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TESTING THE MODEL V84.3A GAS TURBINE - EXPERIMENTAL TECHNIQUES AND RESULTS

BREAK

W. Boehm, D. Raake, D. Regnery, J. Seume, and K. Terjung

Gas Turbine Quality Management
Siemens AG, KWU Group,
Berlin and Muelheim / Germany

1 ABSTRACT

Like its predecessors, Siemens' first Hybrid Burner Ring® machine (Becker et al., 1995) underwent an extensive experimental development and test programm prior to its introduction to the market.

The present paper describes the philosophy of prototype testing, some of the challenges encountered in instrumenting the machine and the methodology and experimental techniques used to solve the problems. In particular, it describes some of the advanced implementations of blade vibration measurements, optical pyrometry and other temperature measurements, and techniques to determine efficiency.

The experimental results confirm the design of the model V84.3A gas turbine and show some potential for further improvements.

2 INTRODUCTION

The prime objectives of the current engineering of heavy-duty industrial gas turbines are:

- high efficiency
- high turbine inlet and hence outlet temperatures
- low emissions
- high reliability and availability
- low specific cost through high power density

The Model V84.3A gas turbine achieves these objectives using prototype testing as a central engineering tool, providing ...

1. quality control for the engineering process through verification of
   - performance predictions
   - controls
   - manufacturability and functionality of design specifications
2. one additional design optimization loop before delivery to the customer, thereby reducing the risk of prototype commissioning for the customer
3. quality assurance for the hardware of the prototype gas turbine and subsequent machines
4. a test bed for component R&D for future generations of gas turbines, testing new components under the most realistic conditions possible i.e. in the machine
5. Empirical data bases for the calibration of component predicition tools under gas turbine operating conditions

The result is a machine that proved 38% simple-cycle (58% combined cycle) efficiency and less than 25 ppm NO_x.

3 TEST PROGRAM AND TEST FACILITY

3.1 Test bed vs. power-plant testing

To all customers, the real test of power generating equiment is the long-term operation at their power plant. Customers do, however, want to reap the benefits of rapid innovation and of the improvement of machine performance. Therefore, the performance and reliability of improved machines must be verified faster than is possible during normal power-plant operation.

Siemens' answer to this challenge is to operate the first machine of each type on its test bed under abnormally severe conditions and beyond the operating range guaranteed to customers. These conditions could not be achieved during full-load operation synchronized with a stable grid in a power plant.

An example of such asynchronous operation is the mapping of the compressor performance and the determination of the surge margin. Extreme operating conditions of the compressor e.g. low grid frequency and high compressor inlet temperature may only be encountered after years of power-plant operation. On the test bed, this severe operating condition is intentionally created by lowering the rotor speed below the guaranteed limits, thus moving the compressor closer to the surge line. To maintain a sufficiently high

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pressure ratio, this test must be carried out under load. The same is true for speed variations to test out blade vibrations.

The rotational speed is the degree of freedom necessary to run the gas turbine at the extremes of the operating range, making the world's largest dynamometer the heart of the test facility.

3.2 Test facility

The prototype tests are conducted at a dedicated facility in the Berlin gas turbine manufacturing plant (Figure 1 and Figure 2). The gas turbine is started with a starting motor which is coupled to the load with a hydraulic torque converter. The variable-speed dynamometer mentioned above is a water friction brake. Figure 3 shows the open casing of the water brake with the six-disc rotor during maintenance. During the V84.3A tests, this load was used up to 180MW.

During routine testing, the rotational speed is varied between 92% and 108% of design speed at various load levels up to base load. During surge testing the rotational speed may be lowered further. The torque of the brake is varied by individually modulating the flow of cooling water in the chambers surrounding the six discs.

The variable speed load is not only critical for mapping the compressor performance and for the determination of the surge margin but also for measuring design-point compressor aerodynamics at off-design ambient conditions (Janssen et al., 1993) and verifying the dynamic response of the rotor blading. The latter is also done between 92% and 108% of design speed at various load levels up to base load.

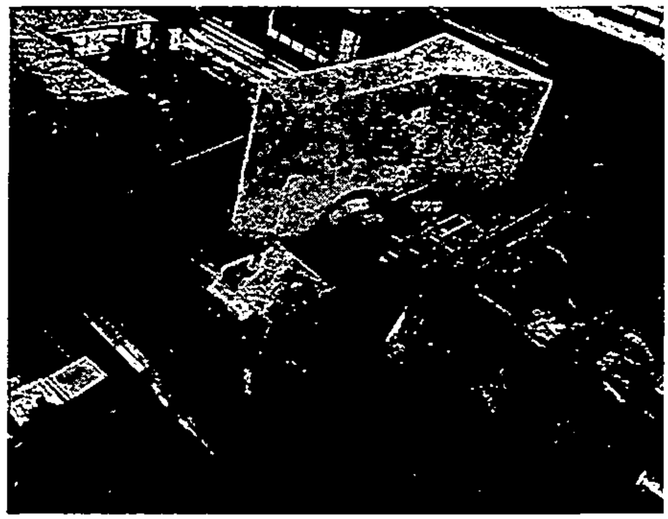

Figure 2: View of the test facility

The test facility also provides the infrastructure necessary for the extensive instrumentation required for detailed diagnostics. More than 300 pressures and 500 temperatures are acquired at 3 second intervals along with the gas turbine's torque, valve positions, and the flow rates of cooling and purging air, water for purging and injection, fuel, and lubricating oil.

Additionally, signals from the control system and the most

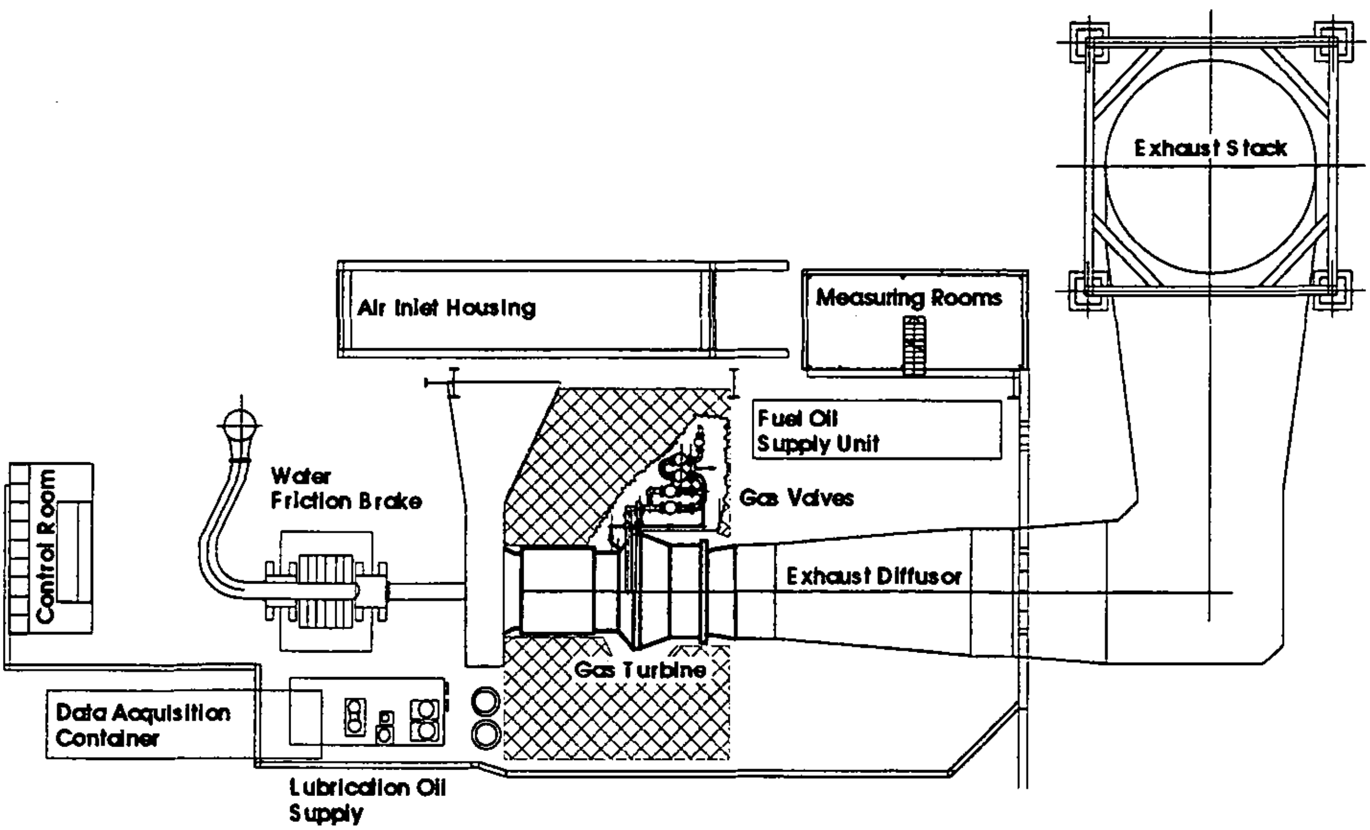

Figure 1: Layout of the test facility

important operating parameters are acquired at 20 Hz. A video flame observation system monitors the flames and heat shields in the combustion chamber via air-cooled endoscopes.

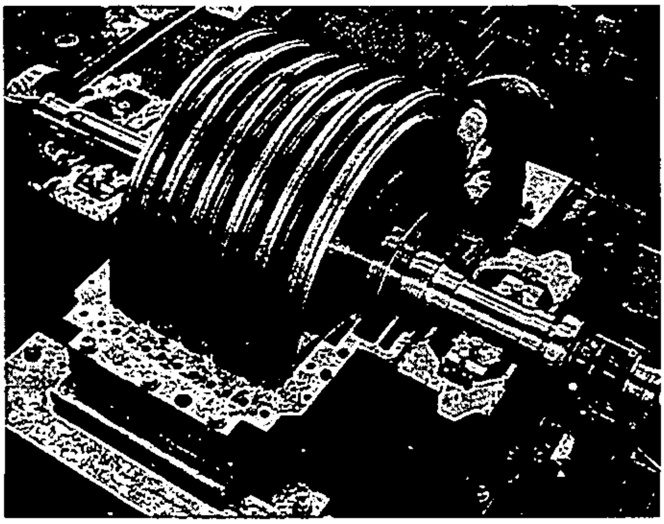

Figure 3: Water breke with six-disc rotor during maintenance

3.3 Test program

To achieve the objectives outlined above, a prototype test program is typically structured as follows:

1. commissioning (fuel: natural gas)
2. performance test at base load
3. verification of safe operation
   - Speed variations assure low stresses in the blading throughout the operating range.
   - Tip clearance measurements verify running clearances.
   - Turbine vane and rotor blade temperature measurements verify proper cooling.
4. combustion optimization on natural gas
5. commissioning for operation with light heating oil, combustion optimization
6. fuel switching, optimization of oil operation, oil starts
7. detailed measurements of component performance, e.g. flow field measurements in the compressorand the turbine and mapping of the surge line

The experimental techniques used in this gas-turbine development program for the investigation of compressor aerodynamics (with pitot tubes, three-hole and five-hole probes) and tip clearances (with electro-mechanical sensors and abrasion pins) have been reported in their application to the model V84.3 (Janssen et al., 1993 and 1994). The present paper will concentrate on techniques which were developed or refined for the experimental support of the V84.3A program.

4 MEASURING POWER AND EFFICIENCY

Power and efficiency are the key quantities for the assessment of the overall performance of the machine. In the absence of a generator, power is obtained by measuring the torque to the water brake and the rotational speed.

Torque: The power output to the water brake is determined using a strain-gage installation on the torque-measuring shaft connecting the gas turbine to the load. The strain-gage signal is transmitted via a telemetry system to the data acquisition system. Figure 4 shows the calibration set-up used to calibrate the strain-gage torque meter. The torque is created by four hydraulic cylinders whose force is measured with in-line force transducers. The fixture transmits the forces of these four actuators to both ends of the shaft. No torque is exerted on the shaft of the gas turbine or the shaft of the water brake. Due to the accuracy of the force measurements, the rigidity of the fixture, and the accuracy of the measurement of the effective radius of each actuator, the torque can be calibrated to within ±0.33% of the value measured at base load. 90% confidence intervals are quoted for random errors throughout this paper.

Shaft frequency: The rotational frequency of the shaft is measured with a conventional magnetic pick-up to ±0.13%.

Power is consequently determined from torque and frequency within ±0.35% at base load.

Fuel flow: During operation on natural gas, the mass flow rate is determined with a conventional orifice meter with an accuracy of 0.8%. The mass flow rate of fuel oil is measured with a Coriolis-type meter to 0.15%.

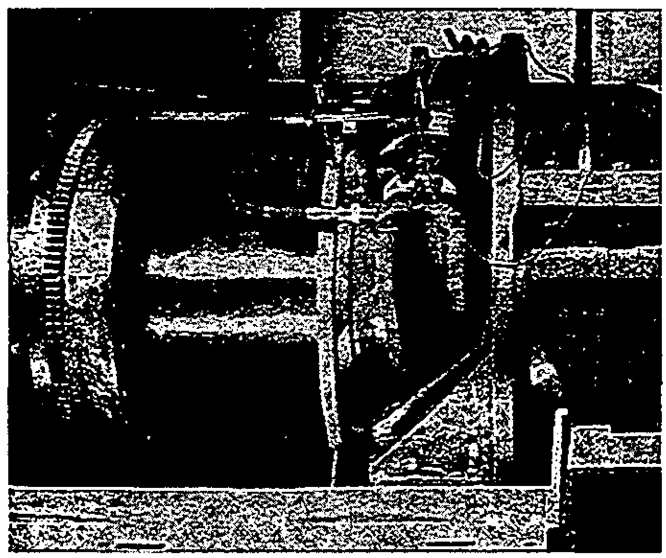

Figure 4: Calibration set-up used to calibrate the strain-gage torque meter

5 BLADE VIBRATION MEASUREMENTS

During the verification of safe operation, blade vibrations are measured continuously as new parts of the operating range are explored. Compressor and turbine blades are highly loaded by centrifugal stresses, high temperature, aerodynamic forces (static and dynamic), and varying operating conditions. To ensure safe operation and to validate the design calculations, blade stress-levels are evaluated during prototype tests covering and exceeding the whole range of operating conditions, i.e. between 92% and 108% of design speed at various load levels up to base load.

Selected blades in seven compressor stages and all turbine stages are equipped with high-temperature strain gages. To gain the best results for all mode shapes of interest, the position of the gages ist deduced prior to instrumenting the rotor from a strain distribution on the blade surface which is determined experimentally and analytically.

For this purpose, blades which are equipped with 50 to 100 strain gages are mounted into a massive steel block and are excited with a shaker. The strain distribution on the blade and the root is measured for every vibrational mode up to a limit frequency. Additionally, the mode shapes are visualized by means of holography (Schönebeck et al., 1984). With the help of this preliminary analysis, the number of strain gages used in the machine is reduced to less than seven per blade.

Vibrational data from all blades of some of the stages are collected with the non-contact blade vibration measurement system.

5.1 Application of sensors and data acquisition

The techniques for fixing a strain gage on a blade are glueing, the flame-spray method and welding of encapsulated gages, the latter of which has proven to be most reliable and is preferred (Schönebeck et al., 1984 and Böhm and Terjung, 1987).

High-temperature strain gages are spot welded onto the blades enabling measurements even in the first turbine stage. The gages are equipped with integrated leads and a metallic shield. The routing of the cables from the gages to the ends of the rotor is determined by the rotor design and strength considerations since extra holes have to be provided. The installation is performed during the stacking of the rotor.

Near the ends of the rotor, the leads are connected to the signal transmitters with flexible wires. To transmit the signals out of the rotating system, two different systems are used at the two ends of the rotor. As the shaft is coupled to the water friction brake at the cold end, no free end is available here. The signals from the compressor region are therefore transmitted by telemetry and the power for the gages and the transmitters is supplied inductively (Figure 5).

At the hot end of the turbine, a rotating power supply and signal conditioning unit is mounted. This unit connects to a slip-ring transducer, which is housed in a cylindrical casing extending the hub of the turbine bearing support. A multiplexing telemetry system is used to collect data from quasi-static transducers (Figure 6).

The dynamic signals, together with some machine operating parameters such as rotor speed, mechanical output, and inlet guide-vane position are recorded with multi-channel tape-recorders. All of the signals are monitored on-line. Some selected signals are analyzed in detail on-line with a specially designed computer system. The latter is used for comprehensive off-line analysis as well.

The monitoring gives an immediate feedback of the blades' answer to the different operating conditions which is essential for critical tests. The detailed analysis shows the influence of different parameters on the eigen-frequencies and the excitation level for the different mode shapes of the blading thus permitting an analysis of the blade loading for any operating condition. Data from the static transducers in the rotating system is acquired, processed and visualized on a PC system (Figure 7).

5.2 Non-contact blade vibration measuring system

Measuring blade vibrations with strain gages yields information about every mode shape affected by the operating conditions but is restricted to very few blades in every stage due to costs and signal transmitting capacity. To overcome these restrictions, dynamic blade tip deflections of all the blades in a row are investigated optically.

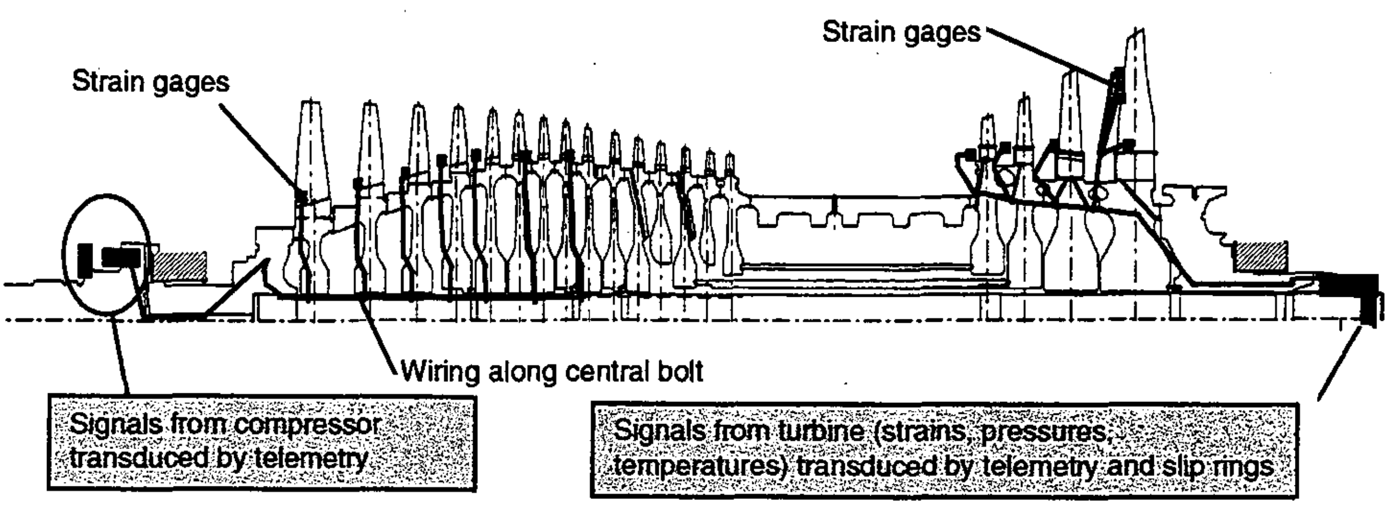

Figure 5: Schematic of the signal path in the rotor

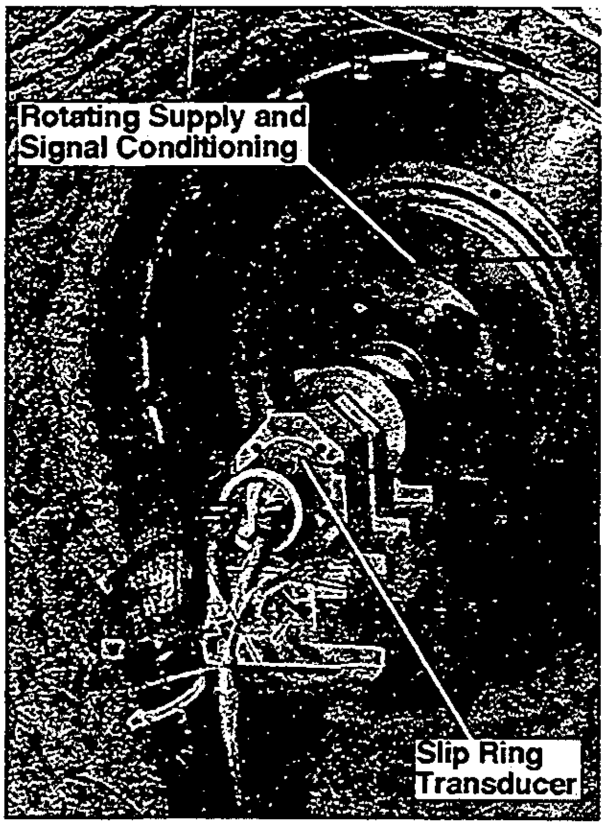

Figure 6: View of signal transducing unit at the turbine end of the rotor

The time a single blade needs to pass between two sensors is measured with high accuracy and is related to the time a non-vibrating blade should need which is calculated from the rotor speed measured with a third sensor and some geometrical input (Figure 8). The time difference gives a value for the instantaneous blade deflection due to vibration. Acquiring the data for several revolutions gives a complete image of the vibrational behaviour for the fundamental mode shapes of every blade in the row. Mode shapes of higher order than four or five cannot be detected as they cause almost no blade-tip deflection (Gloger, 1990).

The third sensor not only serves as a speed sensor but gives a trigger signal to clearly identify the individual blades and particularly those equipped with strain gages. Thus the correlation between blade strains and dynamic blade tip deflection (Figure 9) under centrifugal stresses can be evaluated experimentally.

Data are acquired and processed on a specially designed unit which enables on-line visualization of the vibration amplitude of all blades. The mode shapes are analyzed off-line.

The sensors are cooled fiber-optical probes which have proven to work reliably up to gas temperatures of 800°C. Based on several years of operating experience, this system has been developed into the BeSSI blade vibration information system. It is now in commercial use for monitoring purposes in several power plants (Gloger, 1995).

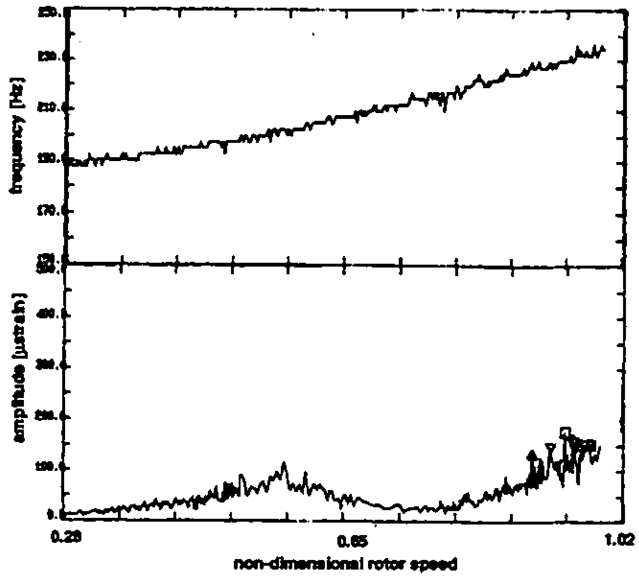

Figure 7: Amplitude and frequency of a vibrational mode versus rotor speed

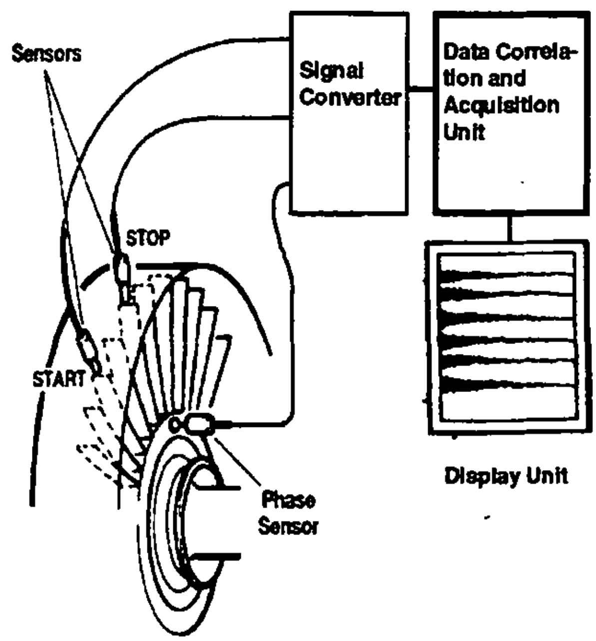

Figure 8: Schematic of the non-contact blade vibration measurement syatem

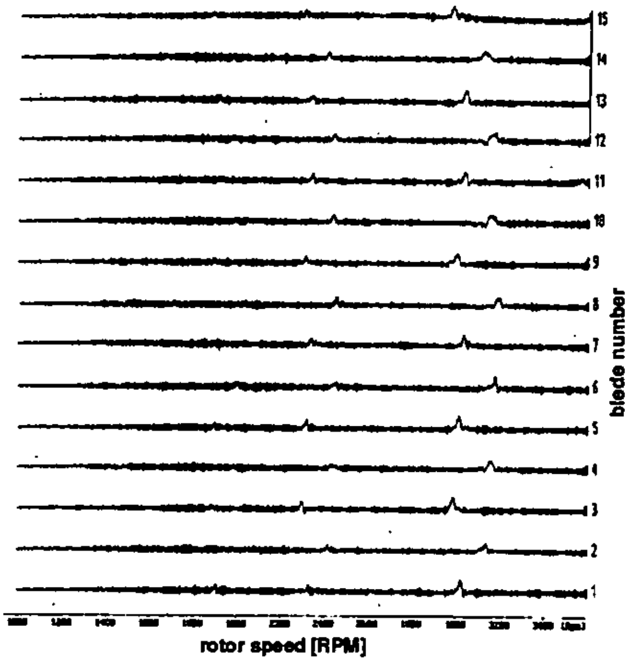

Figure 9: Blade tip vibrational amplitude vs. rotor speed measured with the non-contact system

The parallel usage of strain gages and non-contact techniques permits the use of only 6 instrumented blades per stage because the variability of the blade response may be determined from the optical measurements.

6 THERMOCOUPLE INSTALLATION AND MEASUREMENT

Due to the inherently rigid design of the vanes, mechanical loading is of no concern in the stator of the compressor or turbine. Thermal loading of the turbine vanes, however, is of great importance for an optimization of the cooling air used. Therefore, the surface temperatures of the vanes must be monitored over the wide range of operating conditions of the cooling system mapped out in the test program.

The measurement of surface temperatures in film cooled vanes requires novel installation techniques. The actual junction of the thermocouple must be positioned close to the surface at a well-defined location. The installation must avoid a distortion of the surface contour and consequent disturbance to the flow of hot gases or cooling air. The heat flow in the wall of the blade to the thermocouple must also remain undisturbed. These goals are achieved by machining the tips of thermocouples to defined dimensions. The tips are subsequently brazed into the vanes and finally machined to conform to the surface contour. With this installation technique, the precise position (±0.075mm) of the junction beneath the surface is known and thus the surface temperature of the vane can be calculated.

The true gas temperature upstrem of the first vane is measured directly, allowing energy closure for the calculation of the ISO turbine inlet temperature. The gas temperature is measured with Type S thermocouples in a platinum sheath protruding through the cooling film into the hot gas near the leading edge.

7 PYROMETRY

7.1 Uses of infrared thermometry

Infrared thermometry is a proven technology for non-intrusive measurements of surface temperatures in gas turbine enviroments as described in the early years of application of this technique in gas turbine e.g. by Atkinson and Guenard (1978), Douglas (1980), Benyon (1981) and in the last years by Schulenberg and Bals (1987) and Sellers et al. (1989). Research and development on the application of this measurement technique to gas turbines was reported in the past e.g. by Barber (1969), Rohy and Compton (1973), and in the recent past by Schenk et al. (1994) and Moon et al. (1995). The principal focus of research work on the use of infrared thermometry in gas turbines in the recent years was the development of computer models for the application and evaluation of this technique e.g. by De Lucia and Lanfranchi (1992) and De Lucia and Masotti (1995).

Siemens has been using turbine pyrometry for more than a decade in developing gas-turbine blade and vane cooling technology including the investigation of various designs and of the effects of manufacturing techniques (e.g. laser-drilling vs. EDM).

7.2 Experimental setup

The investigations are carried out with custom-made water-cooled pyrometer probes containing a 90° deviation mirror at the end of the probe. The sighting tube of the probe is purged with nitrogen during the operation of the gas turbine to reduce contamination of the mirror surface. The probe contains Land TBTMS-R7QB thermometers with a silicon detector operating at 1μm wavelength for the infrared temperature measurement. The probe is mounted in a probe traversing unit with two degrees of freedom (axial and circumferential, with respect to the probe axis). The traversing devices are bolted to flanges welded to the turbine casing. The probes penetrate the casing and the vane carrier radially (with respect to the machine). The dynamic seal between the cooling air and the ambient is provided by packings in the traversing device. The static seal between the hot gases and the cooling air is provided by metal bellows.

7.3 Blade and vane temperatures

Figure 10 shows a part of the typical blade temperature distribution of the first stage of the rotor under a certain observation angle. The temperature peaks occuring in this distribution indicate the differences in the cooling air supply of the blades between the normal set and some test blades which differ in the film cooling pattern and in the manufacturing process.

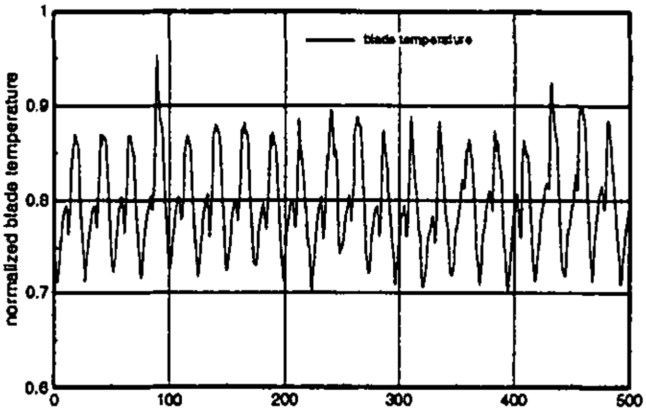

Figure 10: Typical blade temperature distribution for 20 first-stage blades

The obstruction of the flow channel by the pyrometer probe causes periodic fluctuations of the pressure acting on the downstream blading. This excitation of blade vibrations does not lead to excessive amplitudes. A complete mapping of blade temperatures requires an insertion time of 90 seconds. For this duration, slightly elevated blade vibration can be tolerated.

Figure 11 shows typical radial vane-temperature profiles of the suction and the pressure side of the second stage for two observation angles. For such measurements, the probes are positioned just upstream of the leading edge of a vane and downstream of the trailing edge of a different vane. From here, the pyrometers map the temperatures on the surfaces of adjacent vanes.

For the measurement of blade temperatures, the obstruction of the flow channel by the probe is not relevant. During measurements of vane temperatures from a pyrometer positioned upstream, however, the blockage of the flow channel by the probe alters the heat transfer and consequently the temperature distribution on the surface of the vane. Therefore, the surface temperature of the vanes cannot be determined under steady-state conditions.

Changes in surface temperature do not occur spontaneously with the insertion of the probe. The thermal time constant is approx. 1 second for the measurements of interest. With the present technique, the probes are inserted as far as the full depth of the channel in 1/3 of the time constant. During this insertion, the position and the temperature signal are acquired simultaneously. After the near-hub position is reached, the probe is withdrawn quickly and rotated to a new view-angle. After a waiting period of approx. 4 time constants, the procedure is repeated for the new view angle. Thereby, the radial temperature is mapped for several view angles within the confines of the visible surface.

A difference exists between the non-intrusive surface temperature measurement with the infrared thermometer and the reading of a thermocouple, installed in a finite distance from the vane's surface at approximately 65% channel height. It is due to the temperature gradient in the solid and radiation effects. An analytical correction taking these factors into account yields a favourable agreement (Figure 11).

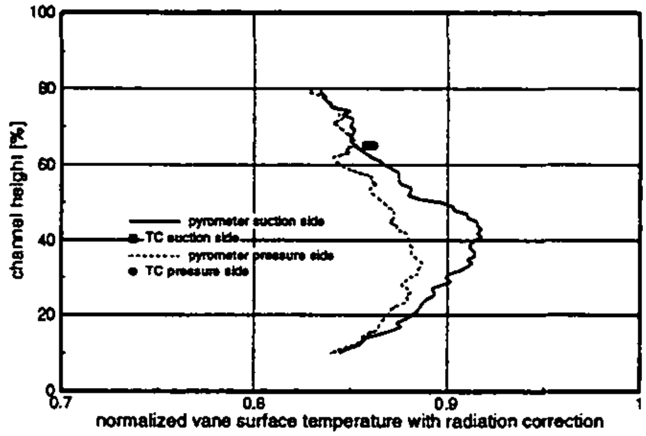

Figure 11: Typical vane temperature-profile of the second stage for two viewing angles

7.4 Application to monitoring

At this test facility, optical pyrometry is a well-established experimental tool for the verification and support of turbine blade engineering. This experience has benefitted the development of systems for the monitoring of blade temperatures and blade cooling such as the LAND 'TBTMS' system which is being offered for the use in Siemens gas turbines.

8 MEASUREMENT OF RADIAL TEMPERATURE DISTRIBUTIONS IN THE TURBINE

To compare the numerically predicted gas temperatures to the actual profiles in the turbine, the radial temperature distributions were measured in various cross-sections of the turbine. These measurements were also carried out to provide boundary conditions for the design of the cooling system of blades and vanes. A typical radial temperature profile upstream of the second stage of the turbine is shown in Figure 12.

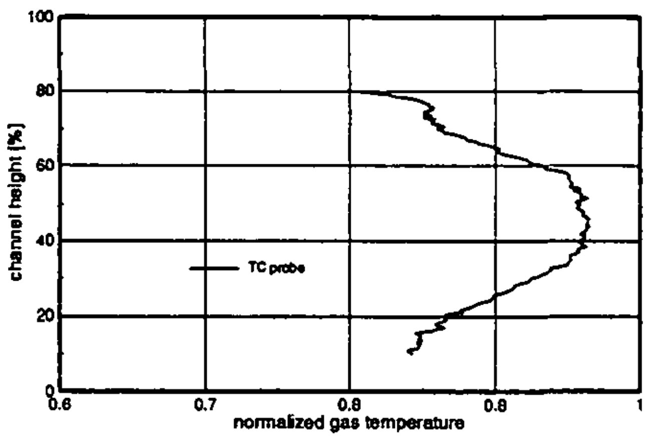

Figure 12: Radial temperature profile up-stream of the second stage

In designing the turbine blading, Siemens adapted Pratt & Whitney's aeroengine technology. The gas and surface temperature measurements show that thus near-optimal vane and blade cooling in the turbine was achieved. On one occasion, the measurements pointed to the potential for a further reduction in cooling air.

9 CONCLUSIONS

With the Model V84.3A Hybrid Burner Ring® gas turbine, Siemens introduced the most efficient (38% simple cycle) among the heavy industrial gas turbines tested to date by judiciously introducing Pratt & Whitney aeroengine technology into the proven robust design of Siemens heavy duty machines without resorting to quantum leaps in technology. Detailed measurements of component performance show that the overall efficiency was attained by close attention to component design such as the aerodynamics or the cooling design discussed above and consequently high component efficiencies.

Component optimization in the prototype machine is based on detailed instrumentation, some of which is implemented in a novel fashion. Examples are a fast-moving optical pyrometer probe for the mapping of vane temperatures from up-stream and the installation of thermocouples with close attention to geometry in regions of high temperature gradients in turbine vanes.

Measuring results are validated based on redundant measurements:

- blade vibrations by strain gages and optical techniques
- radial tip clearances by contact sensors and abrasion pins
- blading temperatures by optical pyrometer and thermocouples

The test facility permits testing the machine throughout and beyond the permissible operating range, primarily by varying the rotational speed over a wide range at loads up to base load with the water friction brake. Thereby, the test bed provides more severe operating conditions than encountered during typical power plant operation in a short period of time. This benefits the customer by providing proven technology despite fast innovation.

10 ACKNOWLEDGEMENTS

The authors gratefully acknowledge the contribution of their colleagues at the test facility who made the tests possible, their fellow test engineers and technicians who helped to carry out the measurements, the gas turbine engineering team who designed this successful machine, and the many colleagues at the Berlin plant whose dedication and hard work made the machine itself.

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