FRETTING FATIGUE CRACKING OF A 936 MW TURBO-GENERATOR ROTOR
FAILURE ANALYSIS AND REFURBISHMENT

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Occurrence of the Failure

In March 1990, turbo generator unit 2 of the nuclear power plant Darlington in Canada tripped on the turbine side as a result of a high level alarm in a water tank. This unit had been commissioned in 1990 as the first of four identical units. The nominal rating of each unit is 936 MW at 60 Hz and 1800 rpm. The operating behaviour at nominal load had shown low levels of vibration of both, bearings and shaft. The absolute values of the vibration amplitudes were so low that vibration can be excluded as being a cause of the failure. However in the 60 hours immediately before the trip there was a clear and progressive increase in the vibration amplitudes. During run-out after the trip, large amplitudes were measured when passing through the critical speeds. In particular, a major component of vibrations was observed at both generator bearings at twice the rotational frequency. This is unusual for four pole generator rotors.

As a consequence of these observations, an Inspection of the generator rotor was carried out. Even the visual inspection of that rotor was sufficient to reveal a crack in the non-drive shaft end. The crack extended from the thread for the retaining nut of the current lead bolts. had a length of ca. 400 mm in the circumferential direction and propagated perpendicular to the rotor axis (Figs.1 and 2). With this older design principle, the retaining nut supports the major part of the centrifugal force exerted by the current lead bolt. A second Smaller crack was found on the opposite side of the hole. This crack, however had not yet reached the rotor surface.

Failure Analysis

in order to determine the cause of failure and the consequent development of a refurbishment solution, an extensive investigation program was defined and performed along with the customer, Ontario Hydro. The program included metallographic investigations of the fracture, material testing, finite element calculations and strain measurements.

For the metallographic investigation, the shaft was broken open at the crack location and the fracture surfaces were examined using stereo microscopy, scanning electron microscopy and the electron microprobe. The materials laboratories of Ontario Hydro and ABB each received one of the fracture surfaces. Apart from 4 beachmarks, the fracture surfaces were flat and featureless, characteristic of fatigue cracks which have propagated under pure tension-compression loading. Using information based on both the position and form of the small crack and the beachmarks on the surface of the large crack, it was possible to clearly identify the thread as the position of crack nucleation. Typical damage features of fretting loading could be observed on the load-bearing surfaces of the thread: fretting corrosion, fiction marks, tom-out material particles, detached oxide layers and micro cracks. These findings were clear from the independent investigations of both Ontario Hydro and ABB.

Down to a depth of 0.3 to 0.5 mm, the micro cracks grew at an angle of about 45 degrees to the surface normal. There was then an alteration in the direction of propagation and the cracks extended at right-angles to the rotor axis and hence also at right-angles to the bending fatigue stresses present. This shape of the crack path and the observed damage characteristics at the surface demonstrate that the fretting process was the cause of crack initiation and early propagation. However, with increasing crack depth, the bending fatigue load increasingly affected the crack propagation and the influence of the fretting decreased until finally the crack extended solely as a result of the cyclic bending fatigue stresses. This means that the bending fatigue loading was responsible for the continued crack propagation. This last phase occurred during a very short time. This is because a rotor running at 1800 rpm experiences within one week nearly 10 million fatigue Cycles. For comparison, beyond this number of cycles a ferritic steel is considered to show no further reduction in fatigue strength, i.e. it is no longer suspectible to fatigue failure.

Neither ABB nor Ontario Hydro found an the fracture surfaces any indications of a material

defect. This result was confirmed by material tests on specimens taken from the rotor in the immediate vicinity of the fracture surface. Both the chemical composition and all mechanical values complied with the prescribed specification for the rotor steel.
In order to investigate the cracks which had formed in the wedge lands of the winding connection slots after 4000 hours of operation (Fig.3), cylindrical specimens were cut from the wedge slots and broken open in the laboratory. Again the typical characteristics of fretting loading were visible on the surfaces of the wedge and wedge slot. Material from the aluminium alloy of the wedges could be detected on the fracture surfaces of the wedge slot specimens down to a depth of ca. 1 mm (Fig.4). This is also a clear indication of the fretting loading. A further indication was given by the determination of the ratio of crack depth to surface crack length, which was less than 0.25 for most of the cracks. This value is characteristic for cracks for which initiation and propagation are primarily driven by shear stresses. If, on the other hand, the tensile stress is dominant then almost semicircular cracks are formed with a ratio of approximately 0.5.

Means of Refurbishment

The cause of failure, namely crack initiation through fretting, and crack propagation as a result of bending fatigue loading, could be clearly identified from the results of the metallographic investigation. This applied both for the crack which propagated from the thread in the region of the current lead bolt and for the small cracks at the end of the winding connection slot. In addition, the observation that changing the wedge material from steel to aluminium did not prevent the occurrence of fretting fatigue provided the important information that simply an alteration of the material combination was not sufficient. This observation did not entirely agree with previous experience gained in previous fretting fatigue tests, in which a higher fatigue strength was determined for the combination aluminium-steel than for steel - steel. Since the rotor dimensions and the bearing span, and hence also the nominal bending stresses, were fixed, the direction for the means of refurbishment was already indicated: avoiding the fretting load through eliminating any surface contact in zones with high fatigue loads. This eliminates the crack initiation mechanism and the bending fatigue stress alone is insufficient to nucleate a fatigue crack. The precept for the refurbishment solution was thereby the development of a contact-free winding connection which can be retrofitted to the rotors.

Contact-free Winding connection

The design principle of the contact free winding connection is described in the following. The thread for the retaining nut was drilled out end the copper current lead bolt was exchanged for a free-standing steel bolt (Figs.5a,5b). Since the mechanical properties of the steel bolt are much better then those of the previously used copper alloy, it was possible to omit the retaining nut. The steel bolt was fixed in the lead in bar through a steel sleeve with a conical form. However, as a result of the much higher electrical resi stance of steel, it was necessary to increase the cooling effect through increasing the gas flow rate. This was achieved by using the full fan pressures in order to improve cooling of the steel bolts. This concept is standard for today's H2-cooled ABB generators.

In addition, the bending fatigue stresses in the region of the current lead bolts were reduced by 20%. This was possible with a stress-optimised geometry in the upper part of the current lead bolt holes. The optimisation was identified based on the results of extensive FE calculations.

The principle of contact, free winding connections was also used for the connection between the bolt and the first winding. The small fretting cracks which had formed at the end of the wedge slots nearest to the rotor body ends were completely cut away. The form of this cut was also optimised In terms of stress level on the basis of 3D FE calculations. The bending stresses could be reduced even below those for the original geometry. The new connecting lead was designed in two parts (Fig.5a). The first part leads self-supported from the upper end of the current lead bolt to the retaining ring plate, where it is supported without electrical contact. The supporting element has a design similar to a bearing pedestal, in which force is

transmitted through a shrink fit. Screws are employed to make the connection to the retaining ring plate. This first self-supported component must be designed taking account of the leading arising from centrifugal force acting on the steel.

The second component is connected directly to the first and leads under the end winding to the connection to the exciter winding. This second part is supported against the end winding, sp that a flat copper profile could be used. It is screwed to the steel end and brazed to the innermost coil.

With this concept, design factors ensure that there is no contact at all with the shaft, so that fretting can be totally excluded as s possible cause of crack nucleation.

The first part of the contact-free winding connection now joins the current lead bolt to the retaining ring plate. Without this connection there would be a relative displacement between these two parts, as a result of rotating bending (see Fig.6). For the forced displacements, the consequent fatigue loading of all components involved is reduced by minimizing the overall stiffness of the connecting elements. The resulting fatigue loads have been minimized by increasing the flexibility of the bolts and steel conductor i.e. ensuring low bending stiffness. For this reason, the rectangular profile of the steel conductor has been slit twice in the radial-axial direction. The conical current lead bolts have been slit in the centre section in such a way as to produce six individual circular segments. These two measures were able to sufficiently reduce the overall stiffness of the bolt end steel conductor without significant impairment of the ability to support the static centrifugal forces. The free-standing steel bolt has been a standard ABB solution for about the last 20 years and has proved itself reliable in many generators. On the other hand, the design with contact-free winding connection is new, so that there is no operating experience. Safety of operation was therefore demonstrated in a full-scale component test (sea section on component testing). In addition, a safety ring, similar to a retaining ring, was connected to the retaining ring plate by a flange, thereby completely enclosing the contact-free winding connection. If the contact-free winding connection or the current lead bolt failed, then this additional design feature would be sure to prevent further damage to other components of the generator.

Finite Element (FE) Calculations

In addition to the laboratory investigations, both ABB and Ontario Hydro performed 3D FE calculations in order, on one hand, to clearly identify the failure mechanism, and on the other hand to find the best restoration solution from the point of view of stress optimisation. The results of the FE calculations were checked with the help of strain gauge measurements made on the coupled shaft. The measurements showed excellent agreement with the results of the FE calculations, which, for the purpose of comparison. were performed independently by Ontario Hydro and ABB.

In the region of the thread for the retaining nut. The nominal bending stress in the coupled condition is 35 MPa. Other half-speed ABB machines reach only about half of this value at similar ratings. Initially, a similar design was also planned for the Darlington rotors. However, with this design it would not have been possible to keep the range close to the operating speed free of critical speeds, since the second critical speed of the generator would have bean located only marginally above the operating speed. For this reason, the diameter of the shaft ends was reduced and the bearing span of the generator rotor was increased, in order to reduce the second critical speed sufficient below the operating speed,

The stress concentration factor at the opening of the current lead bolt hole, i.e. at the most highly loaded point, could be reduced by 20% by performing a stress optimisation of the geometry. After the modification, the maximum bending fatigue stress in the shaft end is still 102 MPa and the mean stress, resulting from centrifugal force and thermal stress, lies in the compressive range. at -16 MPa. These figures show that the situation represents almost symmetrical tension/compression loading, which should be compared with the endurance limit of the rotor material of 410 MPa. The endurance limit was measured and statistically evaluated in the materials laboratories of Ontario Hydro and As9, using s test matrix of 165 specimens. The value of 410 MPa corresponds to a 0% failure probability according to the

evaluation procedure using the arcsin method, as is generally employed in Germany (Reference: B.Lee, J.Denk. G.Ebi "The High Cycle Fatigue Behavior of Darlington Unit 2 Generator Rotor Material` to be presented at the "Fatigue 93" Conference in Montreal. Canada, 1993). From the values of the endurance limit and the maximum loading, it can be seen that after the refurbishment modification of the Darlington generator rotors, there is a safety factor of 4 against fatigue failure.

In order to determine the mechanical loading of the contact-free winding connection, first the displacements and strains were determined at the connection points, using a 3D model consisting of rotor, rotor retaining ring, shaft end, current lead bolt and steel conductor. The maximum deflection experience in the axial direction (parallel to the rotor axis) by the tip of the bolt, with the connection to the winding connection released, was 0.2 mm. In the circumferential direction the maximum deflection was 0.08 mm. The maximum deflections and rotations were used as boundary conditions in a 3D solid FE model of the winding connection and the resulting stresses were calculated for bending and loading by centrifugal force (Fig.7). The most highly loaded point is located at the upper end of the current lead bolt with stresses of 310 + 76 MPa. These values agree within 10% with those measured during the component tests.

Component Tests

In order to demonstrate the component reliability of the contact-free winding connection. ABB performed component tests with the actual components. For this purpose, a test rig was developed and built, with which the service loading and multiples thereof could be simulated (Fig.8). The test rig was designed in such a way that two winding connections could be tested simultaneously. Of course, together with the current lead bolts and the steel conductors, both fixations (support element to the retaining ring plate and the connection of the self-supporting current lead bolt in the lead in bar) were also tested in the actual design.

The rotating bending loading was applied by a vibrating table, which was excited by an eccentric weight. The table was designed in such at way that the deflection of the bolt tip, which in fact describes an elliptical path for each rotation of the rotor, could be exactly simulated. The amplitude of the deflection was determined by the eccentric weight and precisely set by the rotational speed, so that the testing frequency corresponded approximately to the operating speed of 1800 rpm. The centrifugal force was simulated with the help of hydraulic cylinders, which were connected to the bolt tip by loading bars.

Therefore the first rotor with this new design solution went into operation, the components of the new winding connection were tested for more than 100 million, fatigue cycles at three times the nominal load, i.e. 0.6 mm and 0.24 mm deflection of the bolt tip. During the test, the hydraulic prestress was applied in such a way that the bolts were stressed to the maximum centrifugal force over their entire lengths. In operation, the static loads are much lower at the most highly loaded point, i.e. just below the winding connection, since the centrifugal force at this location is significantly reduced. With a test series of eight current lead bolts, it was demonstrated that the new solution is able to support a bad equivalent to three and a half times the service load for an unlimited number of loading cycles.

Design of the new Rotors

The measures described above were necessary for the refurbishment of the available rotors, in order to satisfy the requirements for safe operation. Without these conversions, unacceptable losses would have resulted from the consequent interruption of production. The modified rotors are in operation without any limitations.

For contractual reasons, Ontario Hydro and ABS have agreed to replace the four original rotors with new rotors of new design over a period of time.

The new design concurs with the standard for new turbo-generators at ABB. Fig.9 shows a sketch. The most important characteristics and alterations in comparison with the old design

are as follows:

1. New optimised design of the exciter winding permits a reduction of the slot depths by 40 mm with a simultaneous reduction of the exciter current losses.
2. As a result of the reduced rotor slot depth, a considerably larger shaft diameter is possible in the transition area of the rotor body.
3. The current lead bolt is located in the area with the largest shaft diameter; for this reason, the maximum loading is reduced to less than half of that with the older design of rotor. The connection of the winding is made directly.
4. The shaft end is designed approximately as a "beam with constant bending load". This ensures the required flexibility of the shaft ends in order to keep the second critical speed sufficiently below the operating speed.
5. In the region of the reduced shaft diameter there are no radial holes which could lead to a stress concentration. The area of the shaft ends has less than 25% of the maximum stresses of the older design.

Summary

The occurrence of a crack in the shaft end of the generator rotor in unit 2 of the nuclear power plant Darlington shows that it is still not possible to exclude failures with a sufficient level of certainty. Real robustness of the design requires a complete knowledge of relevant characteristics values. loads. etc. Despite decisive improvements in the areas of the tools employed, such as FE calculation methods, there are still limitations on this comprehensive knowledge. The age-old understanding of robustness in terms of large cross-sections is totally inadequate. This is shown particularly by the experience in other areas where there are exacting safety requirements.

The robustness of a product must be ensured through the robustness of all stops involved in its manufacture. As a result this applies to the engineering process, the forging process, the actual fabrication process, and in the same way to assembly, commissioning and finally commercial operation. For the future, a great effort must be made to achieve major improvements. This is only possible through co-operation between all partners involved, i.e. the manufacturer along with sub-suppliers and customers.

Until the necessary robustness is ensured, failures of large machines will not be totally eliminated. For this reason, the following capabilities will be decisive in providing a competitive advantage:

o Early recognition of faults by significantly improved on-line monitoring.

o Rapid and complete analysis of the cause of failures.

o Ability to rapidly develop reliable methods of refurbishment to limit the costs of lost generating capacity as a result of failures.

o Ability to completely revise current designs to provide improved robustness.

After this failure ABB has shown that in close co-operation with the customer the consequences of such failures can be limited, provided that the above-mentioned capabilities are fully applied.

FIGURE AND TABLE CAPTIONS

Fig.1: Crack in the non-drive shaft end of the generator rotor Darlington 2.

Fig.2: initiation point and size of the cracks.

Fig.3: Fretting fatigue cracks in the wedge lands of the winding connection slots.

Fig.4: Broken-open fretting fatigue crack in wedge land of slot 3.

Fig.5a: Contact free winding connection.

Fig.5b:"3D solid model" of contact free winding connection.

Fig.6: Displacement of the bolt due to bending of the rotor.

Fig.7: Loading of the contact-free winding connection.

Fig.8: Component test with actual components.

Fig.9: New ABB standard design for 4-pole rotors.

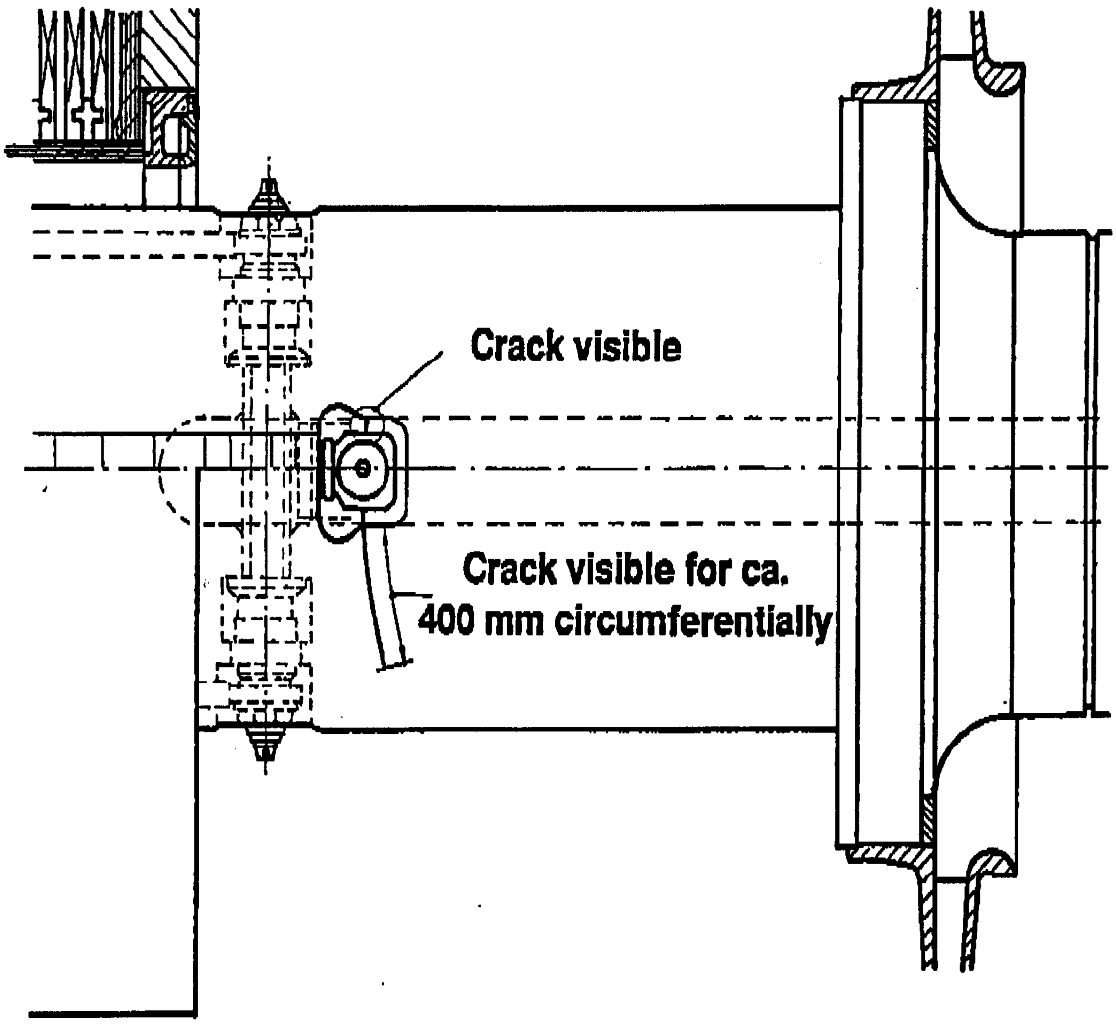
Crack visible
Crack visible for ca.
400 mm circumferentially

Fig. 1

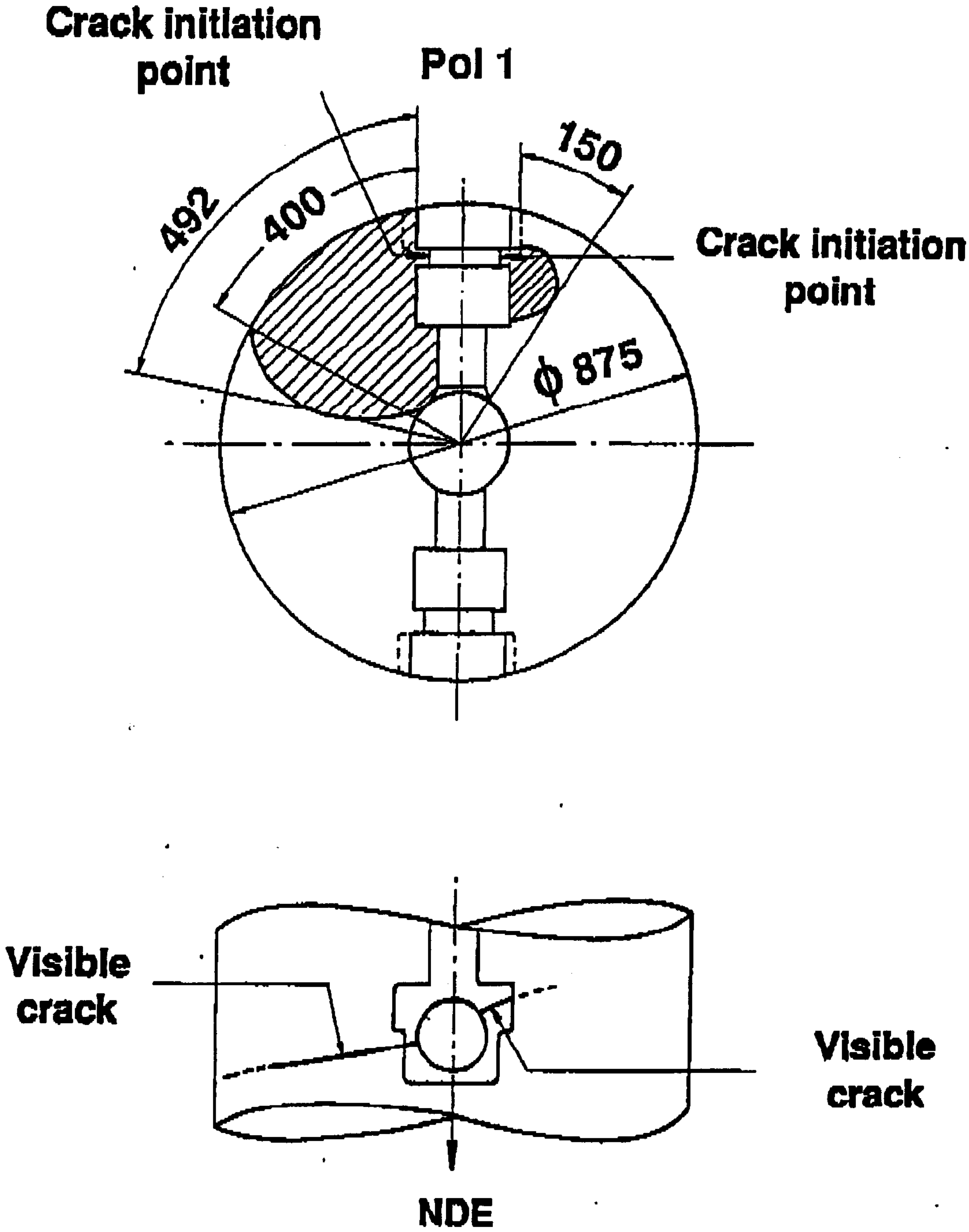
Crack initiation point
Pol 1
150
492
400
Crack initiation point
ϕ 875
Visible crack
Visible crack
NDE

Fig. 2

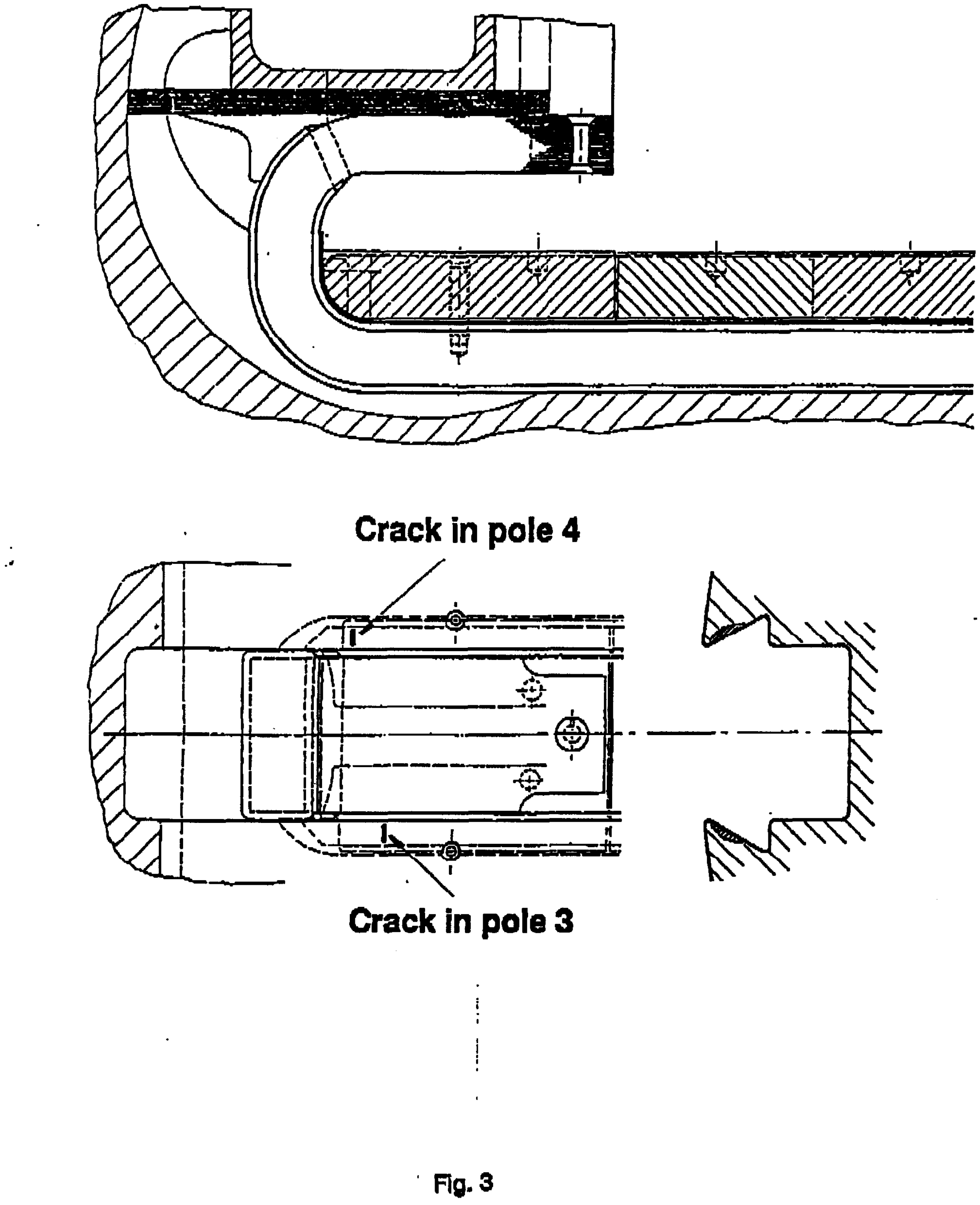

Fig. 3

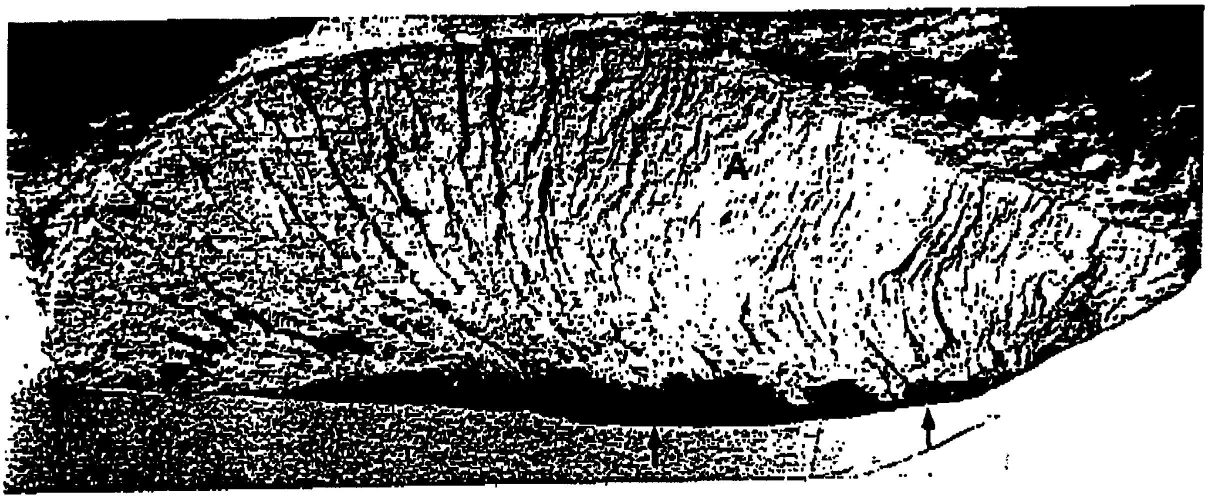

Fig. 4

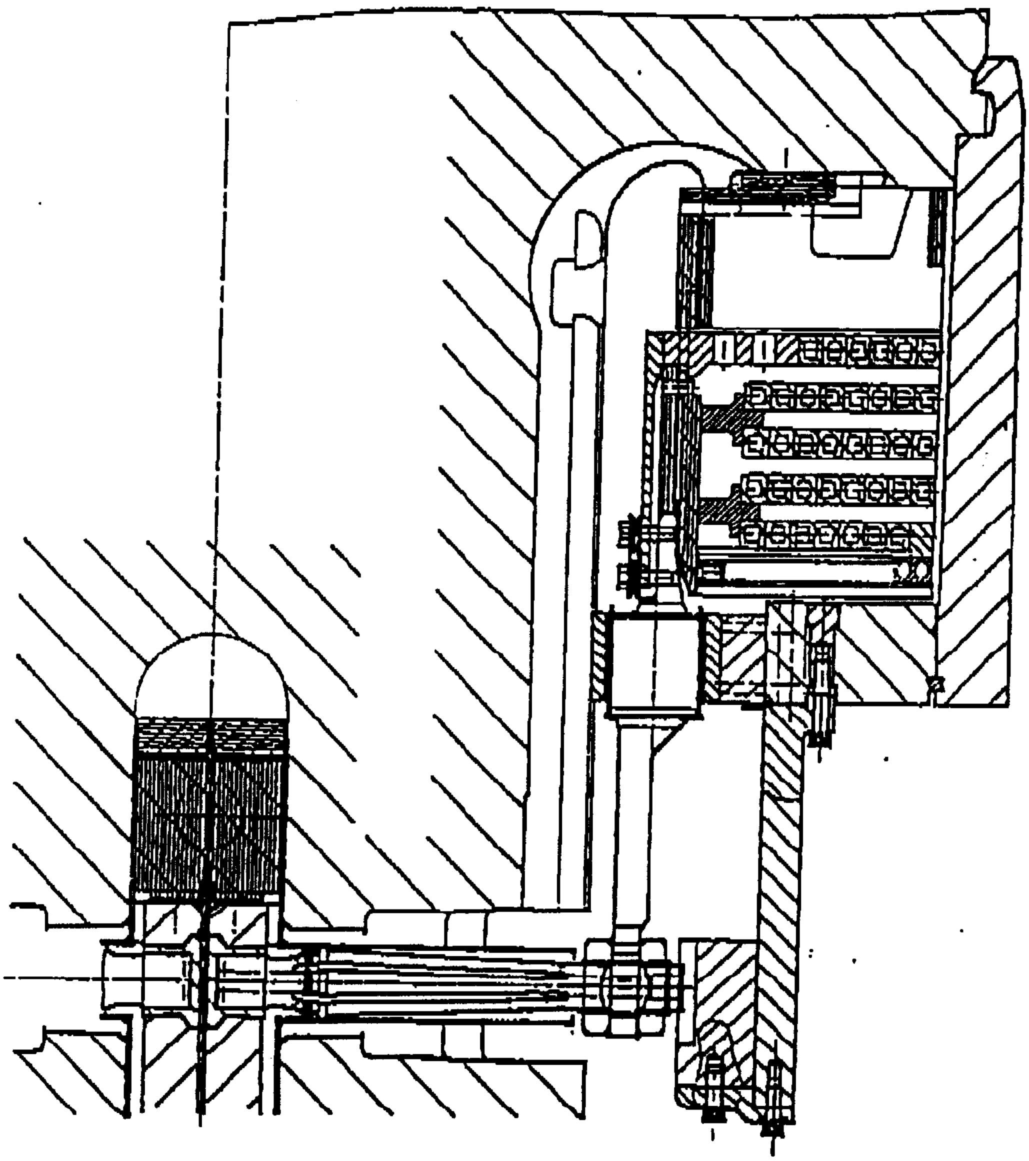

Fig. 5a

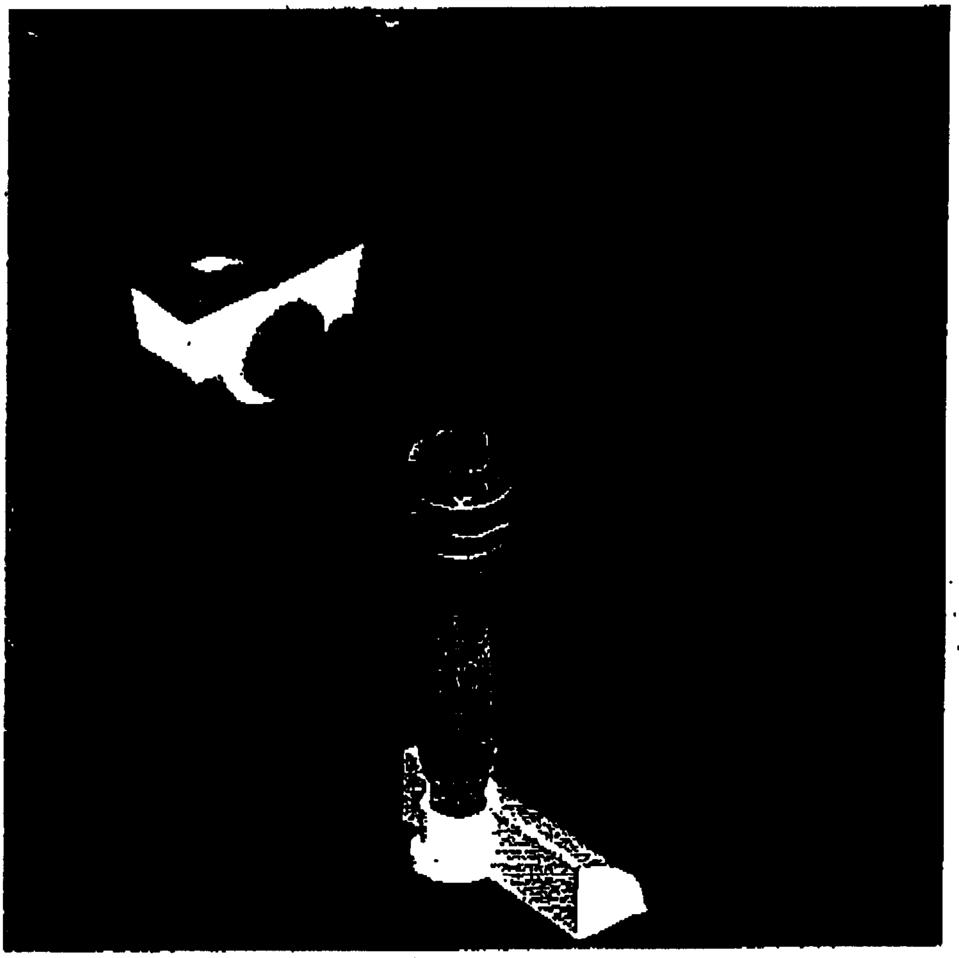

Fig. 5b

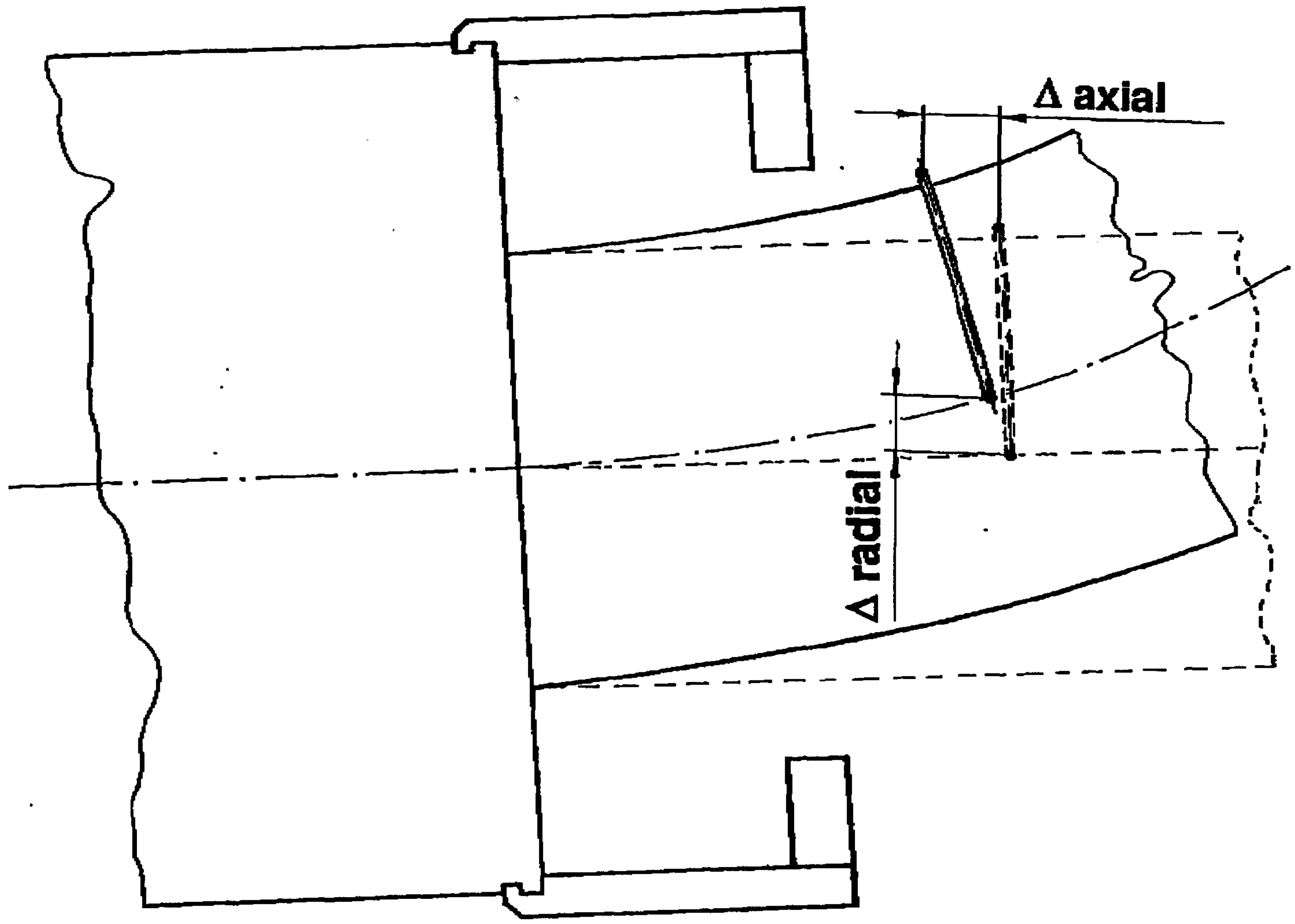

Fig. 6

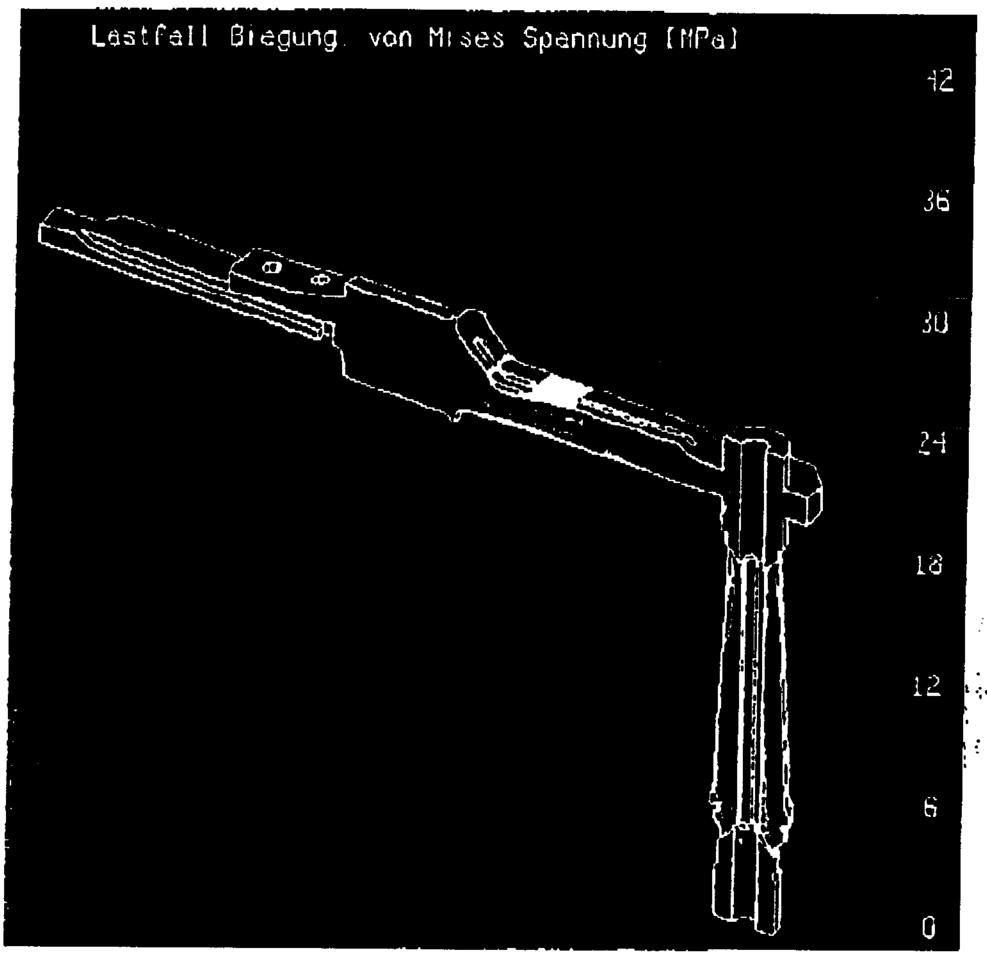

Fig. 7

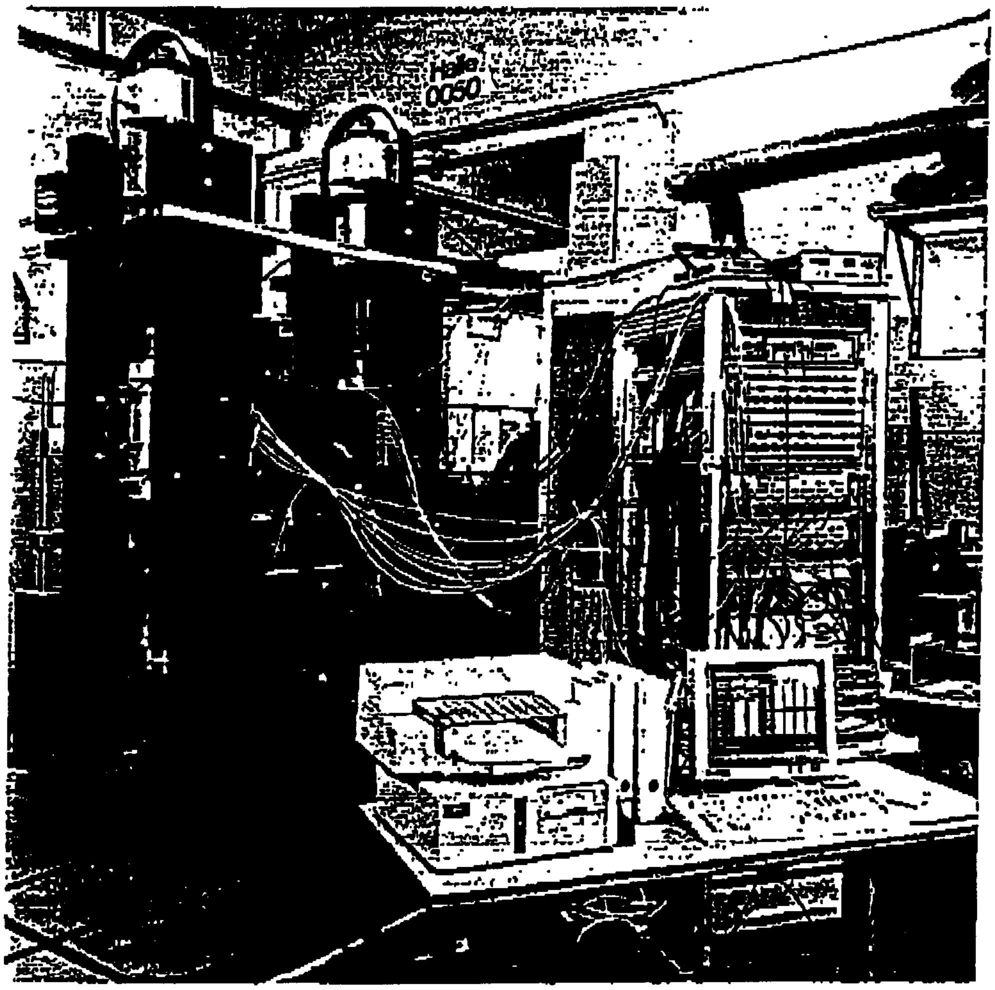

Fig. 8

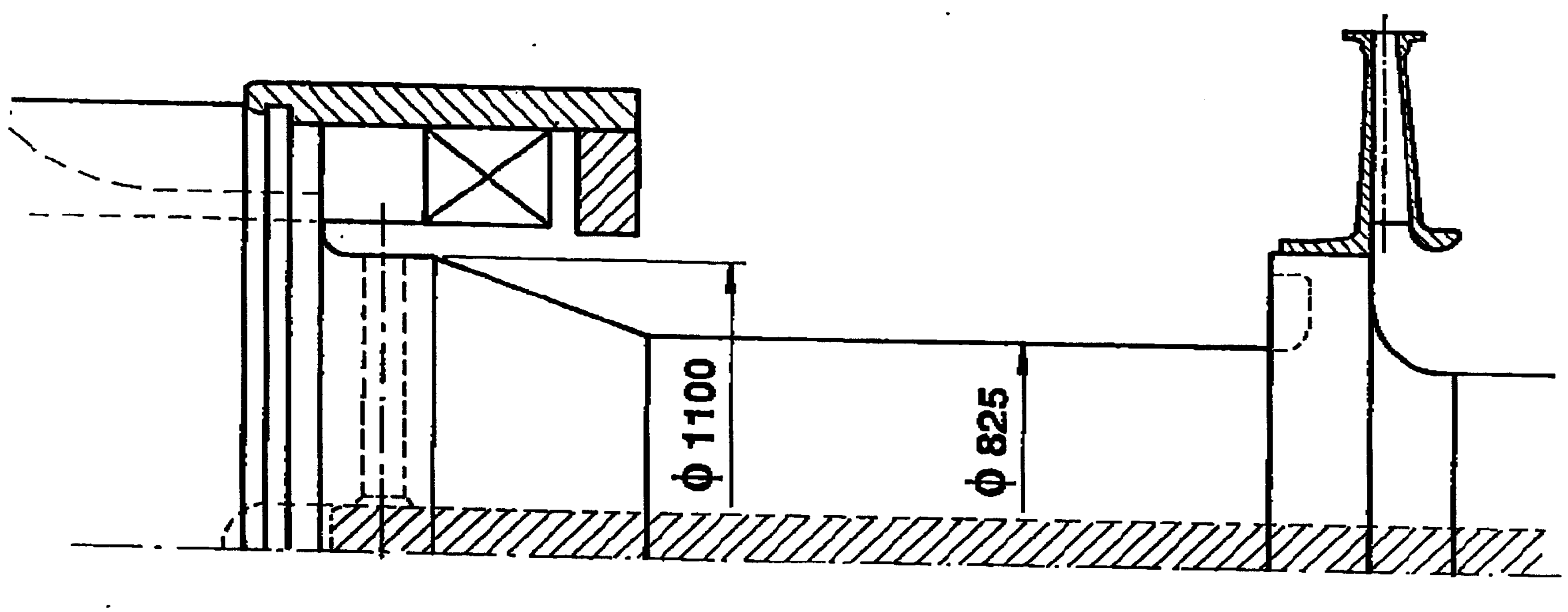

Fig. 9