of this comparison of the operating conditions necessary to provide the NEMA design output of 9280 HP, as derived by the three different methods are shown in tabular form in Figure 9.

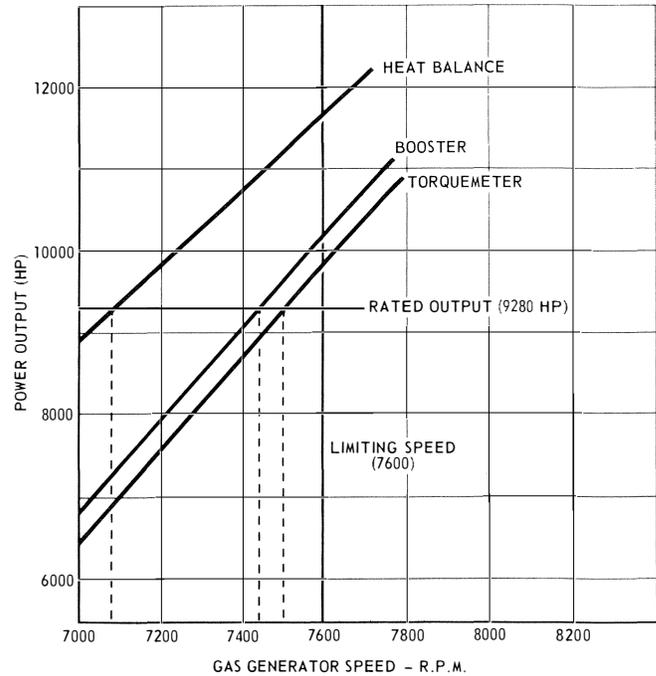


Figure 7. Power Output Test Results (NEMA).

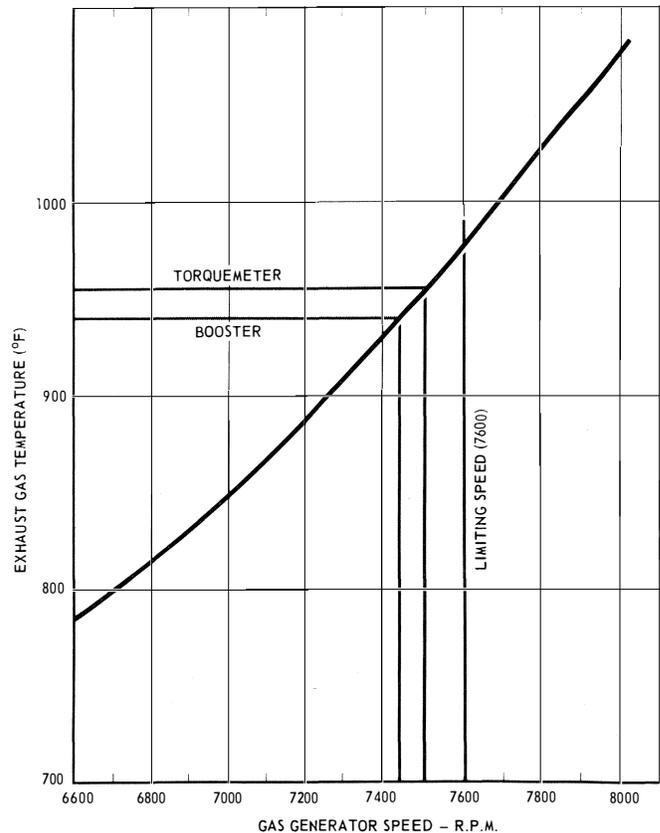


Figure 8. Test Exhaust Temperature at Rated Power (NEMA).

| PARAMETER | DESIGN CONDITIONS | BASED ON TORQUEMETER | DETERMINED BY HEAT BALANCE | BASED ON BOOSTER CALCULATIONS |
|------------------------------|-------------------|----------------------|----------------------------|-------------------------------|
| GAS GENERATOR SPEED (R.P.M.) | 7569 | 7500 | 7080 | 7440 |
| EXHAUST GAS TEMPERATURE (°F) | 986 | 955 | 864 | 940 |

Figure 9. Operating Conditions to Provide NEMA Output of 9280 H.P.

As indicated by the tabulation, the measurements from the torque meter showed reasonable agreement with the predicted design performance, indicating that the re-design aimed at maintaining shaft concentricity and alignment was reasonably successful.

The results based on a heat balance of the gas turbine thermodynamic cycle appear to present an overly optimistic picture of the engine performance, and perhaps serve to underline the sensitivity of this method of analysis to the accuracy of measurement of all the parameters involved.

The calculations of shaft horsepower absorbed by the 'load' booster, rely on the use of certain booster characteristic values whose accuracy we were not in a position to judge; however, the results would seem to confirm that the torque meter readings were not too optimistic.

The results achieved in this field test were adequate for the purposes at the time, but the relatively large differences between the torque meter and the heat balance answers, leave a good deal to be desired if one is trying to make an accurate assessment of gas turbine performance.

In-house Load Test

The experience of the tests just described provided some insight as to the sources of inaccuracies which can creep into the various measurements and served as a guide in prescribing the test procedure and methods of measurement to be used in the next series of tests. This next exercise involved the comparative testing of two like OT-390 gas turbine units. As with the earlier test, the gas generators were subject to the normal acceptance test in the test cell prior to fitting to the power turbine.

Acceptance testing of the combined gas generator/power turbine units was carried out in the factory on a special test rig, using an axial compressor to load the power turbine as illustrated in Figure 10. This time, however, a torque measuring device using toothed wheels mounted on the shaft in conjunction with stationary magnetic pickups was used to sense and measure torsional strain in the shaft during rotation under load. The complete torque sensing device included a solid state electronic "black box" which converted the strain reading to a torque reading, and because it senses speed, it also provided a digital output of power transmitted, calibrated in kilowatts.

One of the aims of this test was to measure the performance over the full range of operating speeds. Variation in load applied to the power turbine at various speeds was achieved by adjusting the load compressor inlet duct butterfly valve. In practice, it was found that the load

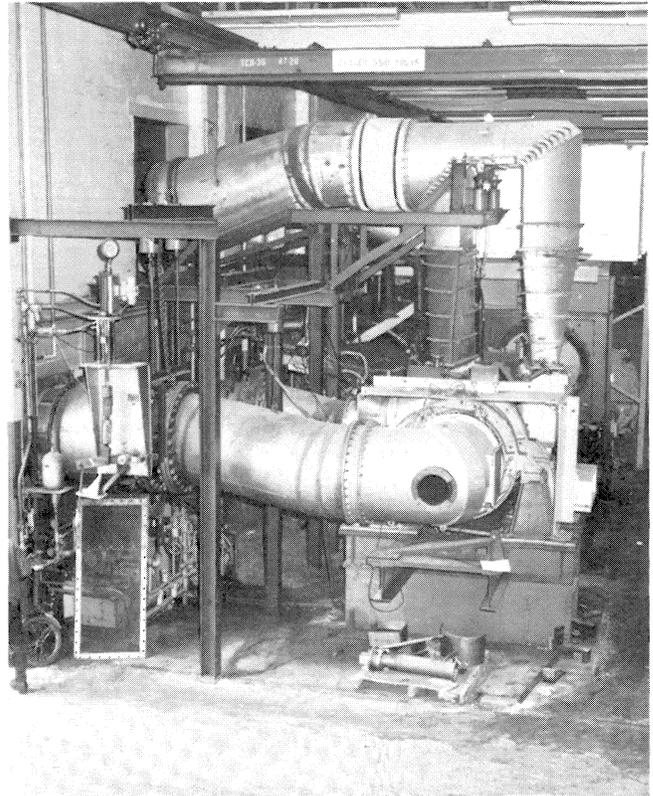


Figure 10. Power Turbine Coupled to Load Compressor.

compressor operating characteristic would not allow sustained full load operation at the lower power turbine speeds.

Since we had no real information on the accuracy or reliability of this new torque meter, we included sufficient instrumentation in both the load compressor and the gas turbine under test to allow calculation of the power and heat rate by several other means for cross-checking. For example:

1. Power Output Determination from Load Compressor

- (1) Measurement of load compressor inlet static and total pressure and inlet and outlet temperature allowed calculation of mass flow and temperature rise, and hence power absorbed.
- (2) Measurement of load compressor inlet and outlet total pressure and speed provided sufficient information to allow picking off from the compressor performance map as presented in Figure 11 the mass flow and efficiency and hence to allow calculation of power absorbed.

For both the above cases, the accuracy of pressure measurement left a lot to be desired since these measurements had to be made immediately downstream of a close-coupled vaned right angled elbow. The problem is illustrated in Figure 12.

2. Power Output Determination from Gas Turbine

- (1) Determination of compressor inlet temperature and static and total pressure allowed calculation

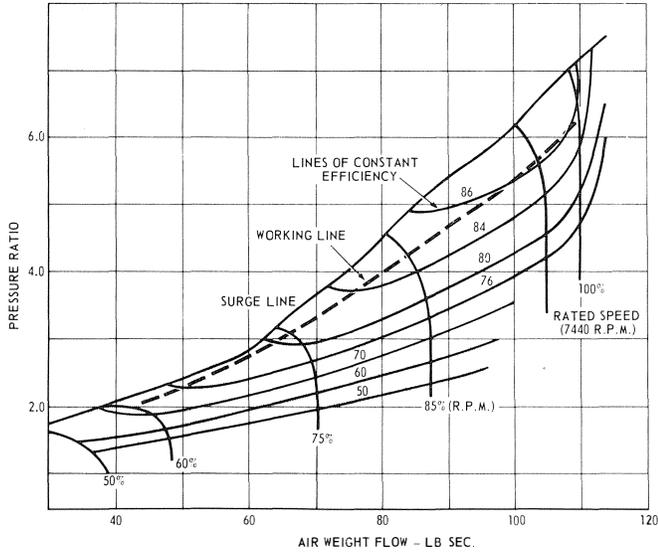


Figure 11. Compressor Characteristic.

of compressor flow. Compressor work was computed using the measured compressor discharge temperature. Then a heat balance calculation, considering power turbine outlet temperature and fuel flow and heating value allowed deter-

mination of the useful power output. In this calculation, allowance was made for the various air bleeds and for the heat loss to the lubrication system. Figure 13 is a "control" volume of the gas turbine illustrating the various inputs and outputs which went into the analysis.

- (2) The above method of analysis was repeated using the standard compressor characteristic to predict the air mass flow.
- (3) Whereas in (1) above the air mass flow was determined by measurement of pressures at the compressor inlet bellmouth, a third method of

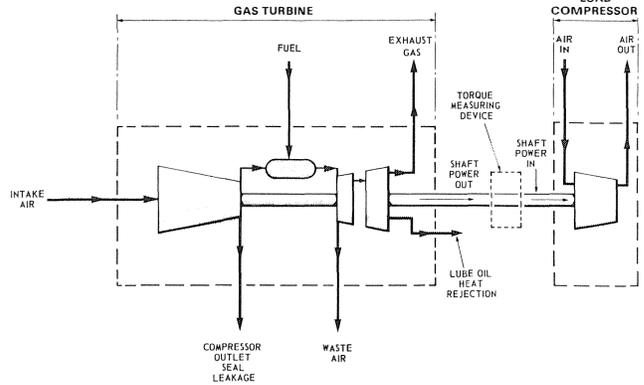


Figure 13. Control Volume for Performance Calculations.

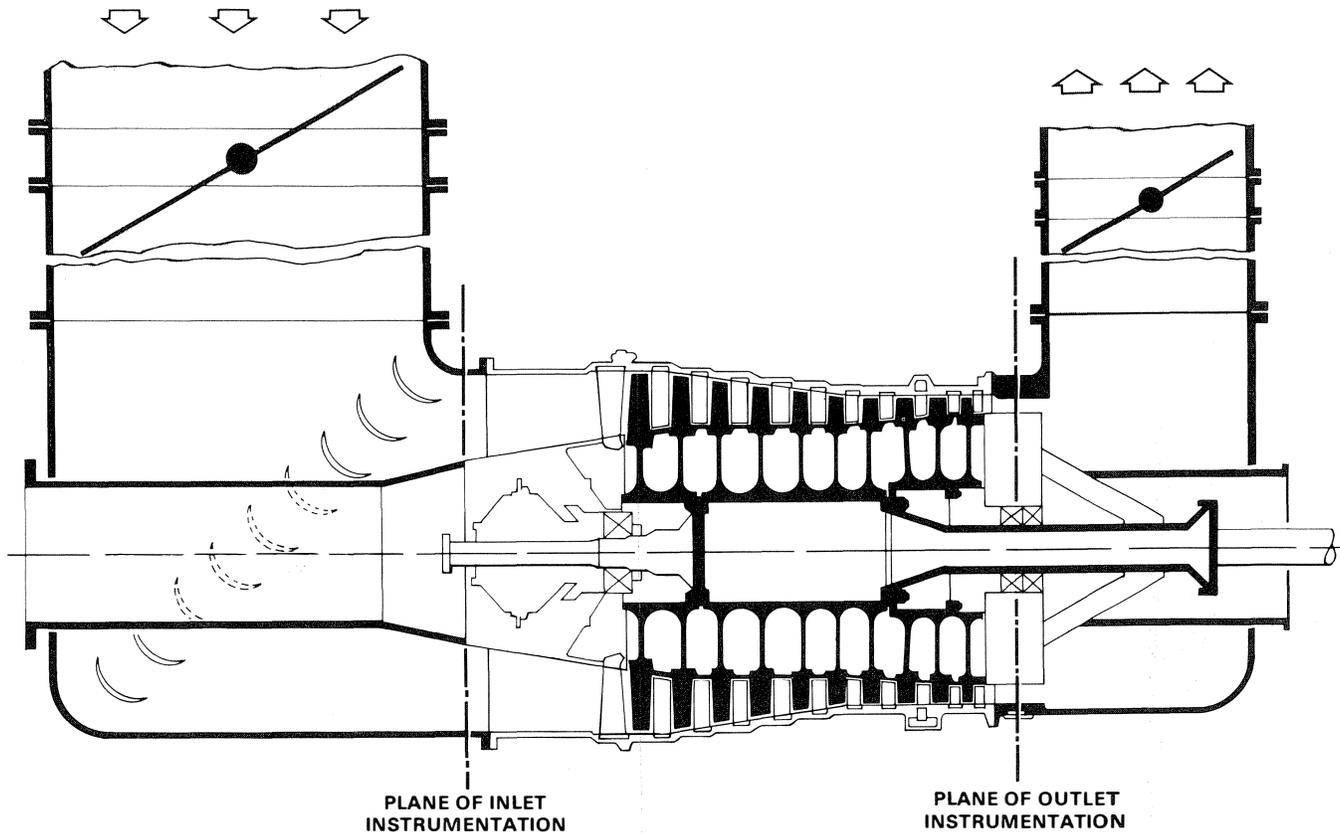


Figure 12. Load Compressor Air Ducting.

air flow determination was employed, using a standard venturi (with appropriate static tapings), located at inlet to the compressor inlet ducting.

The instrumentation applied to the gas turbine itself was the same as that used in the field test first described except that the turbine inlet thermocouples were deleted. The fourteen 5-point rakes were used in the exhaust duct, again to back up the normal six "control" thermocouples. Three sets of data readings at each test point were recorded at 20 minute intervals to ensure stability of conditions, and the middle of each of these three sets was selected for complete analysis, twenty in all.

The power output, calculated by carrying out heat balance analyses on the gas turbine and on the load compressor both turned out to be somewhat different from the results provided by the torquemeter, but nonetheless, they did indicate that the gas turbine is producing more than the required power of 9320 HP at site conditions. This information is shown in the tabulation of Figure 14.

In all these tests, the torquemeter seemed to give quite consistent readings, and in the analysed data the variation did not appear to exceed a total of about two and a half percent. If this is a true indication of its accuracy, we consider it to be a good device for any future tests requiring measurement of power output.

Gas Generator Performance Test

The third of the three test programs undertaken involved an uprating of an existing model of gas turbine, an OT-2100. We will eventually have an opportunity to carry out a full load test on this unit since it will be coupled to an electric generator, and a liquid rheostat has been provided to absorb the output of the generator. This will allow an accurate measurement of the completely installed output. However, for the present we considered it necessary to carry out an evaluation of the performance of the uprated unit before the power turbine loading system becomes available. In particular it was necessary to evaluate the need for rematching the turbine to the compressor to ensure that the uprated power rating could be achieved without overspeeding the gas generator, or exceeding the allowable exhaust gas temperature.

Out of this came the decision to build a gas generator version of this industrial unit, and to simulate the throttling effect of the power turbine by means of a suit-

ably sized conical nozzle, as is done in other gas generator tests. Such a gas generator was built, as shown in the photograph of Figure 15, and tested as a jet engine.

We did not have an accurate measure of the flow parameter (swallowing capacity) of the power turbine and hence the simulation of it in the form of a conical nozzle could be considered to be only approximate. This necessitated carrying out the performance tests of the "jet" engine with three different sizes of conical nozzle, 95%, 100% and 105% of the theoretical area, to ensure that the tests did encompass the actual condition of the power turbine.

Anticipating that the design changes involved in uprating this unit might affect the matching of the turbine/compressor system, and since resizing of the power turbine would be quite costly, provision was made for adjusting the throat area of the gas generator first stage of nozzle guide vanes. In all, three sizes of first stage nozzle guide vanes were tested with each of the three conical nozzle sizes, a total of 9 test configurations.

Analysis of these data allowed determination of the optimum relationship of first stage NGV throat area to final nozzle area. As with the tests on the OT-390 unit, allowance was made for the losses due to air bleeds and lubricating oil system heat rejection.

The installed design rating for the uprated unit had been set at 8640 KW with a heat rate of 4360 k Cal/KW.Hr. This must be achieved without exceeding the rated conditions of 7900 rpm gas generator speed and 1620° F turbine inlet temperature.

| AS DETERMINED FROM | POWER OUTPUT OF GAS GENERATOR (HP) | |
|------------------------------|------------------------------------|--------------|
| | SERIAL #5904 | SERIAL #5905 |
| GAS GENERATOR HEAT BALANCE | 10700 | 10290 |
| LOAD COMPRESSOR HEAT BALANCE | 9640 | 9880 |
| TORQUEMETER | 10130 | 10980 |

Figure 14. OT-390 Measured Power Output.

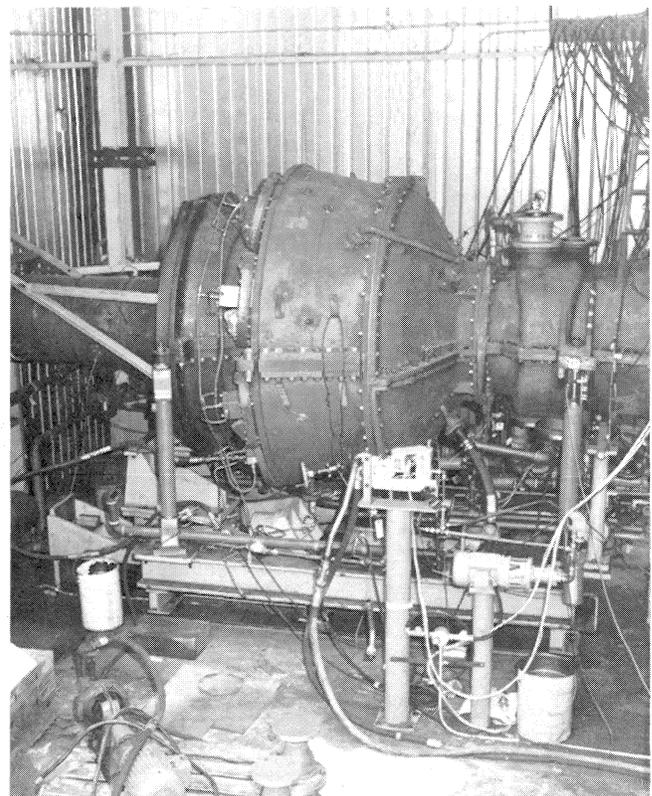


Figure 15. OT-2100 Gas Generator on Test.

The performance of the unit was calculated on the basis of gas HP, and then this was expressed in terms of power turbine shaft output power by applying the appropriate turbine efficiency and exhaust duct loss factors.

The calculated performance data were laid out in carpet plot form to present an easily understood view of the inter-relationships of first stage nozzle guide vane area, power turbine size, speed, exhaust gas temperature, power output and heat rate.

For example, Figure 16 shows how the installed heat rate (calculated from the test data) varies with first stage nozzle guide vane area and with power turbine effective area, at the rated power output of 8640 KW. It can be seen that for all values of NGV area and final nozzle size tested, the heat rate falls within the required limits. The very marked deterioration of heat rate at the larger size of NGV is not yet properly explained, though it may indicate some inaccuracies of measurement. It might be noted that the values of heat rate represented in the carpet plot were based on measurements of fuel flow. A cross check of those results, basing the calculations instead on air and gas temperatures, provided figures which were lower by about 1½ percent.

A second carpet plot, Figure 17, shows the variation of turbine inlet temperature with NGV area and final nozzle size at the rated installed output of 8640 KW.

The graph shows that at optimum NGV and final nozzle sizes, the required power output can be achieved at a turbine inlet temperature considerably lower than the rated value, but also that within the range of final nozzle sizes tested, any of the first stage turbine nozzle guide vanes tested would have functioned satisfactorily.

The third carpet plot, figure 18, provides the key to selection of the first stage nozzle guide vane size. It shows that the gas generator speed can be contained within the rated value, even should the final nozzle size (power turbine flow parameter) be as much as 5 percent oversize, simply by ensuring that the first stage NGV area is kept within the limits 101 percent and 104 percent of the nominal value.

The information presented in these graphs has shown that the gas generator is tolerant of a wide range of values of final nozzle size (power turbine flow parameter), and has allowed selection of a first stage nozzle guide vane throat area, 102 percent of nominal, which will accept the widest margin of variation in the power turbine.

CONCLUSIONS

Perhaps the best lesson we have learned from this work is that we should never blindly accept the data measured from tests, but should be continually comparing and relating such data to other information which may become available from time to time. Eventually you reach the point where you can accept certain techniques of measurement as being reasonably consistent and accurate, and use them as a standard reference base for future work.

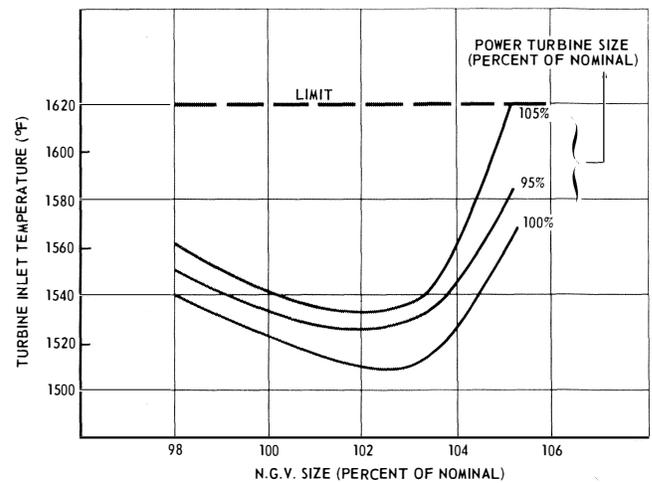


Figure 17. Turbine Inlet Temperature Variation with NGV and P.T. Size.

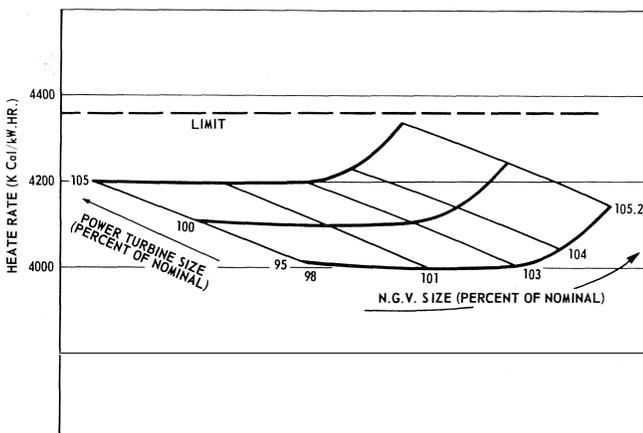


Figure 16. Heat Rate Variation with NGV and P.T. Size.

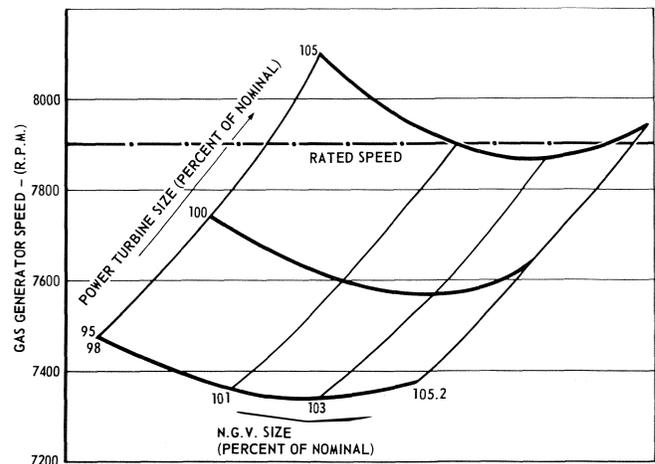


Figure 18. Gas Generator Speed Variation with NGV and P.T. Size.