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EXPERIMENTAL EVALUATION OF THE LOW CYCLE FATIGUE OF THE TURBINE DISC OF A LOW POWER TURBOSHAFT ENGINE

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Experimental evaluation of the Low

Cycle Fatigue Life of the turbine

disc of a low power turboshaft engine

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SUMMARY

The criteria of disc fatigue life prediction for gas turbine engines are, presently, the subject of much world-wide debate.

The experimental data collected during the past twenty years or so have shown the high reliability of crack initiation-based life evaluations, but also their excessively conservative nature.

For these reasons a number of authors(including several certification authorities) are beginning to develop modified lifing criteria and the trend is towards the application of fracture mechanics principles to evaluate the crack propagation life of engine components; the aim of these new methods are to utilise the full material performance without any compromise in safety standards.

Recently, the "2/3 of burst life" method of establishing component safe life has become acceptable. Instead of taking the life to first crack as the design criterion, component tests are run onto burst and the design life is assumed to be 2/3 the number of cycles to failure.

It is clear that this method takes into account, as its implicit condition, the material's toughness under small crack propagation conditions.

This new method was used by Alfa Romeo Avio Stress department in the life clearance of the first stage turbine disc of the AR 318-02 engine.

The most relevant steps of the analysis are briefly summarized here.

1. INTRODUCTION

Large rotating components in aero gas turbine engines are the most critical structural components. Their integrity is vital to the safety of the aircraft and, of necessity, they are highly stressed to meet the stringent demands of weight and engine efficiency.

Turbine discs and compressor impellers of the latest aircraft gas turbine engines are generally made from complex and expensive superalloys (e.g. Ni or Ti based alloys) in either wrought or cast form (3). These materials exhibit good mechanical and creep properties up to 1000÷1100°C so that, under the action of the usual complex load history, low cycle fatigue (LCF) has become the life limiting factor for engine discs in over 75 percent of designs(2). More recently, the combined effect of creep, fatigue and oxidation and their synergistic interactions has become of more concern(4).

Therefore both manufacturers and certification authorities attach very much importance to LCF life evaluation methods.

The conventional method to predict component LCF life was developed during the past twenty years and is based on the so-called "safe life" approach. It is a first crack criterion in which each component has a LCF life that is equivalent to a probability of 1/1000 of initiating a detectable surface crack ($\simeq$ 1/32 in) during its design lifetime.

In practice 99.9% of all discs are removed from service when they are still structurally sound and are rejected although they have considerable useful life remaining (1).

The criterion, therefore, may be safe but it is also excessively conservative(6).

Since gas turbine rotor components are among the most costly of engine components, several authors are developing alternative methods of establishing component safe life, that will be based on a damage-tolerant approach. The common idea is to apply fracture mechanics to model material behaviour in the sub-critical crack-growth range and so operate a component "life on condition"philosophy: only those discs are withdrawn from service which actually have a life-limiting crack. Among these new LCF life management philosophies, the well known "retirement for cause" methodology is at present under implementation (1).

However there is some reluctance on the part of engine manufacturers in applying the damage-tolerant based design concepts for critical components, because quantitative models of crack growth behaviour with a proven safety record have not yet been developed.

A recently fully accepted LCF safe-life evaluation method is the so called "2/3 of burst life method" (6). According to this method, component tests are continued until burst and the design LCF life is taken as 2/3 of burst life (rather than the first crack life). Without any compromise in safety *standards this method allows benefit to be taken from the additional crack-propagation life phase in which modern disc alloys can spend most of their lives.*

At the same time this method is effectively less expensive than the conventional ones, due to the reduced number of full scale tests necessary to establish component burst life.

In the following a LCF life assessment methodology based on the last mentioned criterion will be reported.

This refers to a turbine disc of the turboshaft version of the AR. 318 engine where an essential requirement was a satisfactory LCF life.

2. THE 1ST STAGE TURBINE ROTOR

The first stage gas generator turbine rotor is an integral design cast in INCO 792 + Hf(Fig. 1).

This Ni-based alloy exhibits good mechanical and creep properties up to 1070 °C and an excellent oxidation and hot corrosion strength.

HIP (hot isostatic pressing) was also performed to improve material fatigue performance.

A curvic coupling is machined on the foward side of the disc.

3. THERMAL ANALYSIS

The starting point in every LCF life evaluation is the definition of the loading spectra imposed on the rotating component when the engine performs the typical flight cycle.

The loading spectrum experienced by an engine disc is characterized, mainly, by low frequency stress cycling resulting primarily from centrifugal forces associated with variations in engine speed. In the hot section of the engine, stresses produced by thermal cycling must be superimposed.

The latter are functions of the instantaneous temperature distribution; therefore to perform the thermal stress evaluation it's necessary to carry out a disc thermal analysis for each engine operating condition(steady state conditionsas well as transients).

The thermal analysis is based on basic engine performance parameters as well as on engine and rig temperature measurements to determine heat transfer coefficients.

Typically thermal analyses are performed by the use of computer codes.

A typical temperature plot for the turbine disc under consideration is shown in fig. 2.

4. STRESS ANALYSIS

Disc stress analysis is now carried out combining the temperature distribution just discussed with all other relevant mechanical loadings.

The purpose of the stress analysis is to identify the most critical locations in terms of LCF integrity during a mission profile. At the same time the stress/strain worst cycles are to be evaluated as well as the engine operating conditions.

In this way the complex load history imposed on the component, for the purpose of LCF life evaluation, can be described by very few and simple stress cycles at the critical locations.

The above evaluations are essential for both theoretical LCF life prediction and component full scale testing which must cause failure of the component in the same mode and location as in the real load cycles.

The full stress analysis is carried out by finite element (FEM) computer programs and, to reduce computer time consuption, is performed in two steps:

LCF critical features of the component and the worst engine operating condition are firstly identified, stress analysis being based on a fairly coarse FE model; afterwords based on a model that is much more refined at the critical LCF locations, a detailed stress analysis at worst engine LCF condition is carried out.

In the latter step it's necessary to take into account each component load as well as the combined loads, so that their relative severity can be investigated. Since LCF life is sensitive to cyclic stress level, particular care must be taken with boundary conditions. Local plasticity effects and their resulting stress redistribution are also of concern.

For the turbine disc under discussion, a stress analysis was carried out by means of FEM using the SAP IV computer program.

In the preliminary stage the model in fig. 3 was used. This was a 2-D axisymmetric "blade spread" model. The combined load case resulting from the superposition of centrifugal, thermal and tightnening load was examined and several analyses carried out for both transient and steady state engine conditions.

The results of the above analyses showed that the most critical engine cycle in terms of disc LCF was the typical engine take-off cycle. During this cycle the worst engine condition occurs at 450 s after start-up when the engine is running at its maximum speed (N = 38.100 RPM) and the disc is subject to its highest thermal gradients.

The potential critical LCF locations are at disc bore and at disc rim (which is not slotted). Observation of the intertooth region of the curvic coupling showed that it might also be critical feature due to the stress concentration.

Therefore the detailed stress analysis was carried out based on the refined-bore area model shown in fig. 4. This 2-D model was also used as a boundary condition generator for the 3-D curvic coupling model(Fig. 5).

The results confirm the disc bore as the most critical LCF feature. Stresses at this location are summarizedin tab. 1 for the worst engine condition (at 450 s after start-up). The second LCF limit of the disc design occurs at the front curvic coupling intertooth region, where the peak stress was 735MPa. It should be noted that about 96% of the latter critical stress level is due to the centrifugal loading.

5. *COMPONENT EXPERIMENTAL TESTING*

a) *Spin pit testing*

Full scale tests are carried out to determine component burst life. Experimental testing is performed on an in-house cyclic spinning facility which is generally unheated.
Thus the spin pit cycle must be arranged such that the stress level at component LCF critical locations are the same as those under engine conditions.
The most usual life assessment testing approach is the "reduced" life test approach: this involves creating an overstress on the component so that it can be tested to a failed state in a short time (7).
The experimental results from a cold overstressed spin pit cycle are then adjusted to real engine hot condition using a theoretical cumulative fatigue damage model.
For the above mentioned turbine disc the burst life tests were carried out on a cyclic spinning facility operating at a uniform temperature of 50°C.
Each test was performed in 2 steps: in the first step a spin pit cycle was performed between 2000 and 48.000 (± 200) RPM so that at the disc bore a peak hoop stress of 895 MPa was created. An overstress was therefore obtained of about 13%.
As stated from the performed stress analyses, the front curvic intertooth region was found to be sensitive to centrifugal load. At the spin pit maximum speed condition an artificially high stress of 918 MPa(overstress ≃ 25%) was evaluated in this region.
At such an extreme condition a premature failure at the curvic intertooth region was to be expected.
Preliminary crack initiation life evaluation (according to the Manson-Coffin LCF method) predicted a first crack occurance after 4300 spin pit test cycles (fig. 7).
In order to allow testing to continue to investigate disc LCF life as dictated by the disc bore critical area it was decided to modify disc geometry (Fig. 6);

cutting off the front curvic at the time failure occurence.
In the second step each modified disc was then run on to burst through a new spin pit test cycle with its maximum speed of 49.800 (± 200) RPM, where the bore peak hoop stress was 941 MPa. The overstress in the latter test step was 19%.
A set of 4 discs was tested, test results being summarized in table 2.
It should be emphasized that good agreement was obtained between theoretical and experimental results for the test cycle number at curvic failure(1st test phase).
Fig.'s 9, 10 show typical cracking for the curvic as well as for the disc.

b) Fractographic Examination

A set of test specimens was machined from the failed discs and tested.
For the purposes of the remaining calculations the most important conclusions are:

1. Failure of the tested stage 1 turbine disc occurred from an area of fatigue propagation that had nucleated at approximately the center of the disc bore; propagation extended to a depth of 6.5 mm prior to burst.
2. Two other areas of fatigue propagation were also present at the center of the disc bore and these extended to depths of 3 mm and 5 mm respectively.
3. In all three instances fatigue propagation had occurred in an essentially structure sensitive mode precluding a detailed striation count being undertaken. However isolated area of fatigue striation were apparent alongside propagation facets and their spacing indicated the order of 3000 cycles of crack propagation prior to failure. No evidence of a superimposed high cycle fatigue load was apparent in the fatigue propagation area.

6. COMPONENT LCF SAFE LIFE EVALUATION

Assuming that disc fatigue behaviour could be similar to the material behaviour, based on the available crack propagation data, typical and minimum stress versus disc life curves can be plotted (Fig. 8).

It should be noted that for a rotating disc the hoop stress is the most relevant crack growth governing stress.

In this way the spin pit testing results can be adjusted to the engine stress level using Miner's rule (Tab. 2).

Fig. 8 shows that the initial assumption should be reasonable.

Being conservative it can be assumed that the minimum number of equivalent engine cycles is equal to the component burst life.

The design life of the analysed turbine disc is therefore(Tab. 2):

Design life = DL = 2/3 x 11.585 = 7.723 engine cycles

To have the LCF safe predicted life of the component, fatigue scatter *factors are required to relate actual test results on randomly chosen components to a predicted result for a component of minimum fatigue strength.*

This scatter is of a statistical nature.

For the sake of simplicity we can consider the factors quoted in BCAR (British Civil Airworthiness Requirements) document (see BCAR section C3 Ap.2).

These factors are for crack initiation whereas experience has shown that the large majority of disc cyclic life is in crack propagation.

Crack propagation exhibits a much narrower scatter on results than crack initiation which is affected by surface finish. Thus, application of the BCAR scatter factor is pessimistic.

Because of the total number of performed spin pit tests(4 discs), a factor of 2.5 is applied in our calculations.

The predicted safe life(PSL) for the turbine disc is finally:

$$PSL = \frac{7723}{2.5} = 3.089 \text{ engine cycles.}$$

7. CONCLUSIONS

Use of the 2/3 burst life criterion is standard practice in many engine manufacturer design offices.

A practical methodology has been discussed which uses this technique and avoids any sophisticated statistical analysis.

At the same time it's able to provide the benefits from the above LCF life evaluation criterion as well as complying with both the CAA (British Airworthiness Authority) and FAA (Federal Airworthiness Authority) requirements for life clearence of engine rotating components.

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FIG. 1
1ST STAGE
TURBINE DISC

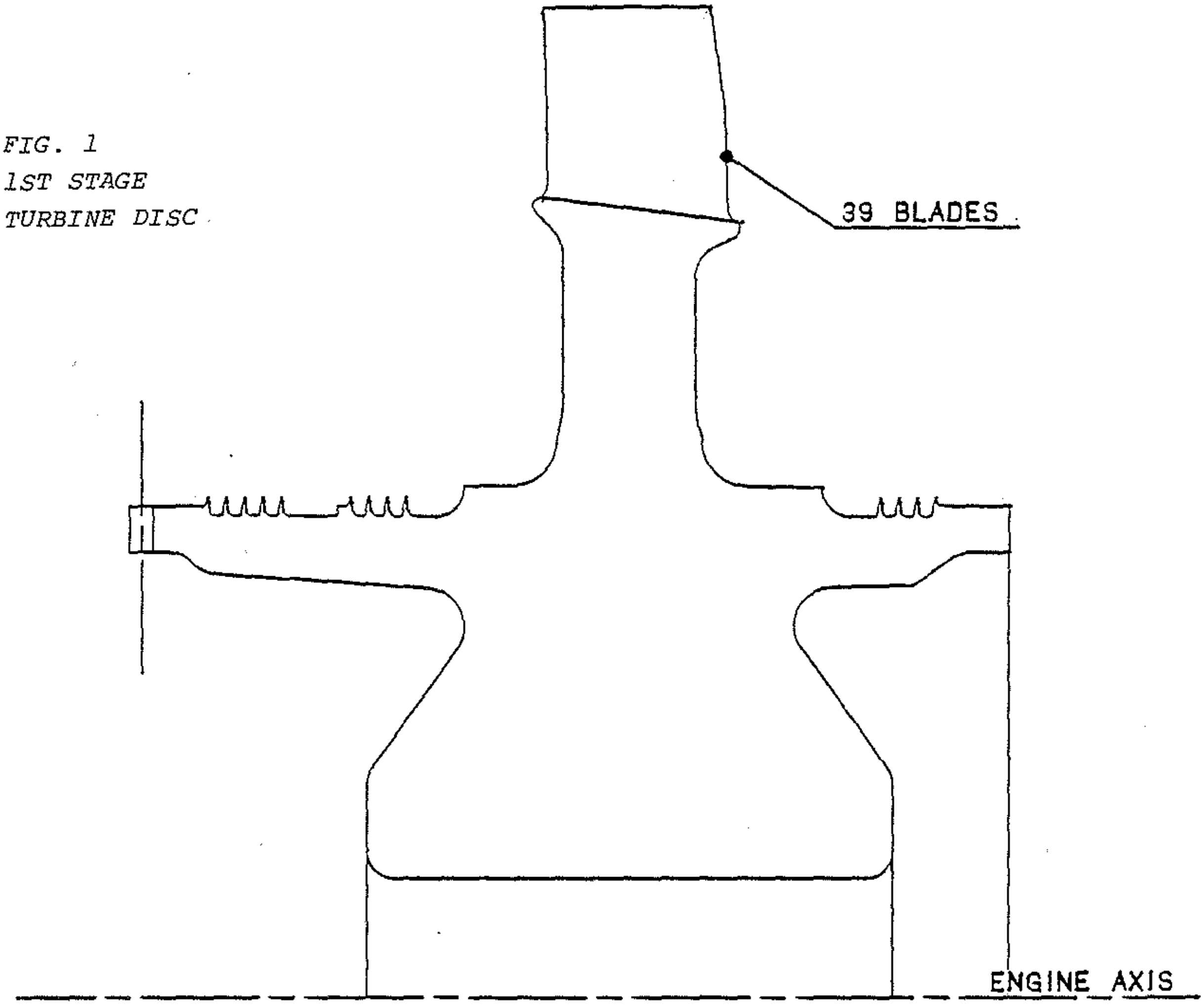
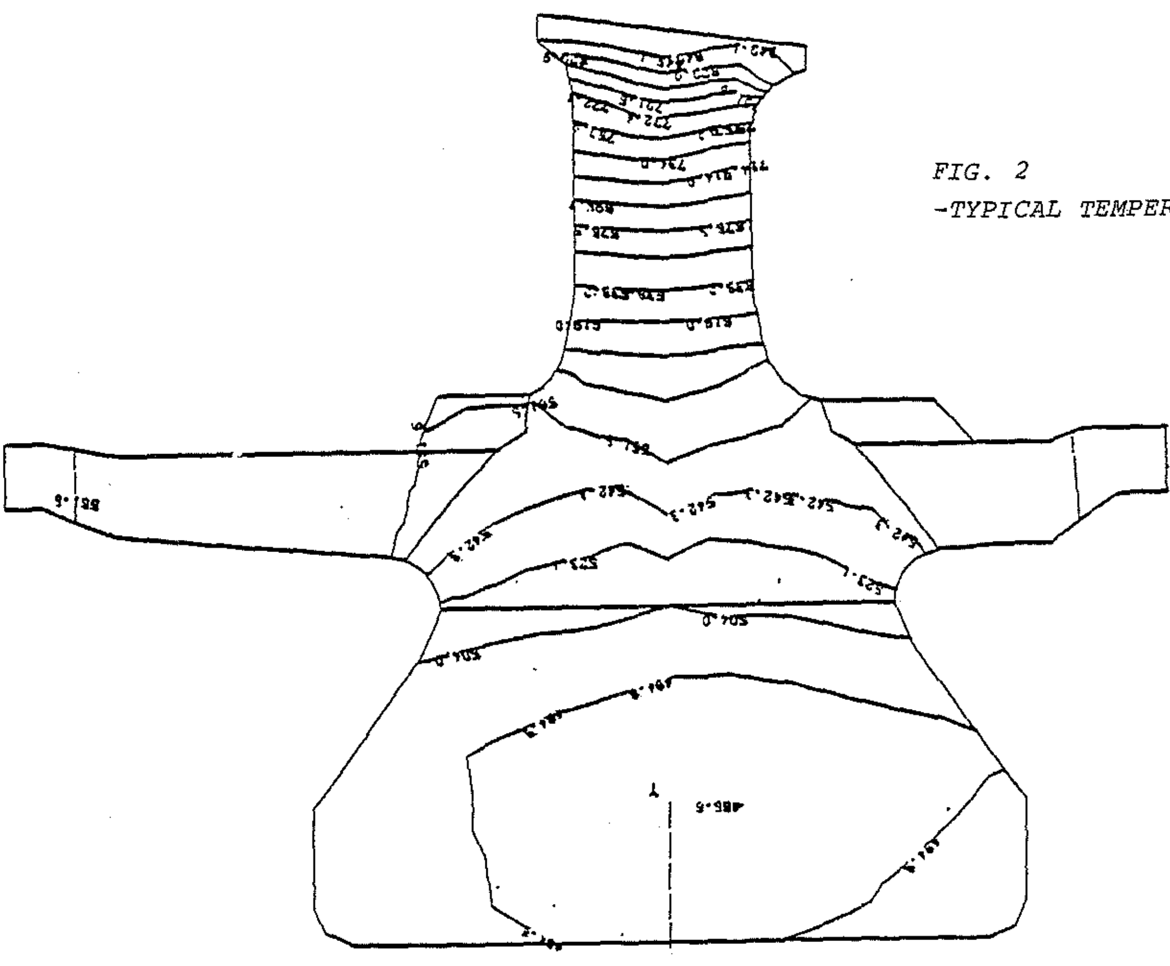


FIG. 2
-TYPICAL TEMPERATURE PLOT-



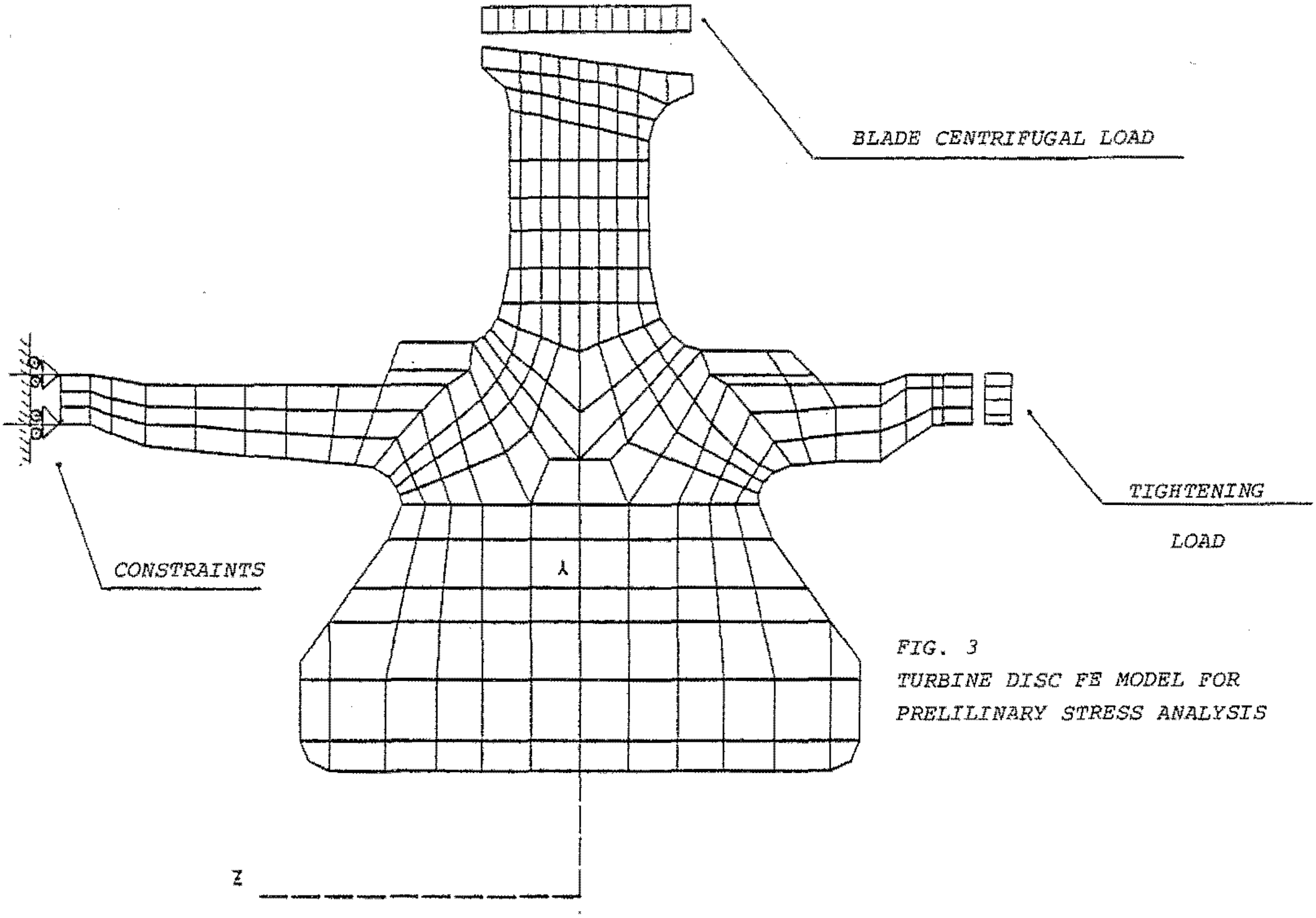


FIG. 3
TURBINE DISC FE MODEL FOR
PRELILINARY STRESS ANALYSIS

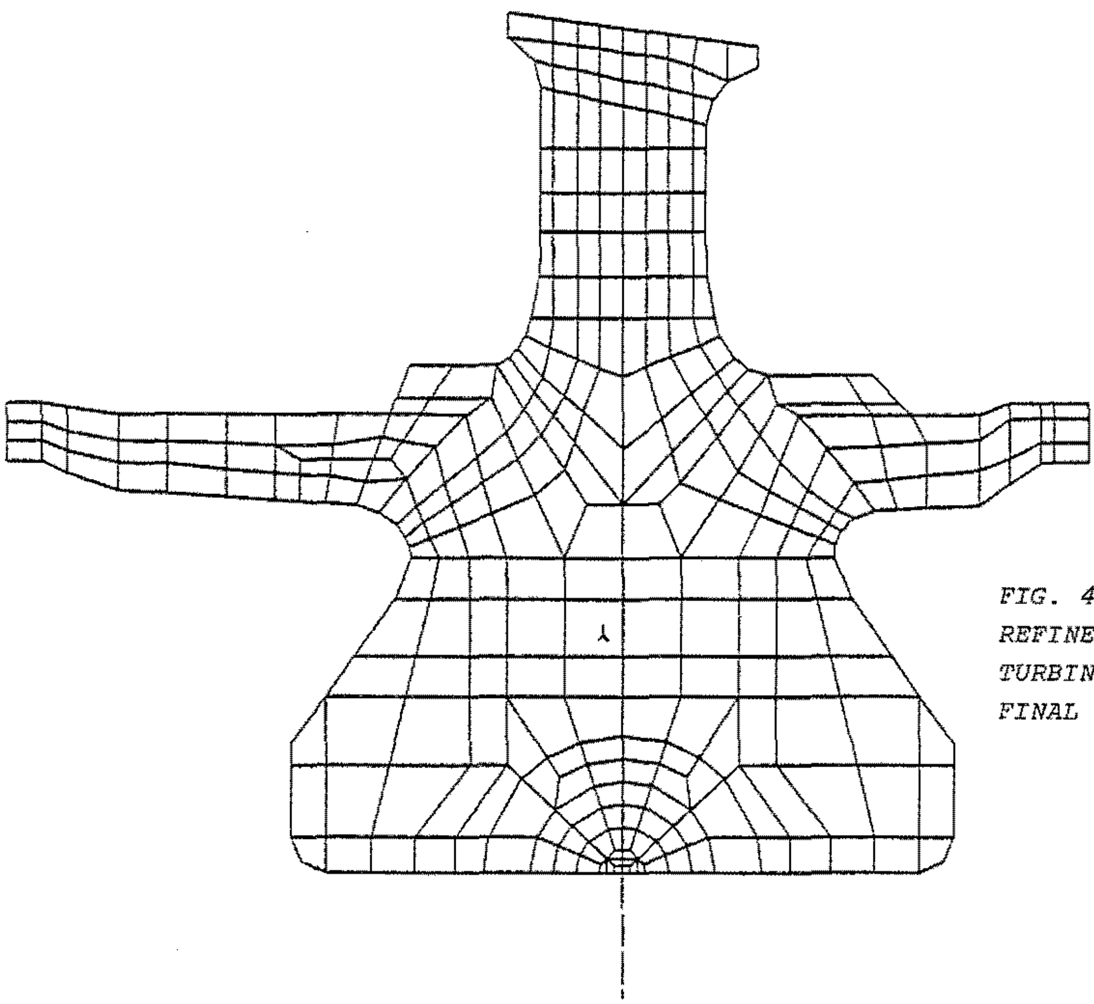


FIG. 4
REFINED-BORE AREA OF THE
TURBINE DISC FOR THE
FINAL STRESS ANALYSIS

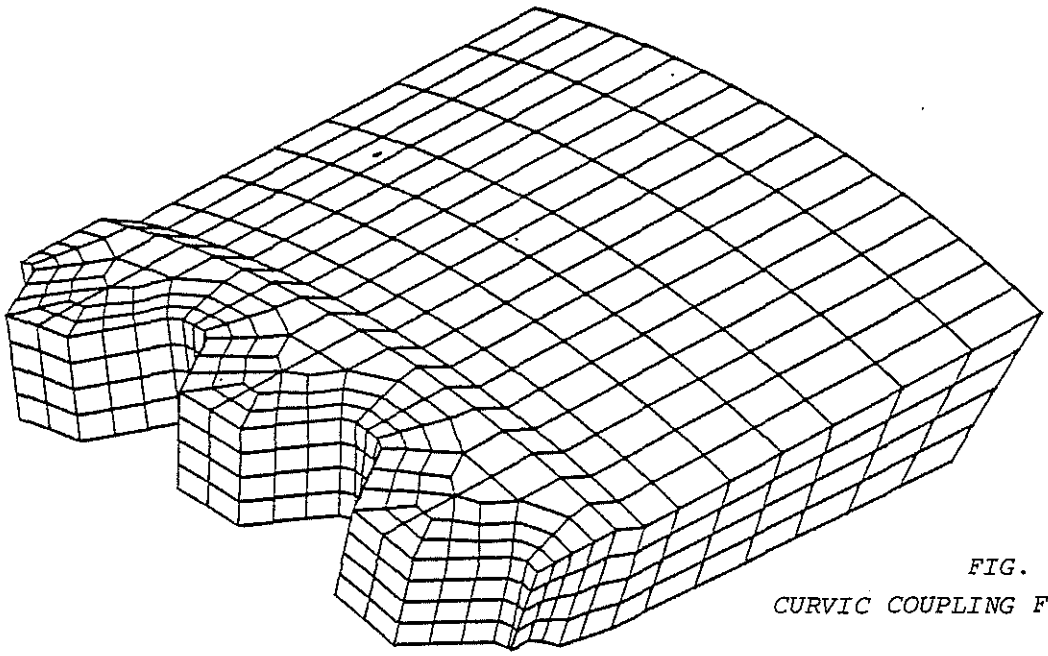


FIG. 5
CURVIC COUPLING FE MODEL

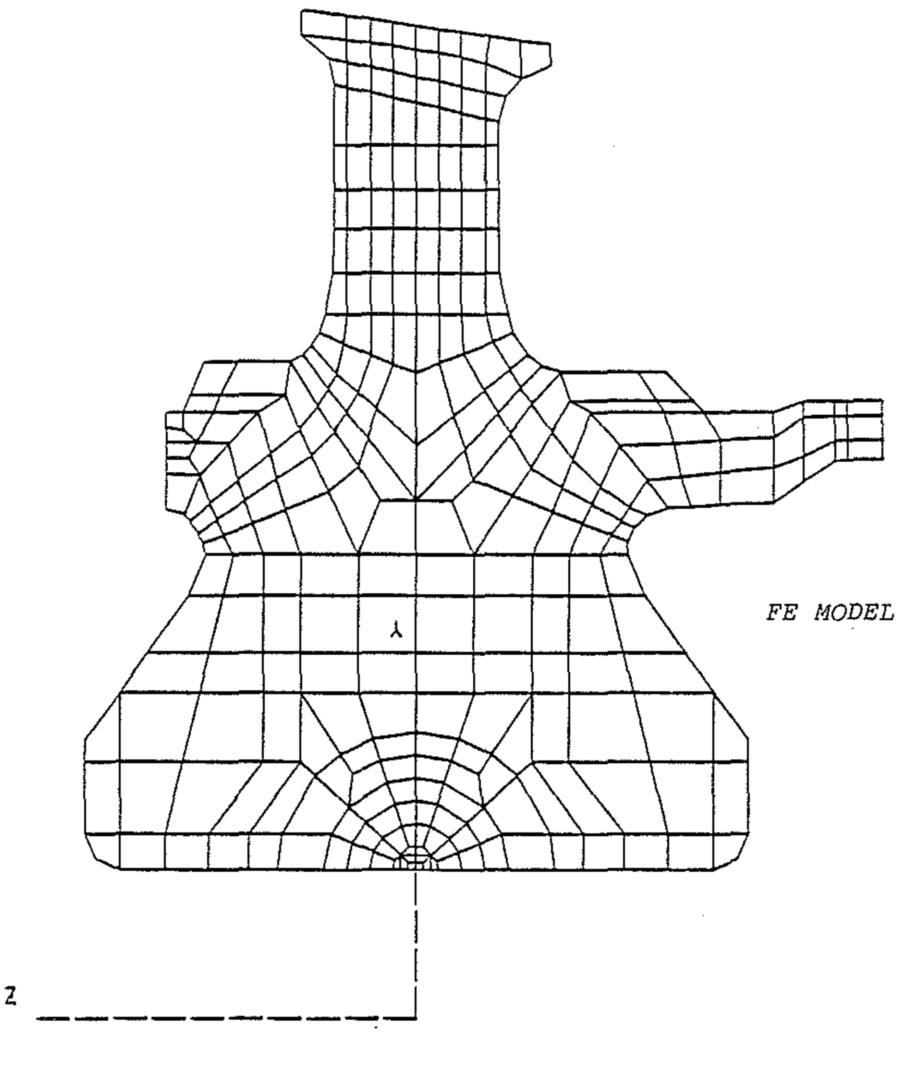


FIG. 6
FE MODEL OF THE MODIFIED DISC

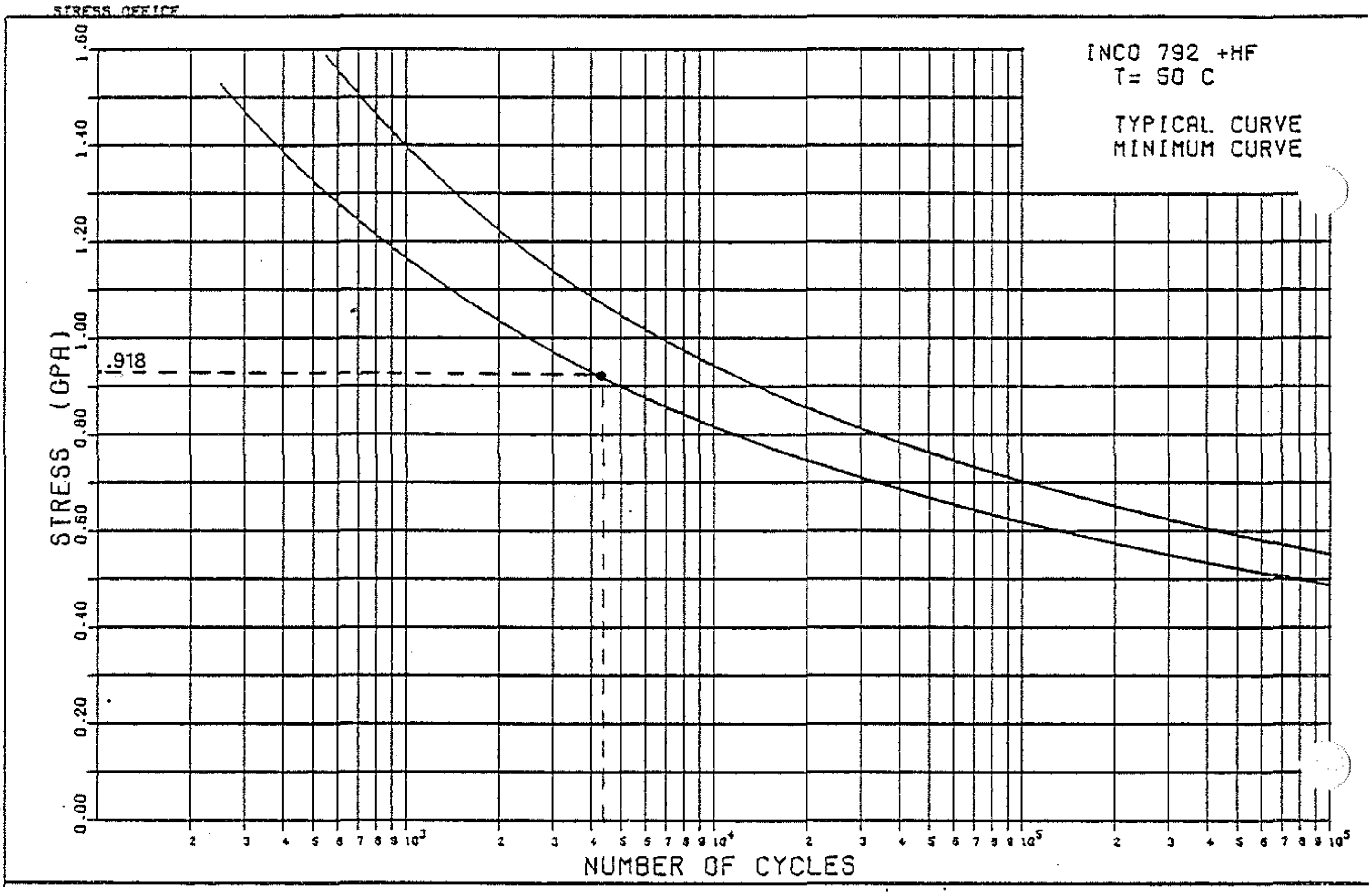


FIG. 7 CRACK INITIATION FATIGUE CURVES FOR INCO 792+Hf(MANSON-COFFIN METHOD)

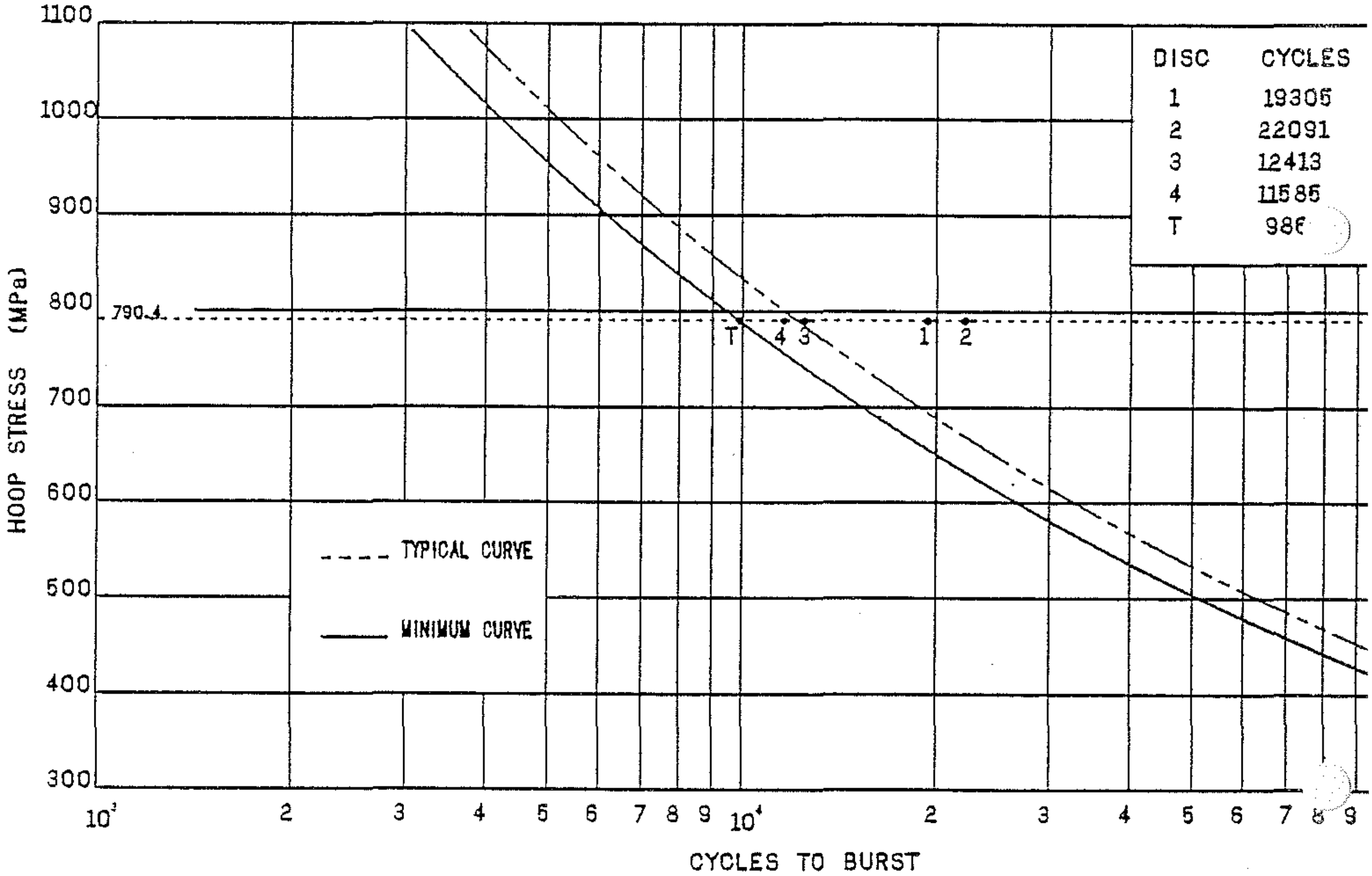


FIG. 8 TURBINE DISC BURST LIFE CURVES

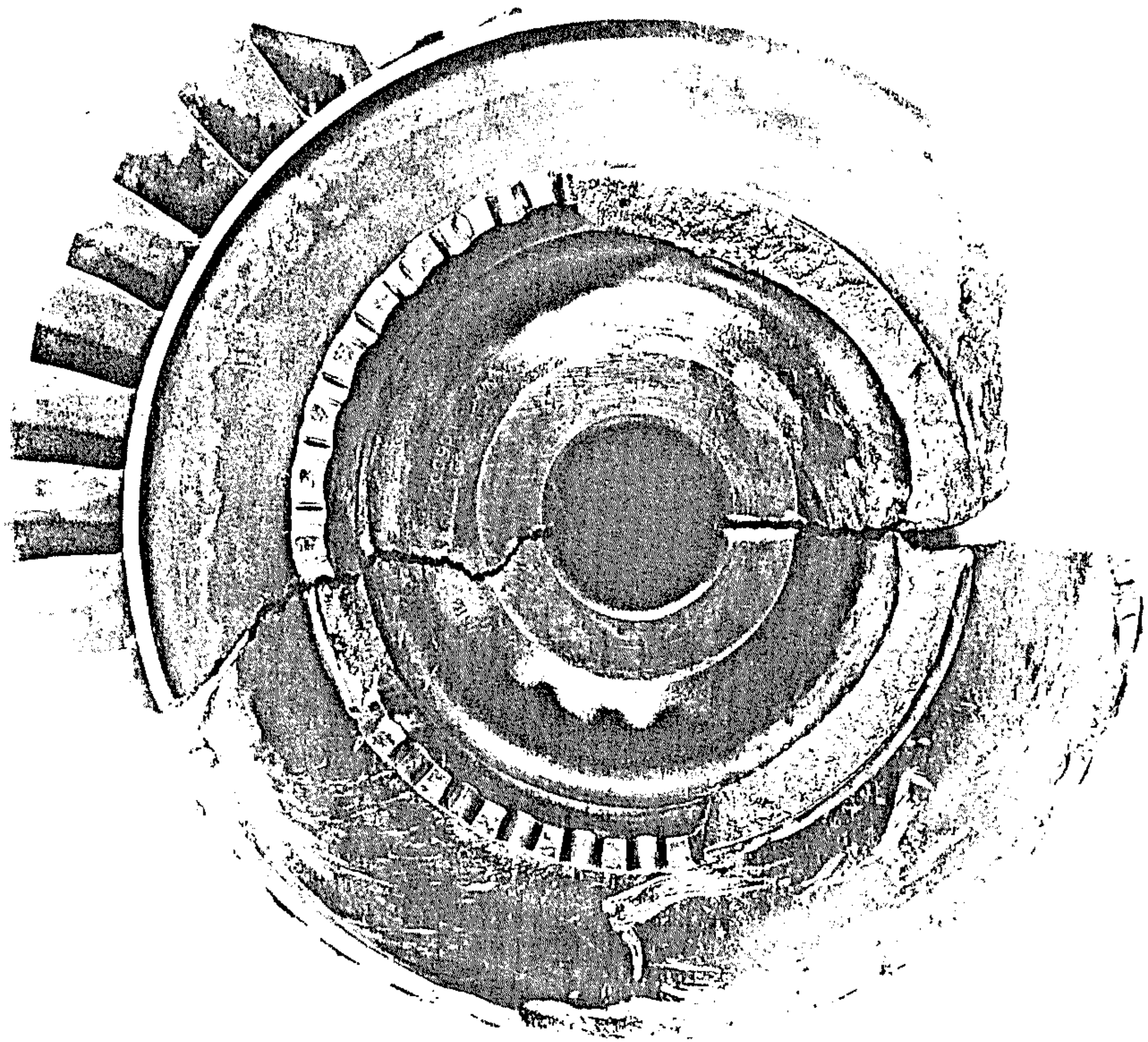


FIG. 9 TYPICAL VIEW OF A TURBINE DISC BURSTED AT SPIN PIT FACILITY

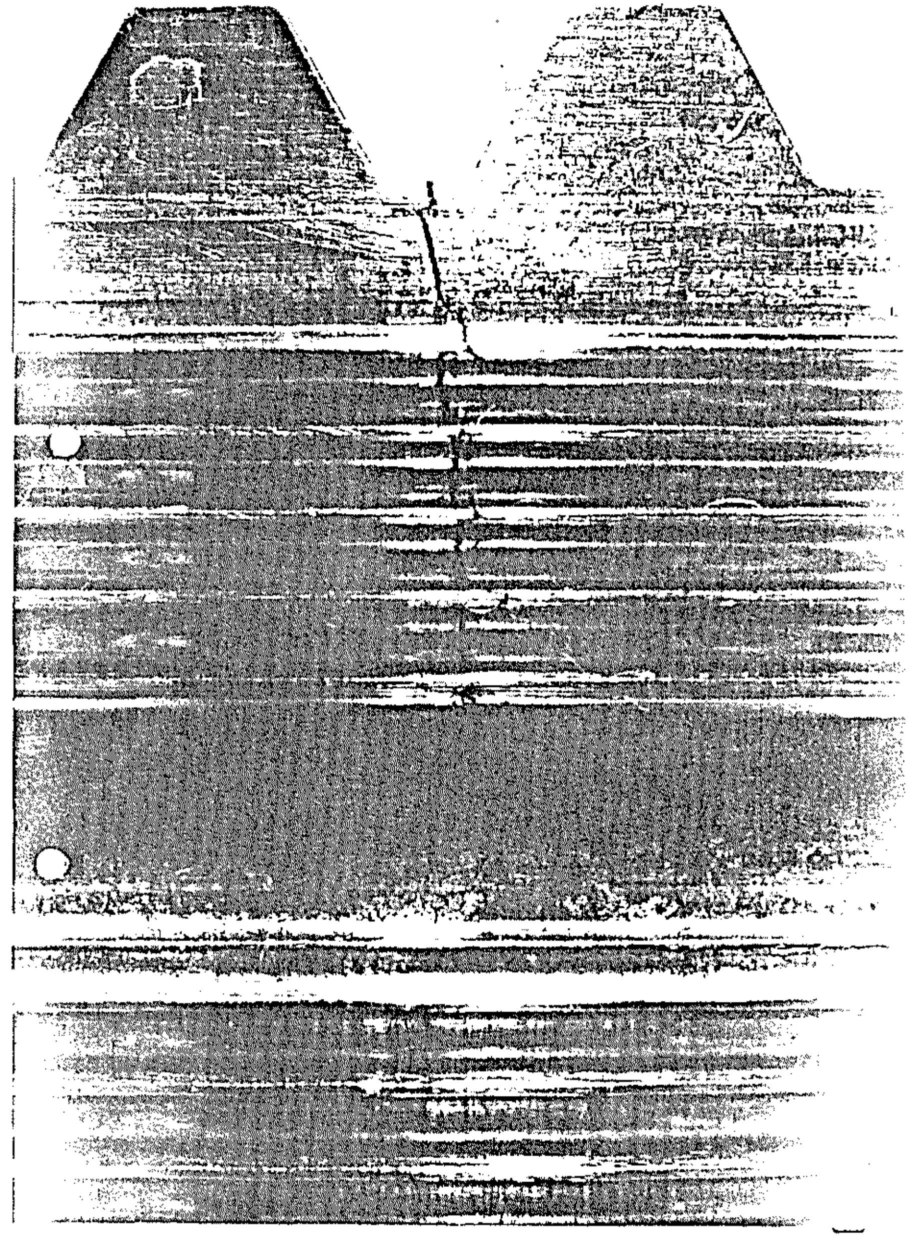


FIG. 10 TYPICAL CRACKING BETWEEN CURVIC TEETH

	STRESSES AT BORE (MPa)			
LOADINGS	σ_{HOOP}	$\sigma_{PR.MAX}$	$\sigma_{PR.MIN}$	$\sigma_{V.MIS.}$
CENTRIFUGAL (N=38100 RPM)	564.2	3.0	-107.1	623.5
TIGHTENING	0.3	1.5	0.	1.3
THERMAL	225.8	-3.6	-126.0	309.3
COMBINED	790.4	0.4	-231.6	928.2

TAB. 1

HOOP, MAXIMUM AND MINIMUM PRINCIPAL STRESSES AND VON MISES EQUIVALENT STRESS AT DISC BORE FOR THE WORST LCF ENGINE OPERATING CONDITION (450 S AFTER START UP)

SPIN PIT TEST RESULTS					
TESTED DISC No.	1ST PHASE (48000 RPM)		2ND PHASE (49800 RPM)		EQUIVALENT ENGINE CYCLES (ENGINE STRESS 790.4 MPa) *
	RIG STRESS (MPa)	RIG CYCLES	RIG STRESS (MPa)	RIG CYCLES	
1	895.5	4500	941.	6521	19305
2	895.5	5277	941.	7355	22091
3	895.5	4000	941.	3268	12413
4	895.5	3500	941.	3245	11585

* USING MINER'S LAW

TAB. 2

SPIN PIT TESTING RESULTS ADJUSTED TO ENGINE STRESS LEVEL