Design and Test of a Large Two-Shaft Gas Turbine

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The largest two-shaft gas turbine in the United States has been operated in combined cycle service at the Oklahoma Gas and Electric Company Horseshoe Lake Station since May 1963. The same design unit has been operated in peaking and stand-by service at Public Service Electric and Gas (Essex, N. J.) station since December 1963. The O.G & E. unit has a NEMA rating of 25 Mw base load. The PSEG unit has a NEMA peaking rating of 29 Mw. The paper describes the design features of this unit and a method of testing the high-pressure set and variable second-stage nozzle under full-load conditions without actually loading the load turbine.


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Copies will be available until January 1, 1965.
Interest in the combined cycle has been growing in recent years. This cycle, by topping a steam cycle with a gas turbine, continues the improvement in heat rate which is the great achievement of the electric utility industry. The gas turbine described in this paper was designed to support advances in combined cycles by providing a large, flexible machine with modern features.

It is also intended for use in another area of growing interest, gas turbine peak power generation.

Two of these machines are presently in service. The first, rated 25,000 kw, is used in the pioneering combined-cycle plant which the Oklahoma Gas and Electric Company has installed at its Horseshoe Lake station. This plant is the first large combined cycle in the world.

The second machine, rated 29,000 kw, has been installed as an outdoor plant at the Essex (N.J.) station of the Public Service Electric and Gas Company and is used for peak power generation.

A view of one of these machines as it is being readied for shipment is shown in Fig. 1.

DESIGN PHILOSOPHY

At the outset of the design, the following applications were considered:

1 Many variations of such a cycle have been proposed. The most popular so far uses the exhaust of the gas turbine (which contains 75 percent of its original oxygen) as preheated combustion air for a boiler. This cycle has been described in detail by McKone and Sheldon (1) and others (6, 7).

2 Numbers in parentheses designate References at the end of the paper.
TABLE 1 AERODYNAMIC/ THERMODYNAMIC DESIGN

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure ratio</td>
<td>6.6</td>
</tr>
<tr>
<td>Firing Temperature</td>
<td>1500°F base load</td>
</tr>
<tr>
<td></td>
<td>1500°F peak load</td>
</tr>
<tr>
<td>Inlet Air Flow</td>
<td>41.7 lbs/sec, 415 lbm/s, (High flow compressor)</td>
</tr>
<tr>
<td>Number of Compressor Stages</td>
<td>16</td>
</tr>
<tr>
<td>Number of Turbine Stages</td>
<td>2</td>
</tr>
<tr>
<td>Exhaust Temperature</td>
<td>970°F</td>
</tr>
<tr>
<td>Nominal Output</td>
<td>25000 to 18000 HP (Depends on application)</td>
</tr>
<tr>
<td>EGT Conditions</td>
<td></td>
</tr>
<tr>
<td>Thermal Efficiency</td>
<td>11.4% Base load (UNV Natural Gas)</td>
</tr>
<tr>
<td></td>
<td>20.6% Peak load</td>
</tr>
</tbody>
</table>

Fig. 2 Lower half of gas-turbine casing with compressor rotor and high-pressure turbine installed

- Simple-cycle base or peak power generation or mechanical drive
- Combined cycle power generation
  - Exhaust-fired boiler
  - Supercharged boiler
- Regenerative cycle

Major attention was given to design details promoting long life and reliability. This led to substantial construction, Figs. 2 and 3, with maintenance schedules and procedures akin to those of the modern steam turbine with which most potential users are familiar.

Consideration was also given to growth potential. Traditionally, gas-turbine designs have been uprated frequently during their economic life as new materials and design techniques become available. Recognition of this trend early in the design effort enables a design to be made which allows future upratings with a minimum of redesign. This philosophy yields an immediately useful result, yet one that can, for a number of years, be kept modern and made more useful.

CYCLE DESIGN

The cycle design is summarized in Table 1. Some further comments may be of interest.

The compressor pressure ratio, 6.6, is typical of land-based gas turbines. It is selected for maximum specific work (output per pound of air flow) at the 1500°F firing temperature. Increasing specific work increases combined cycle efficiency (2). It also indirectly reduces the cost of simple-cycle machines, in that more output is achieved with essentially the same hardware. The selected pressure ratio is not so high as to penalize the regenerative cycle, whose thermal efficiency drops off with higher pressure ratios.

The compressor design chosen for this machine was already in existence, both in this size and in a smaller scale version. It has very good efficiency at its design point (87 percent), adequate pulsation margin, and much satisfactory operation experience.

The two-shaft arrangement allows flexibility.
for matching boiler air requirements of combined cycles. The load turbine can be kept synchronized and delivering power, and the boiler-inlet gas temperature can be maintained, while the air flow is reduced to follow station load reduction. This results in part-load economy superior to the constant air-flow, single-shaft arrangement. (This feature is useful over only part of the load range; considerable station load variation is achieved by varying the excess air in the boiler) (3). Two-shaft flexibility is also used in the same general way to improve part-load economy in regenerative cycles. It is also important in matching the compressor to the turbine. This compressor is sized to run at 3285 rpm with good efficiency while, of course, 60/50-cycle power requires 3600/3000 rpm for the load turbine. Investigation showed that the two-shaft arrangement delivered more power than the same size single-shaft machine run at 3285 rpm and geared to 3600/3000 rpm at the generator. Such machines have been in service many years, in fact, and provide a good basis for comparison. Of course, the compressor could have been sized to operate at 3600 rpm but the reduced air flow would have resulted in a much lower rating.

The two-stage turbine is in keeping with a long standing preference for high-energy impulse stages. The immediate advantage is reduced bucket metal temperature for a given firing temperature. The greater expansion in the nozzle and the higher bucket velocity reduce the bucket entrance temperature while the greater energy removal in the stage reduces the bucket leaving temperature. A comparison of typical high and low-energy first stage shows, at 1600 F average firing temperature:

<table>
<thead>
<tr>
<th>Pressure ratio on stage (total to static)</th>
<th>Bucket metal temp, deg F</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.7</td>
<td>1458</td>
</tr>
<tr>
<td>2.9</td>
<td>1330</td>
</tr>
<tr>
<td>Difference 128</td>
<td></td>
</tr>
</tbody>
</table>

This temperature difference can mean a 20 to 1 ratio in bucket life or, alternately, be reflected as a higher allowable firing temperature for the high-energy stage.

The second-stage nozzle area is variable. This feature allows adjustment of the pressure ratio split and, therefore, the energy split between stages. This flexibility is used to modify the otherwise fixed relation between speed and temperature of the high pressure set. The variable area nozzle provides:

- Lower starting power requirements.
- Automatic accommodations to ambient temperature changes.

Firing temperature has a powerful effect on the usefulness of a gas turbine. For each 100 deg
of firing temperature, the output from a given machine increases about 15 percent and the thermal efficiency about 3 percent simple cycle to 8 percent regenerative cycle. For this reason, there is always a great interest in raising firing temperatures as both the manufacturer and the customer benefit economically. For a given balanced mechanical design, the firing temperature is limited by available materials. The firing temperature chosen for this gas turbine represents the optimum for the materials available and tested at the time the design was established. However, a continuing program to raise this temperature has been successful.

**GENERAL ARRANGEMENT**

Fig. 4 shows the cross section of this machine. The following additional information is of interest:

- Number of struts in inlet = 5.
- Number of combustors = 12.
- Number of 1st stage nozzle partitions = 48.
- Number of 1st stage buckets = 168.
- Number of struts in interstage = 12.
- Number of 2nd stage nozzle partitions = 36.
- Number of 2nd stage buckets = 102.

It will be noted that the combustion is folded well back over the compressor to reduce machine length, while the combustion liners are accessible for maintenance through covers in the forward bulk-
The housing for the interstage bearings (Nos. 2 and 3) is supported from the turbine shell by ten radial struts which provide centerline support to maintain alignment. The top and bottom positions are occupied by cooling and sealing air and lubrication piping.

The inlet and exhaust hoods, shown up and down, respectively, can be set in many radial orientations to fit application needs.

**BUCKET SHANKS AND DOVETAILS**

The first-stage buckets have shanks almost as long as the vanes. There are two reasons for such long shanks: to isolate the wheel rim from the gas path, thus keeping it cooler, and to control the vibratory response of the bucket.

Fig. 5 is a model illustrating the first-stage wheel rim with three buckets and their cover plates, seal pins and a retaining segment. An important innovation here is the support of the cover plates in the wheel dovetail (4) rather than by the bucket platform. The cover plates carry the wheel pocket seal teeth so that the rim of the wheel is three pockets away from the gas path. Cooling air is passed outward along the wheel and cover-plate surfaces, from pocket to pocket. Excellent isolation of the wheel rim from the hot gas is achieved; a 500 deg F temperature drop from the platform to the dovetail has been measured.

The arrangement at the bucket platform, with seal pins between each platform pressed outward by centrifugal force, gives an effect similar to the tie wires sometimes used to control bucket vibration. Damping is contributed by the seal pins and the shank proportions have been selected to turn the natural frequencies of the bucket away from stimulus frequencies existing in the gas stream.

The second-stage buckets are similar in construction although the vanes are much longer and hence require tie wires for vibration control.

**ONE-PIECE SHROUD HOLDER**

Control of tip clearances is a classical problem in turbomachinery. The transient growth of the rotor and the stator during heating or cooling must be matched, and the stator must remain round, to allow tight tip clearances. Substantial casings with heavy sections, split and flanged at the horizontal joint, are slow to heat up and must not be assumed to remain round. The solution employing a one-piece shroud holder (5) centerline supported in the turbine shell was used. This holder responds more rapidly than the casing to gas-temperature changes (by a factor approaching 30), and, as the cross section is circumferentially uniform, is not subject to out-of-roundness distortion.

It will, of course, be somewhat distorted by circumferential temperature variation. This was evaluated using a model in which a nonuniform ring of gas flames was played on the inner surface causing 100 deg F local temperature variation. This presented a much greater temperature variation than encountered in normal gas-turbine operation. The radial distortion was found to be only 0.15 percent of diameter and the distortion disappeared completely upon cooling, indicating thermal stress within the elastic range.

To control the heat input to the shroud holder, insulation is placed between the shroud segments and the holder. This not only reduces any accidental circumferential temperature gradient but also the radial gradient and the resulting stress.
TURBINE COOLING

Generous use of cooling air to control metal temperatures in the turbine region is shown in Fig. 7. It will be seen that two cooling-air circuits pass through the bore of the turbine wheels. This not only simplifies the piping but is used to control thermal stress during starting. During startup, the cooling air is actually warmer than the metal and thus heats the bore at the same time the rim is being heated transiently. This permits rapid starting by limiting the transient radial temperature gradient of the turbine wheels. Following startup, the wheel becomes hotter than the air and the normal cooling function is accomplished.

BEARINGS AND ROTOR ARRANGEMENT

Each shaft of the gas turbine is supported on two bearings. The gas generator set journals run in conventional three-lobed babbitt bearings. The load shaft runs in tilting-shoe bearings which have babbitt faces. This latter arrangement is used because the load pressure on these bearings varies considerably due to changes in relative vertical expansion of the turbine and driven machine and may become quite low under some conditions. The tilting-shoe bearing does not become unstable when lightly loaded.

Kingsbury thrust bearings are used on each shaft, as can be seen in the cross section, Fig. 4.

Materials

The casing materials vary from gray cast iron in the colder part of the compressor through nodular cast iron, carbon-steel fabrication to cast carbon steel at the turbine shell. The exhaust hood and skin are carbon steel fabrications.

The first-stage nozzle is cast segmented X-45. The second-stage nozzle partitions are type 310 austenitic steel.

Fig. 7 Cooling-air circuits. Numbers refer to compressor stage from which air is extracted.

Fig. 8 Results of factory test of Model 8362 gas turbine. Locus of normal second-stage nozzle area would have been the only operating points available if the nozzle were not variable.
The first-stage buckets are M252 alloy; the second stage, A-286. U-500 first-stage and S-590 second-stage buckets are used for higher temperatures and ratings. The wheels and other rotor parts are Cr-Mo-Va steel.

FACTORY TEST

Because this is a new design, it was desired to subject the first machine to a rigorous factory test. A test facility capable of absorbing full output was not available. A consideration of the likelihood of exposing design deficiencies led to the conclusion that the gas-generator set (the compressor, combustion and first-stage turbine) included all of the novel design features so that a full load (i.e., full temperature) test of this portion would be of great value. A scheme was devised wherein the load turbine and shaft were omitted and temporary flow straighteners were installed to remove the whirl introduced by the second-stage nozzle. Because this nozzle is variable, it can be used to adjust the back pressure on the first stage to simulate any desired load. Adjustment is automatic since the gas-generator set governor normally positions this nozzle to hold speed at the set point. By setting called-for speed and called-for exhaust temperature, then, the gas-generator set was made to operate over its full normal range and did not "know" that the load turbine was missing.

The load was put up the stack in the form of hot gas instead of mechanical shaft power at the load shaft. Consequently, the stack-gas temperature (about 1200 F) was above the design limit of the existing stack. To cope with this problem, a series of water-spray nozzles was installed behind the flow-straightener blades.

Evaporation of water spray in a fast-moving stream requires considerable flow length because the droplets quickly acquire full stream velocity, reducing the heat-transfer rate. However, it was felt that the stack surfaces would be protected by impingement of the water droplets in regions where the stream was not already cooled by evaporation. It was not determined during the test which phenomenon was active but no damage was done to the stack or silencers.

Of course, without mechanical load measurement, only limited performance data could be gathered. However, the flow capacity, heat consumption, compressor, and first-stage turbine efficiencies were measured and compared with predicted values as shown in Fig.8.

As a proof test of the operation of the gas turbine and its accessories, this test was very valuable, as many minor troubles were identified and corrected. Development of the control system was completed during factory test. A reduction of the first-stage tip clearances was made between factory tests as a result of correction of any assembly error in the first-stage shroud. First-stage turbine-nozzle leakage was discovered and corrected.

It will be noted that this test method is similar in many respects to the static testing of aircraft gas turbines.

CONCLUSION

The general description and major features of this new design and its factory testing have been described. Two of these machines are in service, and hopefully, user experience will be the subject of future papers. Such information is of great value in advancing the art of designing this interesting and useful type of prime mover. We can all look forward to continuing advances and new applications.

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2 Ibid., pp. 362, 363.
3 Ibid., p. 366.
4 US Patent Pending.
5 US Patent Pending.