THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
345 E. 47th St., New York, N.Y. 10017

94-GT-319

Printed in U.S.A.

THE MODEL V84.3 SHOP TESTS: TIP CLEARANCE MEASUREMENTS AND EVALUATION

M. Janssen, J. Seume, and H. Zimmermann
Gas Turbine Technology
Siemens AG, KWU Group
Ruhr, Germany

ABSTRACT

The design of high-performance gas turbines requires the reliable prediction of blade tip clearances. Excess clearances allow a portion of the hot gas to flow over the blade tips without performing useful work. The tip leakage flow disturbs the flow field which results in additional losses. Moreover, insufficient blade tip clearance may cause interference which can reduce turbine life. In conventional turbomachines, the blade tip clearances vary markedly with the operating condition of the turbine, essentially as a result of variations in gas temperatures and rotor speed.

Siemens tests prototype gas turbines in its own test facility. An extensive experimental program is devised to verify design calculations regarding strength, aerodynamics and thermodynamics. Among other measurements, the minimum operating tip clearance is measured by abrasion pins. Electro-mechanical sensors measure transient tip clearance during a selected duty cycle consisting of turning-gear operation, cold start, idle operation, as well as part-load, full-load, and most importantly, hot-start. In the present paper, the compressor and turbine tip clearances measured during such a load cycle are compared with calculated predictions. The experimental instrumentation for the prototype gas turbine, as well as design calculations, are presented.

The results show that the new Model V84.3 gas turbine does not exhibit critically small clearances during cold start nor during hot-start due to the careful matching of magnitude and the time constants of the thermal expansion of the blades, discs, blade-ring carriers and casing.

1. INTRODUCTION

Radial clearances between rotating and stationary components in gas turbines are strongly dependent on the load cycle. A load cycle consisting of cold start, partload, intermediate cooling, hot-start, part-load and cooling-off is used to assess the critical condition of hot-start. For compressor and turbine, the hot-start is typically expected to lead to the most critical situation, since in general the blade-ring carrier and blades cool down much faster than the discs, reducing the clearance. This clearance will be used up during hot-start due to immediate centrifugal load.

The instationary thermal behaviour in compressor and turbine is different. Figure 1 shows the time dependent radial clearances for an assumed load cycle.

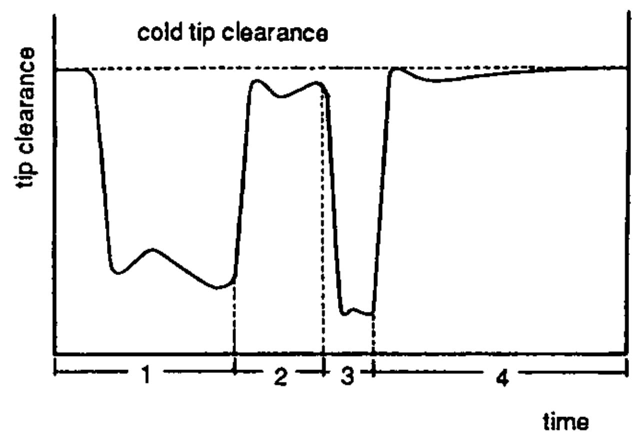

FIGURE 1: TYPICAL LOAD CYCLE:
1=COLD START, 2=INTERMEDIATE COOLING, 3=HOT-START, 4=COOLING-OFF.

Presented at the International Gas Turbine and Aeroengine Congress and Exposition
The Hague, Netherlands — June 13–16, 1994

1. Cold start, part-load: After starting the gas turbine, the centrifugal force on the rotor and the blade reduces the clearance values until running speed is reached. In the example shown the elongation of the blade-ring carrier is bigger than the sum of the elongations of the blade and the disc. The stationary operating clearance is reached after a certain time of full-load operation.

2. Intermediate cooling: Shutting down the gas turbine, the centrifugal load vanishes immediately. Depending on the thermal behaviour of the single components the tip clearance moves towards the cold clearance. The machine is kept in turning-gear operation for a predefined period of time until the hot-start.

3. Hot-start, part-load: In the chosen example the clearance is smaller than during the cold start because the disc remains at a higher temperature than the other components. Therefore, the immediate centrifugal load results in the minimum clearances.

4. Cooling-off: The clearances change in the same manner as before during intermediate cooling.

There are some differences in the elongation behaviour of the components of compressors and turbines. Therefore different clearances are chosen. Turbines have to suffer much higher temperatures than compressors. Thus, component-specific cooling is used to render clearances less dependent on the temperature of the working gas.

A sufficiently large clearance value during hot-start is as important as a small steady state operating clearance. The latter will influence the efficiency of the whole turbine tremendously. Tip clearance flow influences the secondary flow as described by Hah (1986) and Lakshminarayana (1970). The importance of an optimum tip clearance in compressors and turbines is shown by Peacock (1989). They show also that in some current large-scale compressors, it is reckoned that a tip clearance enlargement of 1% of blade height yields a 1.5% drop in efficiency. Turbines are designed with a much wider range of reaction than compressors. This and the higher pressure ratio across a turbine stage means that rotor tip pressure difference is likely to be greater, Peacock (1989). The method applied in the design of advanced turbomachinery is the reduction of the different thermal expansion of rotor and casing by active (e.g. impingement cooling) and passive clearance control (e.g. compensating casing). For controling clearances different measuring techniques have been developed and applied in the past.

Currently, rub pins, electro-mechanical pins, electric probes and laser-optical probes are available for measuring the clearance between rotating and stationary components in gas turbines. Rub pins are soft metal pins which are installed at the measuring point and record the closest point of approach. Electro-mechanical pins are driven by a motor making contact with the blade tip at different operating conditions. Electric probes are both inductive and capacitive probes driven by oscillators. Laser-optical probes are based on light reflection transferred by fiber optic cables. Rub pins and eletromechanical pins were chosen for the present study.

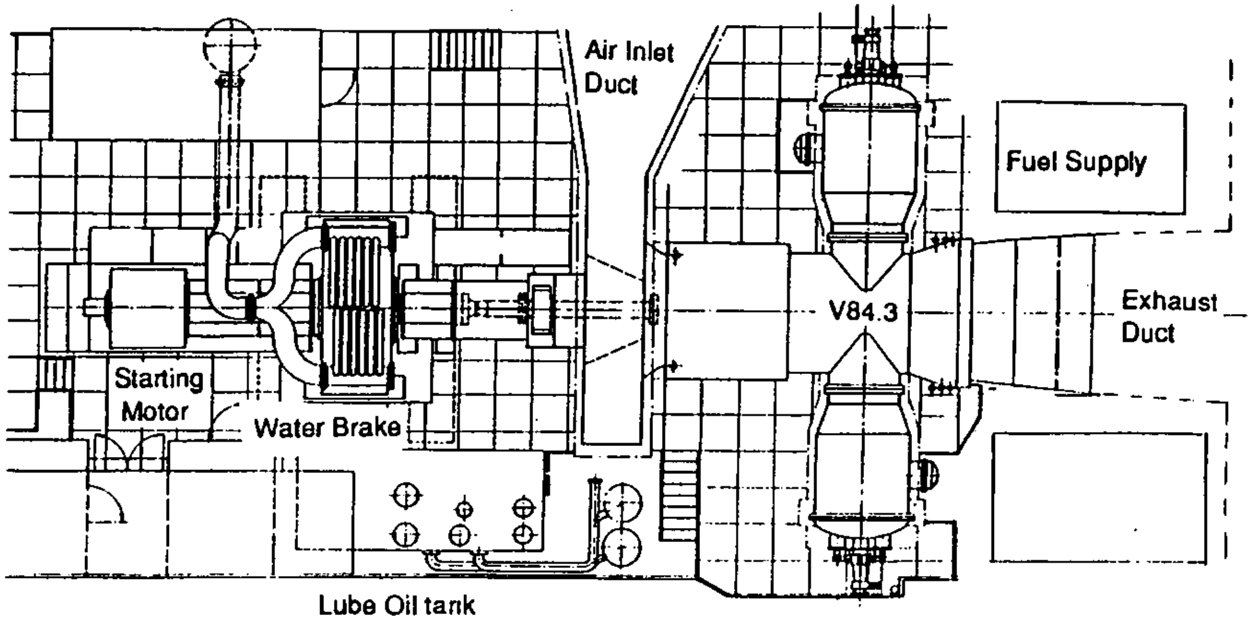

FIGURE 2: LAYOUT OF THE SIEMENS / KWU TEST FACILITY.

2. MODEL V84.3 PROTOTYPE-TEST

From April 1992 through February 1993, a series of prototype tests was conducted on the new-generation Model V84.3 gas turbine for 60 Hz operation. At its Berlin prototype test facility, Siemens has the unique opportunity to test its heavy-duty stationary gas turbines over a wide range in speed and load up to full load. The test facility is described by Deblon (1978). The layout is shown in Fig.2.

The power output delivered by the gas turbine is converted into thermal energy by a water brake rigidly coupled via a torque-measuring shaft. The heat generated is rejected by three cooling towers. The water brake permits off-design operation of the turbine over a wider load range and speed range than is possible in power plants.

3. CALCULATION METHOD

For the calculation of the transient clearances between blade tip and shroud the transient behaviour of the following components affected by changes in temperature and centrifugal forces has to be modeled, see Figure 3:

- blade-ring carrier,
- rotor disc and
- blade.

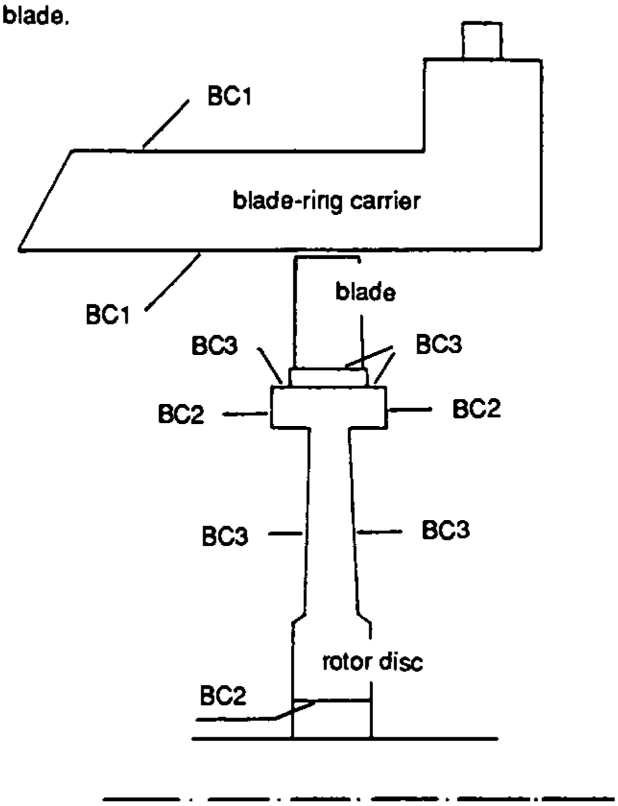

FIGURE 3: GEOMETRY OF A COMPRESSOR STAGE WITH BOUNDARY CONDITIONS.

In the compressor, the shroud is the inner contour of the blade-ring carrier; in the turbine the shroud is a part of the platform of the stator vane.

In the following, the clearances between the blade tip and the shroud are considered. However, the procedure applies also to sealing clearances between the inner stator rings and the rotor discs.

The transient tip clearance Δc results from superposition of the elongations of the single components

$$\Delta c = \Delta c_{cold} + \Delta l_{BC,T} - (\Delta l_{RD,T} + \Delta l_{RD,CF}) - (\Delta l_{B,T} + \Delta l_{B,CF}),$$

where c stands for clearance, l for length and the subscripts BC for blade-ring carrier, RD for rotor disc, B for blade, CF for centrifugal force and T for temperature.

For the elongations of the single components the temperature field has to be determined. This calculation is based on Fourier's law of heat conduction and Newton's law of convective heat-transfer. There are various commercial codes of finite-element or finite-volume type that solve this kind of problem (e.g. ADINA, ABAQUS). Therefore, no details of the numerical procedure adopted for the calculations are presented in this paper. The same applies to the calculation of elongations due to centrifugal forces.

Next, attention is focused on the formulation of the transient boundary conditions. As an example, the geometry of a compressor stage, sketched in Fig. 3, is considered. It shows the above mentioned components (rotor disc, blade-ring carrier and rotor blade), the transient behaviour of which will result in the radial clearances of interest.

We assume the following time-dependent boundary conditions to be known:

- rotational speed,
- inlet guide vane setting and
- total temperature, total pressure at the compressor exit.

These data are either deduced from the design conditions of the compressor and the ideal transients for cold start and power increase up to base load, which result in the design tip clearances or by prototype test data taken to verify the design procedure. The latter will be discussed in a following section.

With the aforementioned data and the off-design calculation results of the compressor, all temperatures at the boundaries (see Fig. 3, BC1 to BC3) and the body forces are known. The remaining unknown boundary conditon is that for the heat-transfer coefficient α. This coefficient is a function of Reynolds number Re and Prandtl number Pr which in general is known. The dependence between α and the Nusselt number Nu is in accordance with the similarity principles:

$\alpha = Nu\,\lambda / l$,

with the heat conductivity λ.

For the example shown in Fig. 3, the evaluation of the heat-transfer coefficient is based on the following formulations of the Nusselt number, see VDI tables (1991).

- BC1, flow over a flat plate:

  $Nu_{lam} = 0.664\, Re^{1/2}\, Pr^{1/3}$

  $Nu_{turb} = 0.037\, Re^{0.8}\, Pr\, /\, (1.+ 2.443\, Re^{-0.1}\, (Pr^{2/3} - 1))$

  $Nu = (Nu_{lam}{}^2 + Nu_{turb}{}^2\,)^{1/2}$

- BC2, turbulent flow in a tube of length l and hydraulic diameter D:

  $Nu_D = 0.0214\, Re_D{}^{0.8}\, Pr^{0.45}\, (1+ (D/l)^{2/3})$

- BC3, turbulent flow over a rotating disc with radius R and rotational speed ω:

  $Nu_R = 0.018\, Re_R{}^{0.8}$ and $Re_R = (\, c\text{-}\omega\, R)\, R\, \rho\, /\, \eta$,

  where $(c - \omega\, R)$ is the relative velocity with c as absolute velocity and the dynamic viscosity η.

The Prandtl number, as well as the thermal properties of the material, is assumed to be known. The Reynolds numbers are calculated from the velocities depending on the compressor inlet mass flow, the bleed mass flows and the compressor exit conditions.

The elongations due to centrifugal forces acting on the rotor disc ($\Delta l_{RD,CF}$) and on the blade ($\Delta l_{B,CF}$) are calculated for nominal speed. During start up and shut-down, the transient is calculated by the dependence

$$\Delta l_{CF(n)}\, /\, \Delta l_{CF(n=no)} = (n/no)^2 \;,$$

where the change in time of the rotational speed is assumed to be known.

For the temperature elongation of the blade ($\Delta l_{B,T}$) the rotor blade is assumed to be infinitely thin, which means that the temperature of the blade is at all times equal to the gas temperature in the case of the compressor and equal to the steady-state metal temperature of the turbine.

4. MEASURING TECHNIQUE

The objective of the tip clearance measurements is twofold:

- guarantee the safe operation of the gas turbine,
- provide quantitative support for the development of design tools.

4.1. Setting tip clearances during assembly

During the assembly blade-tip clearances are set in two steps:

1. With the top of the casing removed, the blade-ring carriers are centered about the rotor by moving them laterally such that the tip clearances at the horizontal joint are symmetrical in each stage of the compressor and the turbine. This open tip clearance measurement is part of the assembly procedure. The magnitude of the tip clearances in the turbine is set by moving the rotor axially.

2. After the assembly is complete, the vertical centering of the rotor and an adjustment of the horizontal centering are carried out according to tip clearance measurements at selected stages, i.e. two for the first and one for the second to fourth blade-ring carriers of the compressor and one for the blade-ring carrier of the turbine. A final cold tip clearance measurent is part of the quality-control procedure of the assembly.

4.2. Measuring hot tip clearances

In the Model V84.3 gas turbine, hot tip clearances are measured in the following stages during prototype testing:

Compressor: Stages 1, 4, 8, 13, and 16

Turbine: Stages 1, 2, 3, and 4

The clearance probes are inserted into the machine at four circumferential positions, one each at the top and the bottom, and two near the horizontal casing joint.

The minimum hot tip clearance is measured with abrasion pins.

The transient tip clearance is measured with electro-mechanical probes.

The agreement between the various measuring techniques was assessed by repeated cold tip clearance measurements with various techniques. The 90% confidence intervals are listed in Table 1. The measuring techniques for hot clearances are discussed below

Method	90%-confidence interval (+/-)	
	Compressor (mm)	Turbine (mm)
Quality control	0.13	0.40
Abrasion pins	0.10	0.19
Electro-mechanical sensors	0.06	0.06

TABLE 1: 90%-CONFIDENCE INTERVALS OF TIP CLEARANCE MEASUREMENTS.

4.2.1. Abrasion pins

Early in the prototype testing program, abrasion pins were used to determine the minimum hot tip clearance which is crucial for the safe operation of the machine. The abrasion pins can be inserted in the five selected compressor stages and all four turbine stages at the same time. Thereby a consistent set of minimum hot clearances can be generated.

The tapered pins are abraded by the blade tips. In the compressor, copper pins were used because temperatures are below 400°C, (see Table 2). Abrasion pins in the turbine require austenitic stainless steel to provide high-temperature strength and corrosion resistance.

	Material	Diameter
Compressor	copper	6 mm
Turbine	austenitic stainless steel	5 mm

TABLE 2: MATERIAL AND SIZE OF ABRASION PINS.

Prior to a hot clearance measurement, the cold tip clearance was measured using the abrasion pins as well. During turning-gear operation, the threaded pins were turned towards the blading until audible contact (i.e. one click per revolution) was made between the longest blade and the pin. The resulting cold-tip measurement agreed with the quality-control measurement within the range of uncertainties.

Figure 4 shows two abrasion pins of different length before they were mounted in the compressor's blade-ring carriers (top). In the turbine, the abrasion pins (bottom) are spring-mounted against the shroud portion of the platform of a turbine vane. The springs were mounted in the large, cylindrical housing (right) at the end of a pipe which is threaded into the outer casing of the machine. Thus, the reference point for the measurement is always near the blade, and thermal expansion of the casing relative to the shroud of the blades is compensated by springs inside the housing.

4.2.2. Electro-mechanical tip clearance measurements

The PC-controlled transient tip clearance measuring equipment was made in house using Rotadata probes. A low-voltage contact-measuring technique was chosen. The data acquisition program and electronics can accommodate up to eight sensors, thus covering two stages at a time.

A single data point is generated with the following sequence of events:

1. From the start position, the contact pin is advanced towards the blades to the inner edge of the shroud at a pre-determinded speed.

2. From the edge of the shroud until contact, the pin is advanced at a speed-dependent rate, in steps of approximately 3 μm per revolution.

3. As soon as contact is made, a flip-flop is set in the controller of the actuator and the probe tip is withdrawn to the starting position. When the data acquisition program reads the electronics the next time, the revolution count of the actuator spindle is transferred to tha PC. After the last of the eight sensors has yielded a result, the set of results is transferred to hard disk and a printer. Only then will the data acquisition program start a new measuring sequence.

Since the feed rate of the probe tip is identical for all probes, the time from the start to the end of a set of measurements is dictated by the largest clearance. It is this time of the completion of the last measurement which is recorded as the time of the measurement in the data discussed below.

The probe tips are shrouded with nitrogen which is provided to each at a pressure higher than the static pressure of the stage under investigation. The nitrogen prevents air and - more importantly - hot exhaust from contaminating the interior of the probes. Since the withdrawal of the probes is triggered independently of the PC' s data acquisition cycle, the maximum possible interference between blade and probe is 3 μm per contact. Wear of the probes was not observed. This tactile measuring technique has proven robust and sufficiently fast to resolve the thermal expansion transients of interest.

Figure 5 shows the assembly of the electro-mechanical tip clearanca probe. The probe is spring-loaded against an adapter in the blade carrier (compressor) or directly against the shroud portion of a vane (turbine). Thus, the reference for the measurement is always near the blade, and thermal expansion of the casing relative to the shroud of the blades is compensated by the spring. The actuator is mounted at the end of a pipe which is threaded into the outer casing of the machine. The close-up of the actuator (Fig. 6) shows the electric connector (left) and nitrogen supply fitting (to the left of the flange, pointing up). The close-up of the Rotadata probe shows the tip protruding from the probe (Fig. 7).

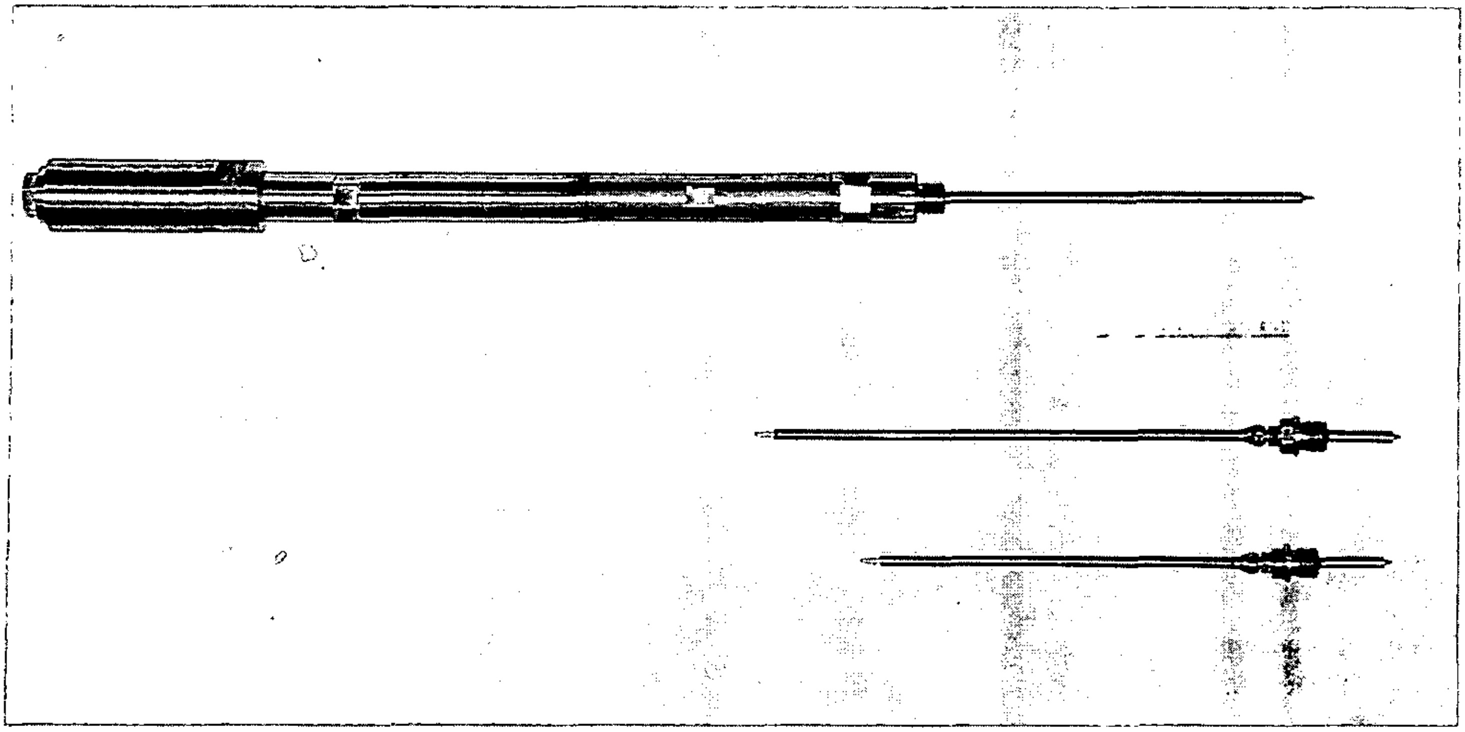

FIGURE 4: ABRASION PINS (ONE FOR THE TURBINE, ABOVE; TWO OF DIFFERENT LENGTH FOR THE COMPRESSOR, BELOW).

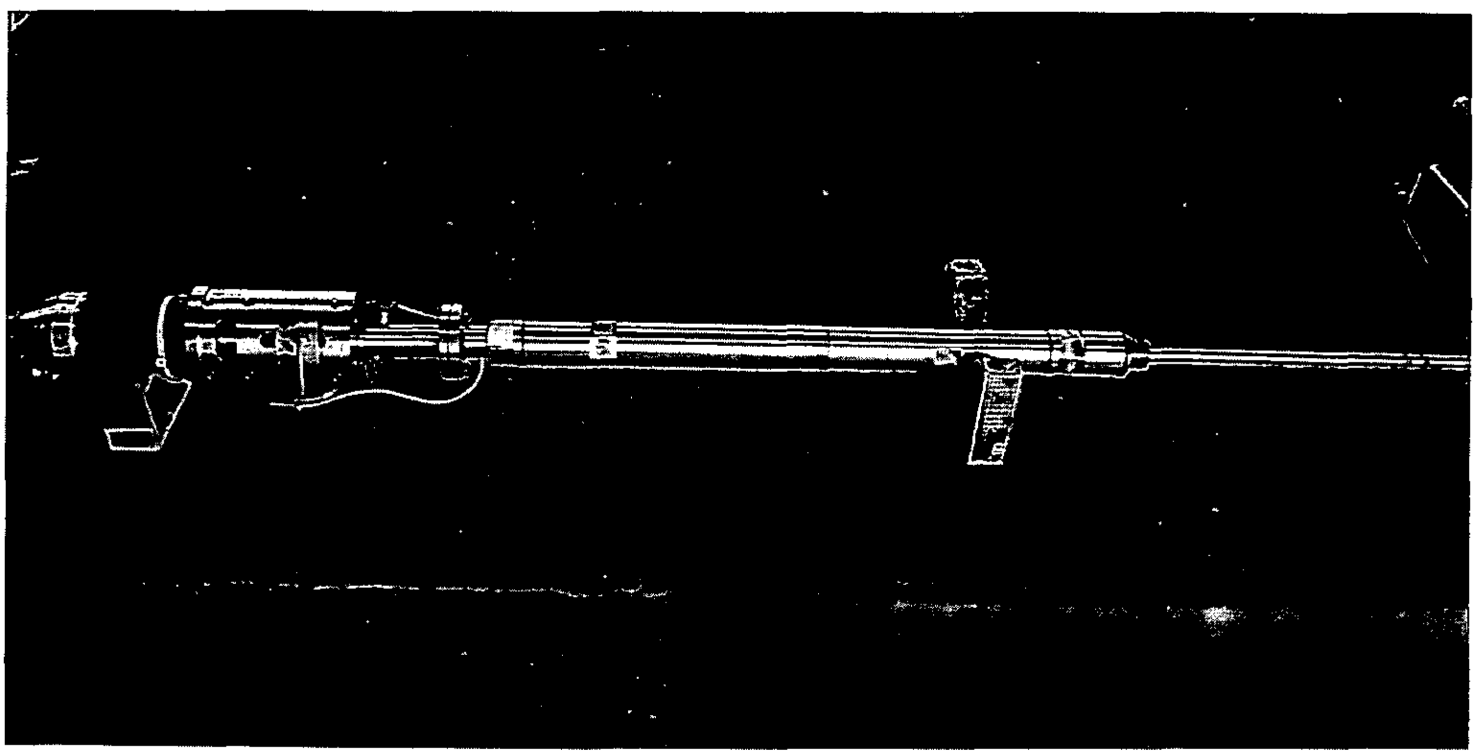

FIGURE 5: ASSEMBLY OF THE ELECTRO-MECHANICAL TIP CLEARANCE PROBE.

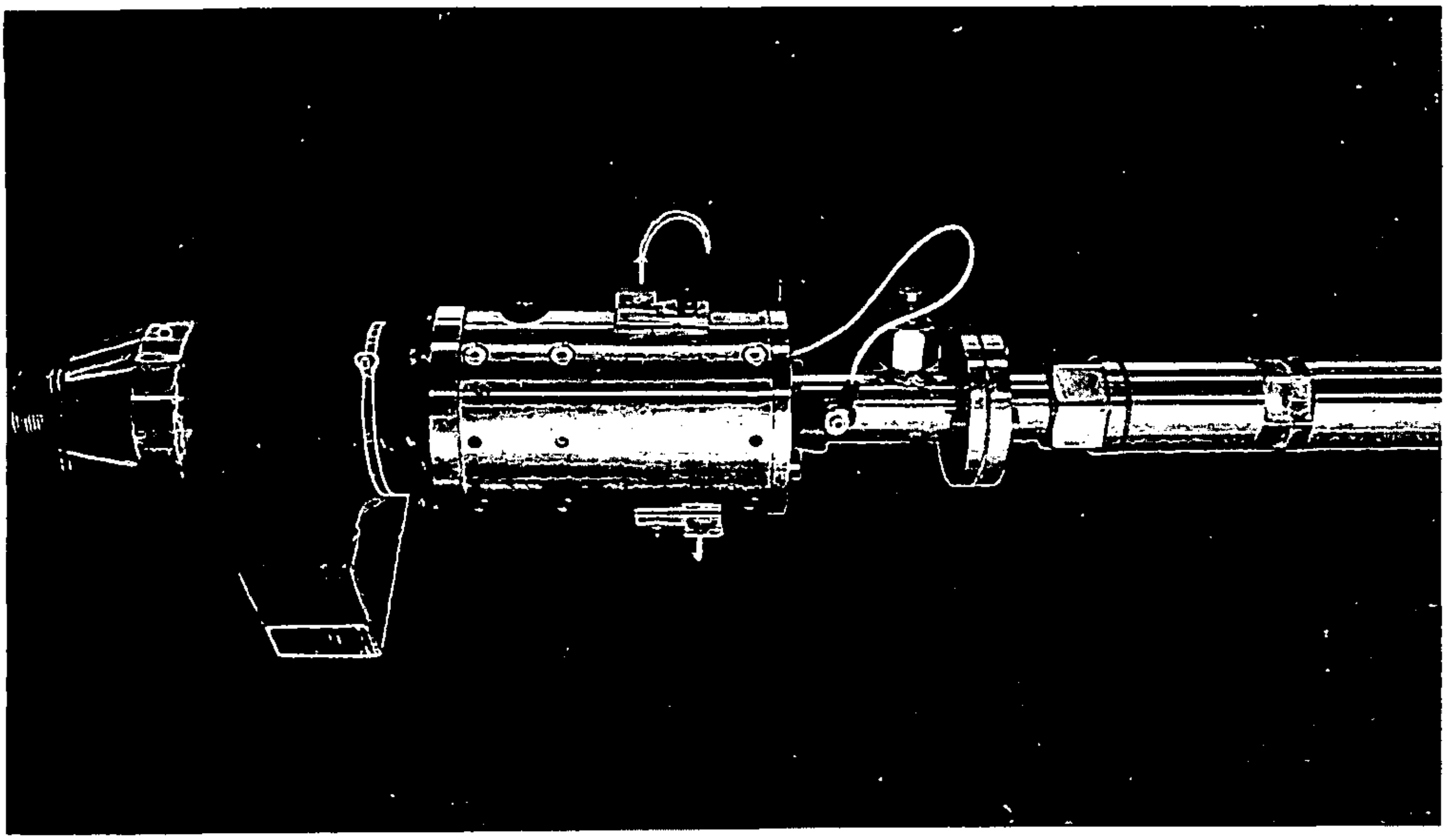

FIGURE 6: ACTUATOR WITH ELECTRIC CONNECTOR AND NITROGEN SUPPLY FITTING.

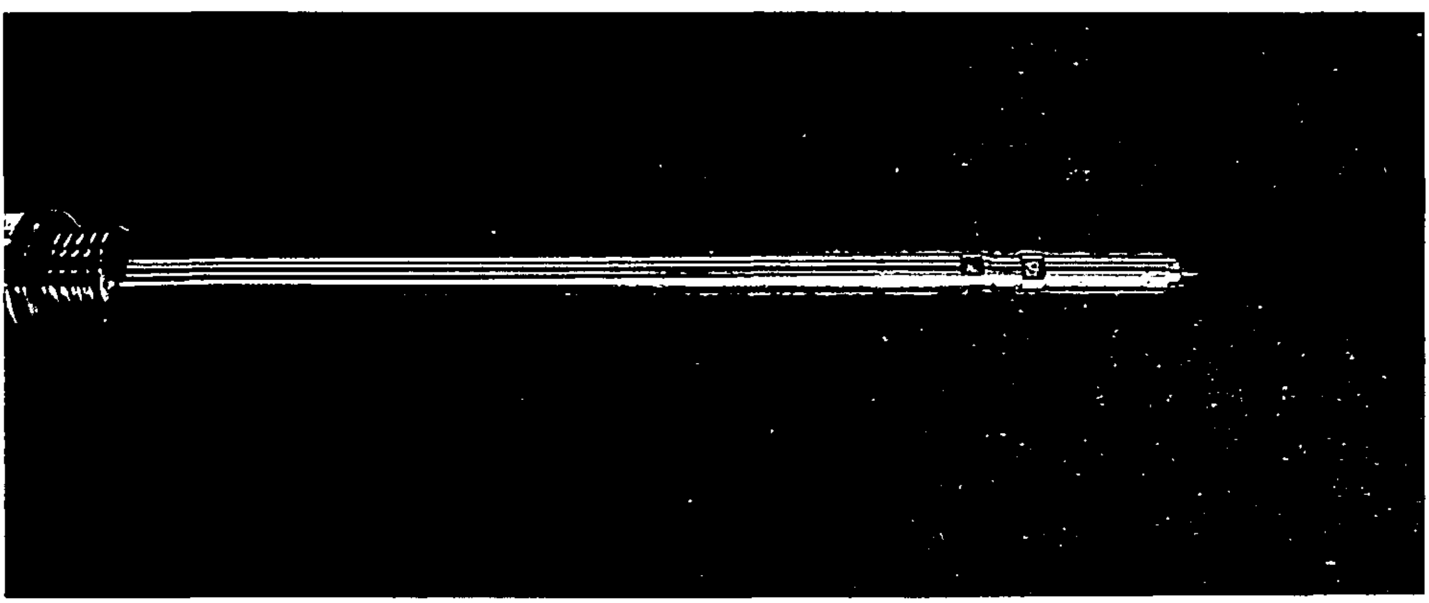

FIGURE 7: TIP PROTRUDING FROM THE ROTADATA PROBE.

5. COMPRESSOR TIP CLEARANCE RESULTS

Tip clearance measurements were performed in five stages of the compressor. In all cases the measurements were carried out with abrasive pins as well as electro-mechanical sensors. For the evaluation of the tip clearances at all operating conditions, the measurements were performed for the cycle described in Fig. 8 in terms of rotational speed, inlet guide vane setting and power output. For the discussion of the measurements and the comparison of measured and calculated results, the transient clearances for the 13th and the 16th stage are considered.

As mentioned earlier, the boundary conditions for the calculations are taken from the prototype test data and the off-design calculations for the compressor.

5.1 13th compressor stage

During cold start the clearance decreased to about 55% due to the centrifugal forces acting on the rotor disc and the blade, as well as the quick warm-up of the blade, Fig.9. The fast temperature response of the blade-ring carrier results in an increase of the clearance within a short time during loading. The clearance decrease during part-load operation was dominated by the thermal expansion of the rotor disc due to increasing metal temperatures. The radial clearance resulting after 30 minutes of part-load operation did not reflect the stationary hot clearance since the rotor disc had not reached a steady-state temperature distribution. This becomes obvious on studying the clearance after the very beginning of the intermediate cooling interval (shut-down) which is

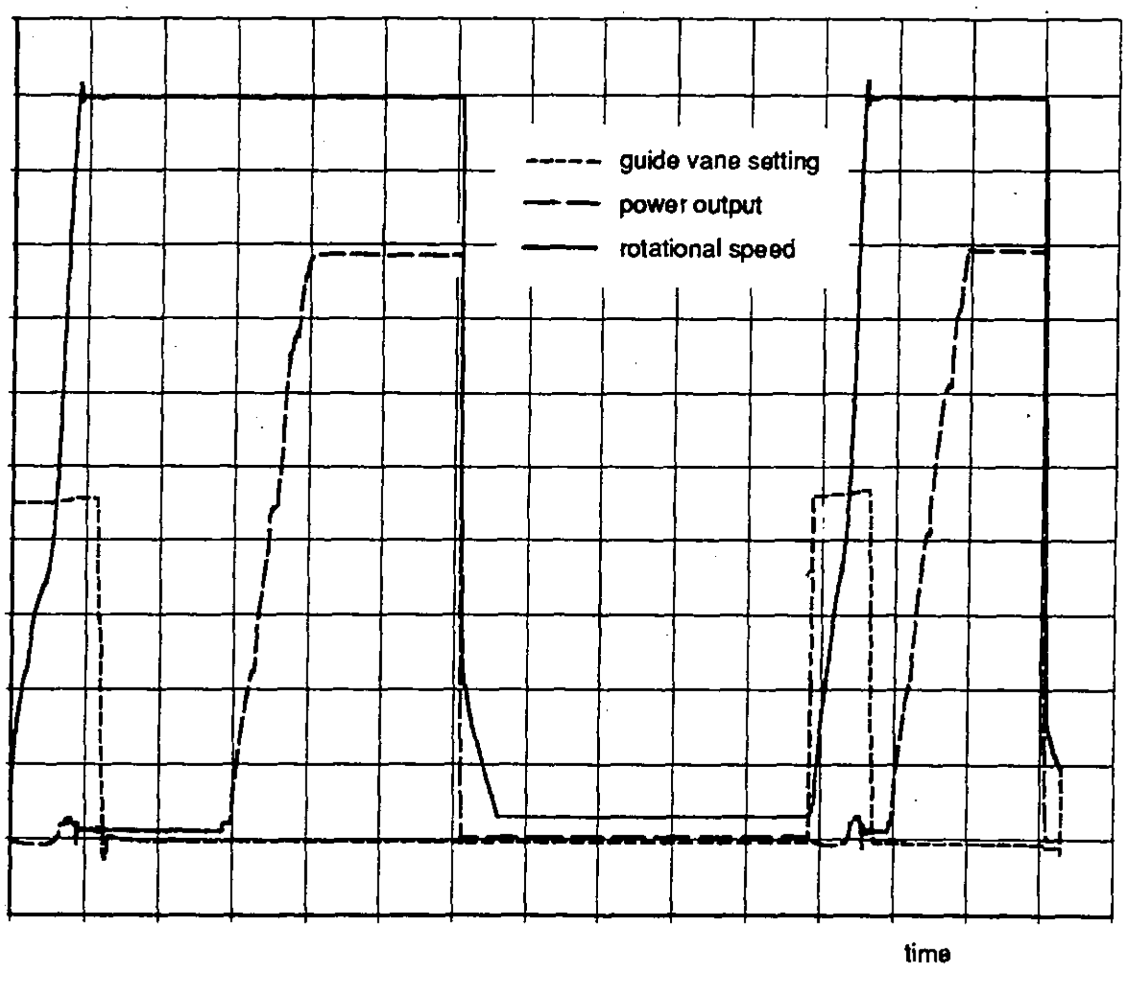

FIGURE 8: LOAD CYCLE IN TERMS OF ROTATIONAL SPEED, INLET GUIDE VANE SETTING AND POWER OUTPUT.

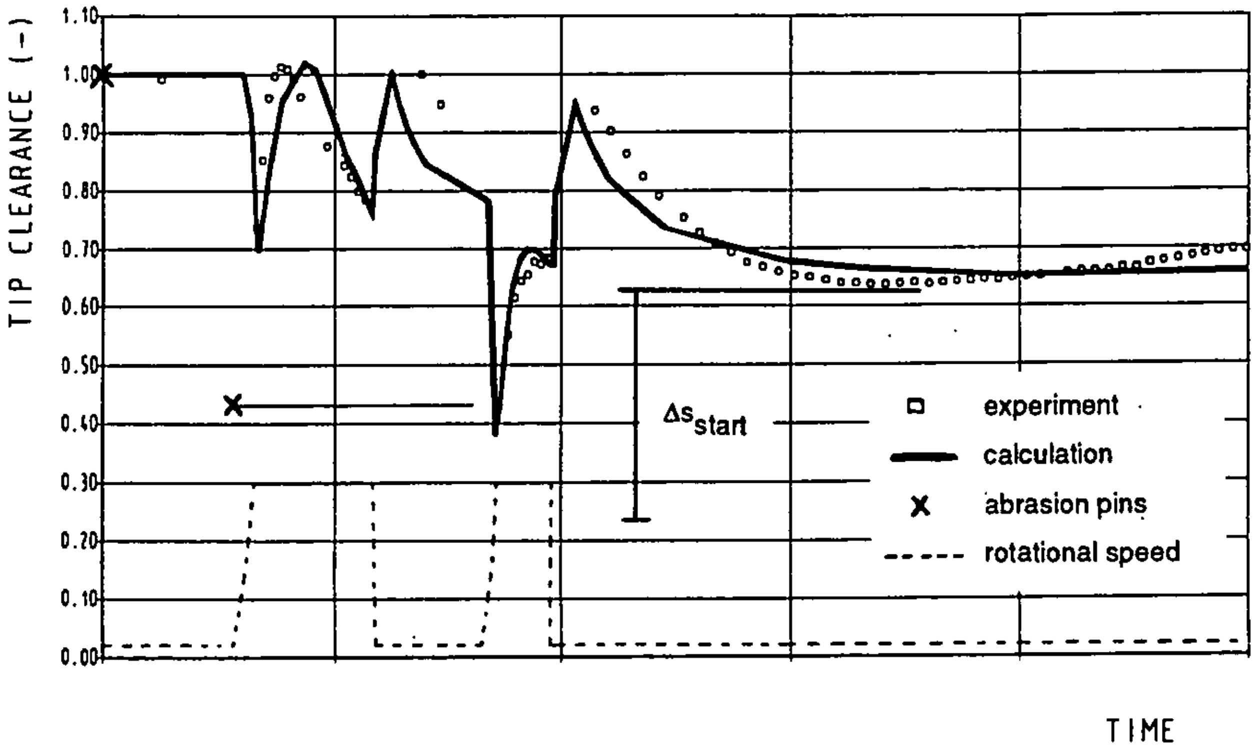

FIGURE 9: OPERATING CLEARANCE FROM PROTOTYPE TEST AND CALCULATION FOR THE 13TH COMPRESSOR STAGE.

larger than the cold clearance at the beginning of the test run. Knowing by calculation that the temperature elongation of the blade-ring carrier and the rotor disc are identical to within a tenth of a millimeter, the clearance immediately after shut-down will match the cold clearance if all the components had reached a steady-state in terms of temperature. Driven by the fast cooling-off of the blade-ring carrier after shut-down, the resultant clearance decreases below the cold clearance. As a result, the minimum clearance during the hot-start is lower than the one measured during the cold start of the machine. After reaching nominal speed, the clearances become larger again as a result of the quick warm-up of the blade-ring carrier. During the second part-load operation, the rotor disc is heated up to almost its stationary temperature distribution, which can be seen by observing that the clearance level after the second shut-down of the machine was close to the measured cold clearance. The transient clearance after the second shut-down reflects very clearly the fast cooling-off of the blade-ring carrier and the rather slow cooling-off of the rotor disc, resulting in a minimum clearance of approximately 60% of the cold clearance after 2 to 3 hours.

For the estimation of the smallest possible clearance the minimum clearance during cooling-off should be considered. Restarting the machine at a time when minimum clearance is reached yields a clearance given by the measured minimum clearance minus the elongations due to centrifugal forces on the rotor disc and the blade as well as the thermal elongation of the blade. These elongations add up to Δs_{start}, shown in Fig. 9 as a thick bar. The superposition of the minimum clearance during cooling-off on Δs_{start} shows that even in the case of a hot-start, at the time at which the minimum clearance is reached, the rotor blade does not touch the blade-ring carrier.

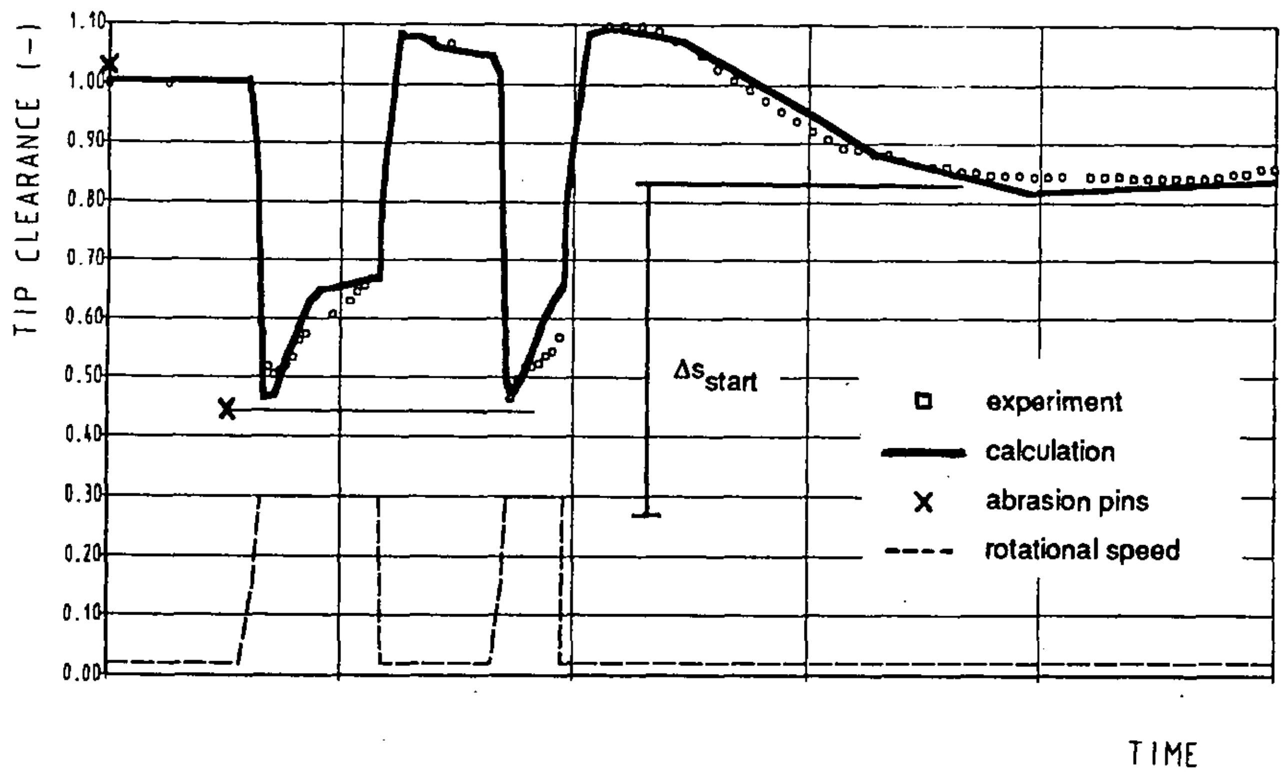

FIGURE 10: OPERATING CLEARANCE FROM PROTOTYPE TEST AND CALCULATION FOR THE 16TH COMPRESSOR STAGE.

5.2 16th compressor stage

The clearance measurements for the 16th stage were conducted simultaneously with the ones for the 13th stage, (Fig. 10). The transient clearances are not very different from the ones dicussed in detail for the 13th stage. For that reason we will focus here on the comparison with the transient clearances of the 13th stage.

The most obvious difference is the moderate increase of the clearance after reaching the nominal speed. The steep gradient in the 13th stage at the beginning of the first, as well as the second load operation does not occur in the 16th stage. This is due to the faster warming-up of this stage. The design of the high-pressure compressor (Fig.11) shows that the 16th rotor disc is internally cooled or heated by the bleed downstream of the 14th rotor row. This design causes the rotor rows No. 14 to 17 to expand as fast as the 4th blade-ring carrier and allows minimum design cold clearances to be achieved.

As can be seen in Fig. 10, the decrease in the clearance during cooling-off measured in the 16th stage is 30% less than the one measured in the 13th stage.

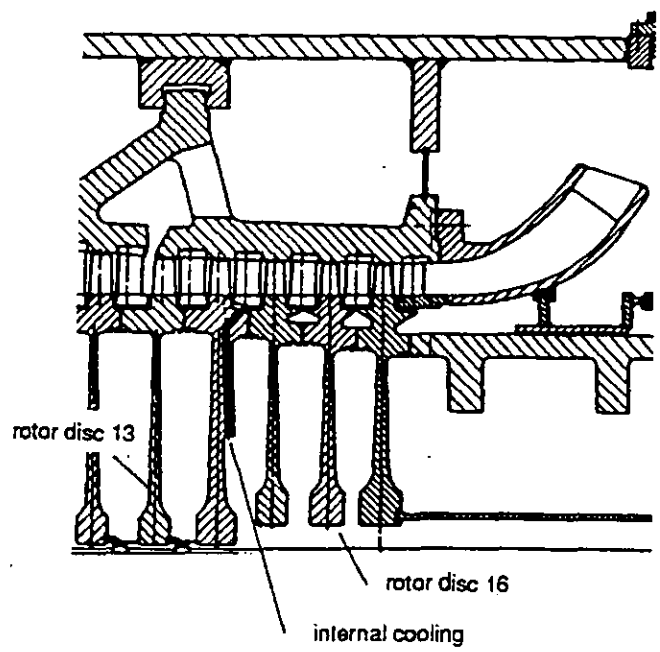

FIGURE 11: HIGH-PRESSURE COMPRESSOR DESIGN.

As a comparison of the measuring techniques the minimum clearances measured by abrasive pins have been entered into Fig. 10. The clearances measured by abrasive pins confirm the electro-mechanical probe measurements.

The comparison of calculated and measured transient clearances is very satisfactory. The differences are within the uncertainty of the measurements discussed in Table 1. One of the main conclusions that can be deduced from these comparisons is that the adopted design tools are improved by correcting the calculations of the heat-transfer coefficient. This renders a further reduction of the cold clearances possible.

6. TURBINE TIP CLEARANCE RESULTS

The transient thermal expansion of the turbine components is difficult to estimate in the early design process since it depends on the conditions of the main flow and, more importantly, on the cooling and leakage flows. The formulation of the boundary conditions is based on the design calculation of the cooling and leakage system. Due to the cooling flows, components like the cooled blade-ring carrier in turbines do not follow the load cycle as closely as blade-ring carriers in compressors.

To prove the assumptions mainly of the heat transfer coefficient used in the design process, measured temperature data of the blade-ring carrier from prototype tests were compared with calculations. With the measured data, the elongations can directly be compared with the calculated ones for the entire load cycle and corrections of the heat transfer coefficient can be derived. Prototype test results are now compared with the calculated operating clearances and the expected clearance transient in Fig. 1. Qualitatively, Figs. 12-14 agree with Fig. 1.

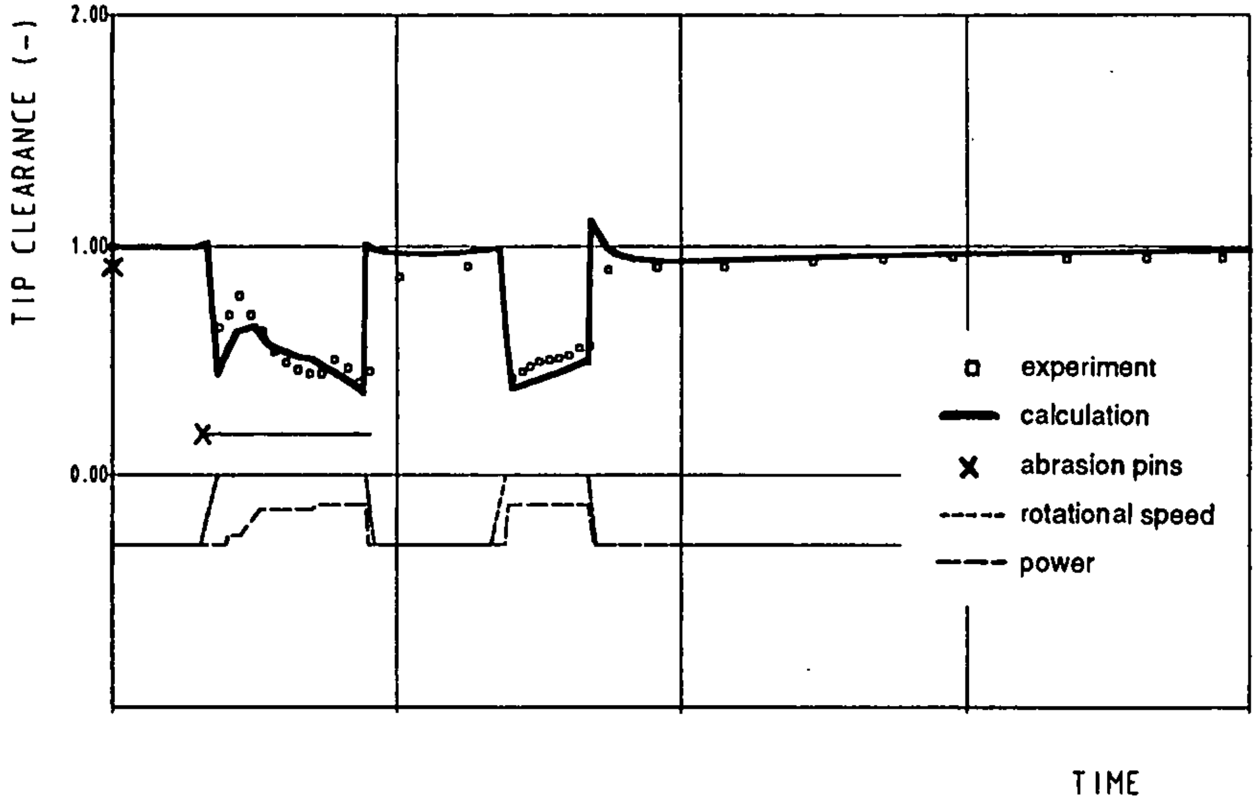

FIGURE 12: OPERATING CLEARANCE FROM PROTOTYPE TEST AND CALCULATION FOR THE 2ND TURBINE STAGE.

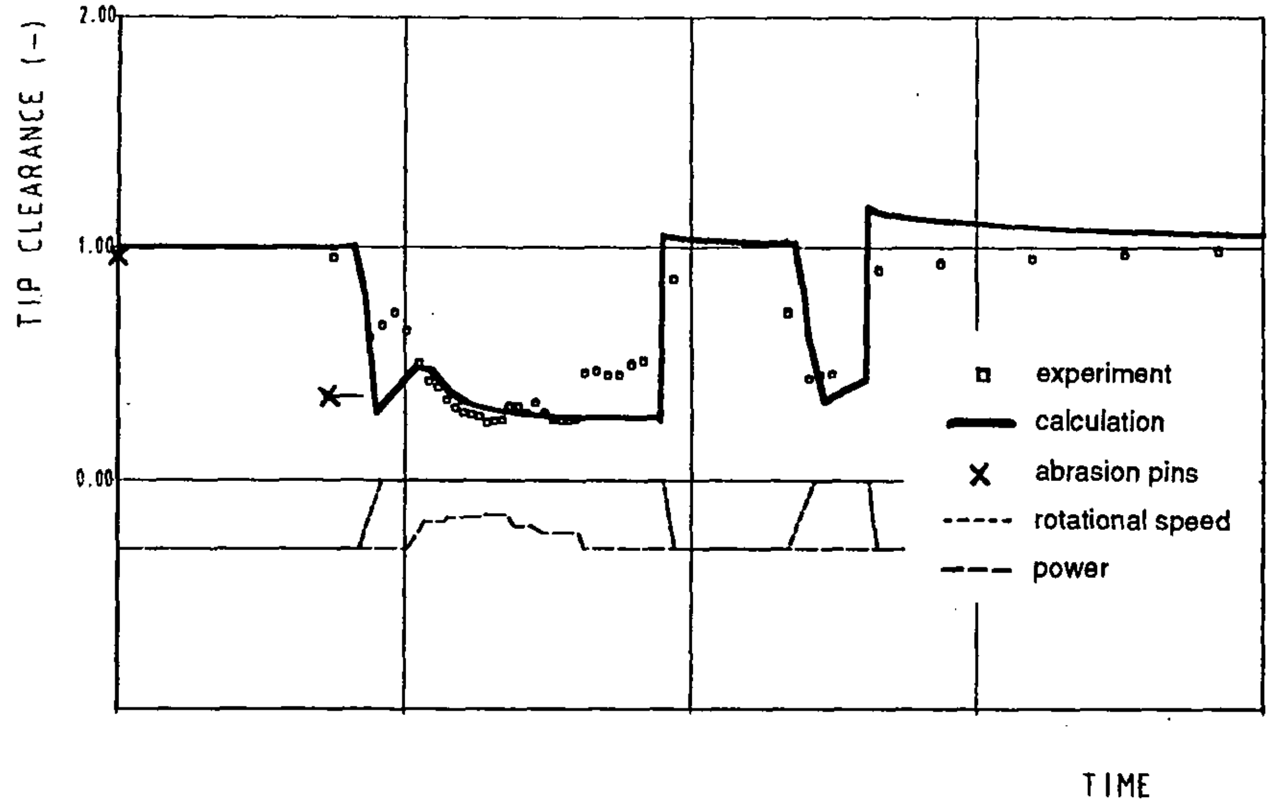

FIGURE 13: OPERATING CLEARANCE FROM PROTOTYPE TEST AND CALCULATION FOR THE 3RD TURBINE STAGE.

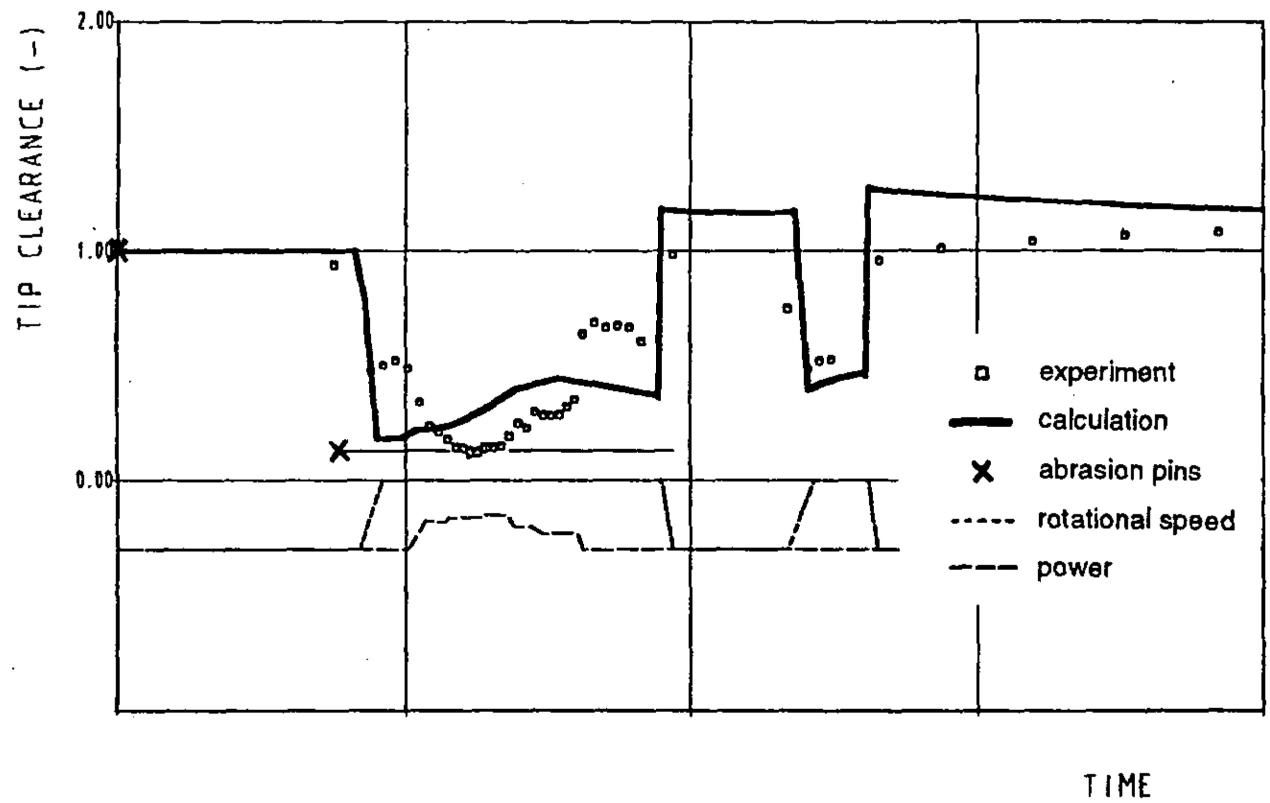

FIGURE 14: OPERATING CLEARANCE FROM PROTOTYPE TEST AND CALCULATION FOR THE 4TH TURBINE STAGE.

After cold start the tip clearance decreased rapidly due to the centrifugal load acting on the rotor disc and blade. The power increase leads to a very fast thermal elongation of the blade-ring carrier and the blade followed by the thermal elongation of the rotor disc. Especially for stage 3 and 4 (long blades) the deviations are quite high due to changes in power output (turbine inlet temperature). This can be attributed to the simplified modelling of the thermal elongation of the blades. Thermal blade growth due to load changes are resolved in the calculation only for start and shut down.

At the start of intermediate cooling (shut-down), the clearance increased due to reduced centrifugal load and decreasing blade temperatures. Under the same operational condition as during cold start idling the tip clearances behave similar because the rotor disc and blade-ring carrier keep the heat. During shut-down the clearance change due to centrifugal load is close to the one at cold start. The superposition of the elongations of the blade-ring carrier, blade and disc result in an almost constant clearance during the intermediate cooling phase. This shows a well tuning in the thermal behaviour of the single components.

After almost an hour of intermediate cooling, the hot-start reduced the clearance much faster than during the cold start because the rotor disc retains its temperature longer than the blade-ring carrier and the blades.

The transient clearance during cooling-off shows a much faster approach of the tip clearance towards the initial cold value than for the compressor (see Section 5). This phenomenon has already be discussed above (intermediate cooling).

Measurements and calculations agree in that, contrary to the expectation in Fig. 1, hot-start does not represent the critical condition. The clearance remains above the operating values of the cold start. This conclusion can be drawn from the Figures 12-14, with the hot-start effect even decreasing from Stage 2 to Stage 4. The end of the transient shows that the initial cold tip clearance is nearly reached. The operating clearance after cold start is small which guarantees optimum energy conversion.

7. CONCLUSION

The main conclusions that can be drawn from the measured 13th and 16th-stage compressor tip clearance results and from the comparison with calculated clearances are firstly the verification of no blade abrasion even during hot-start, secondly the possibility of reducing cold clearances in the last three compressor stages due to the fast warming-up of these rotor rows and thirdly the good agreement between measurements and calculations which renders a further optimization of the rotor disc and the blade-ring carrier design possible in order to achieve a further reduction in hot clearances and, therefore, an improvement in compressor efficiency.

The calculated clearance distributions agree well with the measured turbine 2nd, 3rd and 4th-stage tip clearance results. Due to the well tuned design of the single components the superposition of the blade-ring carrier , blade and disc results in an almost constant clearance during the intermediate cooling phase. At hot-start, the sudden centrifugal load in addition to the already thermally expanded state of the rotor does not result in a critical minimum clearance in the turbine. The operating clearance in the usual cold-start case is small, minimizing tip clearance losses.

The excellent matching of component cooling in the turbine is only possible because of the interior cooling air ducted through the rotor. This matching of thermal transients of stator and rotor is possible because the Siemens rotor is assembled of individual discs, each of which can be matched individually which is not possible in solid rotors.

Data from prototype test and calculation represent a useful database for a further improvement of the design tools.

REFERENCES

Hah, C., 1986, 'A Numerical Modelling of Endwall and Tip-Clearance Flow of an Isolated Compressor Rotor', Journal for Gas Turbines and Power, Vol. 108, pp.15-21.

Lakshminarayana, B., 1970, 'Method of Predicting the Tip Clearance Effects in Axial FlowTurbomachinery', ASME Journal of Basic Engineering, Vol. 92, pp.467-481.

Peacock, R. E., 1989, 'Turbomachinery Tip Gap Aerodynamics - A Review', Proceedings 9th ISABE, Athens, pp.549-559.

Deblon, B, 1978, 'Full Load Test of the 80 MW Gas Turbine V94.2 using a Water Brake', ASME 78-GT-68.

VDI - Wärmeatlas (Berechnungsblätter für den Wärmeübergang), 1991, 6. Auflage, ISBN 3-18-401083-X, VDI - Verlag.