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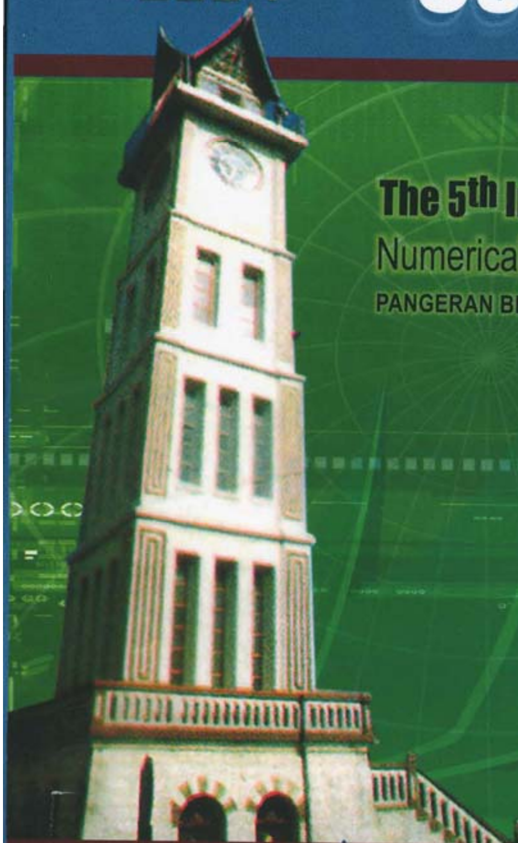

The 5th International Conference on Numerical Analysis in Engineering 2007 (NAE2007)

PANGERAN BEACH Hotel | PADANG, West Sumatra | INDONESIA | 18-19 May 2007

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- Fracture Behaviors
- FEM in Forming Process
- Computational Mechanics
- Static & Dynamic Problems
- The Atomic/Molecular Dynamics
- Biomechanics Problem
- Response of Biological Materials
- Analysis of Machine Element Design
- Computational Methods in Chemical Engineering
- Computational Methods in Electrical and Electronic Engineering
- FEM Application in Geotechnical & Structural Engineering
- Numerical & Experimental Fracture Mechanics
- Numerical Analysis Tools for Web-Based Applications
- Computational Methods in Thermo & Fluid Mechanics
- Artificial Intelligence Application in Engineering, such as: Expert System; Pattern Recognition; Neural Network; Genetic Algorithm; etc.
- Noise and Vibration Analysis

DAY 2 – SATURDAY, 19 May 2007 (continue)		
	VENUE A (Regular Conference): Sati III Room (First Floor)	VENUE B (Regular Conference): Basa III Room (First Floor)
10.40-12.40	Session A. 7. 2: BIOMECHANICS IMPACT PROBLEMS ; BIOLOGICAL MATERIALS ; Chairperson: Dr. Eng. Sandro Mihradi	Session A. 8. 2: NOISE AND VIBRATION ANALYSIS ; ANALYSIS OF MACHINE ELEMENT DESIGN Chairperson: Dr-Ing. Ikwansyah Isranuri
Keynote Speaker	1. Enriched Free Mesh Method: Patch By Patch Stress/Strain and Displacement Improvement ***G. Yagawa[1] and H. Matsubara[2]***	1. Improving the Enhancement of Corrupted Signal in Noise: An Optimation of Blind Deconvolution EVA Method by Reducing Samples Size Based on Maximum Crest Factor ***Nirbito W., Sumardi T.P., and Tan C.***
	2. Evaluation and Characterization of Engineering Thermoplastics- Male Sheet Type Components under Impact ***M. Nizar Machmud, Masaki Omiya, Hirotsugu Inoue and Kikuo Kishimoto*** 3. Properties of Bagasse Fiber Reinforced Polyester Composite Before and After Fiber Treatment ***V. Vilay, M. Marianti, R. Mat Taib, K. Phasomsouk and Mitsugu Todo*** 4. Micellar Polymer Flooding for Improving Oil Recovery ***Ahmad Tawfiequrrohman Yiliansyah, Wahyu Hasokowati, Suryo Purwono and Bardi Murachman*** 5. Impact Respon of GFRP Human Head Model Subjected to Impact Loading ***Bustami Syam, Hendri Nurdin, Samsul Rizal and Basuki W.S.***	2. Conceptual Design and Analysis of Automotive Bumper Beam ***Mohd Nizam Sudin, Mohd Fadzli Abdullah, and Shamsul Anuar S.*** 3. Analysis and Redesign of Door Lock Using Finite Element Method ***Jaya Suteja and Sunardi Tjandra*** 4. Designing Material Handling Fixture for Workshop Activities Based on Strength Analysis, Cost Consideration, Safety and Health Approach (A Case Study On Workshop Latu Murni – Madiun) ***Puspo Utomo, Linda Herawati, and Ervan Sutanto*** 5. Vibration Response of Jack Wood Material (Artocarpus heterophyllus lamk) as Alternative Wood Bearing to Propeller Shaft of Traditional Fish Boat ***Ikhwansyah Isranuri dan Esra Keliat*** 6. Numerical Solution of Fatigue Stress in Journal Bearing ***Hasan Basri***

THE 5th INTERNATIONAL CONFERENCE ON NUMERICAL ANALYSIS IN ENGINEERING (NAE 2007)

18 - 19 May 2007, Pangeran Beach Hotel, West Sumatera, INDONESIA

Date : March 20, 2007
Ref. No : NAE2007/IC-STAR/0703/10
Attach : - Registration Form
- Guidelines for the preparation of paper

Mr. Hasan Basri
Department of Mechanical Engineering
Sriwijaya University
Kampus Unsri Inderalaya
Jalan Raya Palembang-Prabumulih Km.32
Inderalaya, Ogan Ilir

Re: Invitation to Submit Full Paper

Dear Sir,

We are pleased to acknowledge the receipt of the abstract of your paper entitled **"Numerical Solution of Fatigue Stress in Journal Bearing"** to be presented at the upcoming 5th International Conference on Numerical Analysis in Engineering (NAE2007) to be held at Pangeran Beach Hotel, Padang-West Sumatera from 18-19 May 2007.

You are welcome to prepare print-ready paper(s) and submit to the secretariat. Information on the preparation of manuscripts is enclosed. Please follow the instructions as closely as possible and limit your paper to a maximum of TEN pages, inclusive of figures and tables.

Please note that the full manuscript should reach us by **20 April 2007.** Do not hesitate to contact us if you need any further information.

We look forward to an exciting and successful International Conference and to meeting you in West Sumatera, Indonesia, on May 2007.

Sincerely Yours,
Organizing Committee,
5th Int. Conference on NAE2007

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Chairperson

For Correspondence:

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NUMERICAL SOLUTION OF FATIGUE STRESS IN JOURNAL BEARING

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Abstract

Failure of fatigue is damaged materials where caused frequent load. Fatigue owing to some factors, which is Stress concentration on fatigue, Stress life, Effect size and surface, and Change properties of surface. The fatigue failure of a material is dependent on the interaction of a large stress with a critical flow. In essence, fatigue is controlled by the weakest link of the material, with the probability of a weak link increasing with material volume. This phenomenon is evident in the fatigue test results of a material using specimens of varying diameters. From this research we can get effect of concentration stress on strength fatigue with S-N method. On this method only count fatigue life or endurance limit from Journal bearing housing. By Finite Element Analysis, it is not so easy to determine fatigue life. When we find the first yield point, it means this point is in the highest stress state. Then we can refer S-N curve. In this paper, the effect of bearing and housing elasticity on the stress field, which could result in surface fatigue in journal bearing, has been investigated. This condition is proved with occurred slip lines on surface of specimen. These slip lines are caused on some thousands stress cycles. Additional crack is happened immediately and finally long enough crack. So that formed unstable crack that caused fracture of brittleness or fracture of toughness because section of specimen cannot keep down load.

Keywords: FEM, Fatigue Life, Yield Point, S-N curve, Journal Bearing

1. Introduction

While many parts may work well initially, they often fail in service due to fatigue failure caused by repeated cyclic loading. Characterizing the capability of a material to survive the many cycles a component may experience during its lifetime is the aim of fatigue analysis. In a general sense, Fatigue Analysis has three main methods, Strain Life, Stress Life, and Fracture Mechanics.

According to independent studies carried out by the Battelle group in 1982, between 80-90% of all structural failures occur through a fatigue mechanism and the estimated annual cost of fracture and fatigue to the US was 4.4% of GDP.

Furthermore the Battelle Study concluded that this could be reduced by 29% by application of current fatigue analysis technology.

In the past, fatigue analysis was largely the domain of the development engineer, who used measurements taken from prototype components to predict the fatigue behavior. This gave rise to the traditional “Build it, Test

It, Fix It" approach to fatigue design. This approach is known to be very costly as an iterative design cycle is centered on the construction of real prototype components. This inhibits the ability to develop new concepts and reduces confidence in the final product due to a low statistical sample of tests. It is also common to find early products released with 'known' defects or product release dates being delayed whilst durability issues were addressed.

A more desirable approach is to conduct more testing based on computer simulations. Computational analysis can be performed relatively quickly and much earlier in the design cycle.

Confidence in the product is therefore improved because more usage scenarios can be simulated. It is not recommended, however, that these simulations completely replace prototype testing. It will always remain desirable to have prototype signoff tests to validate the analysis performed and improve our future modeling techniques. However, the number of prototype stages, and hence the total development time, can be reduced.

The following subsections are including: bearing in general, journal bearings, thrust bearing, other types of bearings, rotor-bearing system. Coupled thermomechanical non-linear finite element models have been developed to study 2D and 3D rolling, and rolling plus sliding contact problems. The various less or more realistic material constitutive models have been used to model behavior of bearing materials. The contact stress fatigue is considered as a primary wear mechanism. The damage process under contact loading such as, for example, is the cracking, spaling, and tribological reaction, can be study by the finite element method. We can study the mechanics of the sub-surface or near surface modes of rolling contact failure.

In this paper we overview the physical behavior responsible for fatigue stress from initiation to final component failure of journal bearing.

2. Journal Bearing

The bearings are important enough to be studied because if the shaft's orbit is not stable, or the bearing is not well designed, contact between the shaft and the bearing will appear.

The plain journal bearings are fully used in hydraulics due to their small size, low price, and its capability of carrying load.

The journal bearing appears in the finite element equations as a spring and damper. It appears at one node, linking the shaft to a rigid structure. It has 4 degrees of freedom:

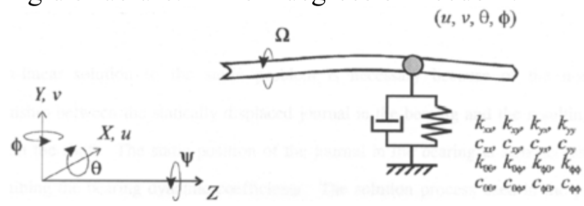


Figure 1. FE Equation of Journal Bearing

The first thing we need to do is determine the static (mean) position of the shaft in the bearing.

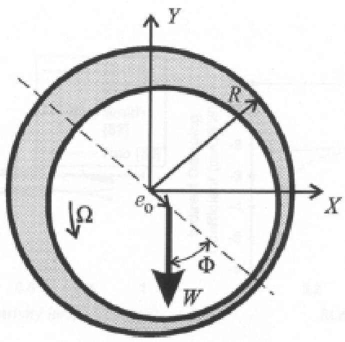


Figure 2. Static Bearing Calculation

Static lateral forces act on the shaft to push it to one side of the bearing.

- The weight of the rotor (in horizontal rotors),
- Generator magnetic forces due to stator offset,
- Static forces,
- Forces due to thrust bearing angular misalignment,
- Forces due to guide bearing misalignments (when there are 3 or more guide bearings).

In static equilibrium the net fluid forces *Fx and Fy* are balanced by the static lateral shaft forces.

Two important parameters are obtained from the bearing test: (i) bearing yield ($\sigma_{b,yield}$) and (ii) bearing ultimate ($\sigma_{b,ult}$) of the material; where bearing stress is defined with the following relation: σ_b=P/(Dt). The yield parameter is defined as the stress at a 2% permanent hole deformation, which is a definition comparable to the tensile yield. Bearing ultimate is defined as the first maximum load peak, which generally was the maximum stress reached.

For the material model, it is assumed that it behaves as an isotropic material with isotropic hardening. Uniaxial tensile test data are simplified into a trilinear behavior, consisting of (i) an elastic part, (ii) plastic part up to necking (15% plastic strain) with stiffness equal to 1467.67 MPa, which is followed by (iii) a description of the necking behavior.

3. The Physics of Fatigue

Fatigue is defined as 'Failure under a repeated or otherwise varying load which never reaches a level sufficient to cause failure in a single application.' Fatigue cracks always develop as a result of cyclic plastic deformation in a localized area. This plastic deformation might arise through the presence of a small crack or pre-existing defect on the surface of a component, for both cases it is practically undetectable and unfeasible to model using traditional Finite Element techniques.

August Wöhler was the first engineer to study fatigue failure and propose an empirical analysis technique. Between 1852 and 1870, Wöhler studied the progressive failure of railway axles. He constructed the test rig shown in Figure 2, which subjected 2 railway axles simultaneously to a rotating bending test. Masses were suspended from the ends of the axles and the axles rotated till failure. Wöhler then plotted the nominal stress vs. the number of rotations to failure on what has become known as the SN diagram. Each curve is still referred to as a Wöhler line. The SN method is still the most widely used today and a typical example of the curve is shown in Figure 3.

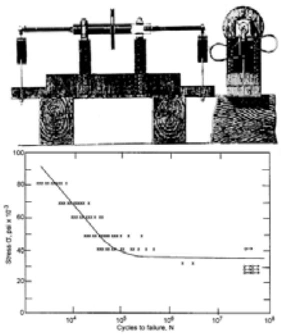


Figure 3. Wöhler's Rotating Bending Fatigue Test

4. Fatigue Life Prediction

The fatigue life prediction follows the strain life approach used for notched geometries. Surface grooves are treated as microscopic notches, where elastic stresses and strains are converted to local plastic stresses and strains in the notch root. Different methods can be used for this conversion, depending on the stress state in the notch root and the applied loading. The most well-known approach is that due to Neuber which relates nominal elastic values to notch root stress and strain as $\varepsilon\sigma = K_t^2 \varepsilon_n^2 E = C$, where C is a constant and en is nominal elastic strain. Plane stress is assumed in this analysis, which can be shown not to be the case for a circumferentially notched bar. A method for general stress states is outlined later.

A fatigue concentration factor has therefore been defined as $k_f = \Delta\sigma(N_f)/\Delta\sigma_{nom}$, where snom is the nominal stress for the notched specimen failing at Nf cycles, and Ds(Nf) is the stress range evaluated from the fatigue life curve. Kf is related to Kt by the notch sensitivity $q = (k_f - 1/K_t - 1)$. This parameter is known to vary with material and notch geometry. Several expressions for this dependence have been proposed. They are all semi-empirical equations which try to account for the material volume influenced by the notch; a sharp notch will have a steep stress gradient into the material, thus the volume of elevated stress will be smaller than for a blunt notch. Neuber and Peterson among others have tried to relate q to the notch root radius using a material parameter. In a previous work, Neuber's expression for Kf was successfully applied in predicting fatigue in the HCF range. For a random surface topography, however, the notch root radius is hard to define. Based on a large amount of empirical data, Siebel and Stieler expressed Kf by the relative stress gradient.

5. Fatigue Strength and Fatigue Life

The fatigue strength of a welded component is defined as the stress range which fluctuating at constant amplitude causes failure of the component after a specified number of cycles (N). The stress range is the difference between the maximum and minimum points in the cycle. The number of cycles to failure is known as the endurance or fatigue life.

6. S-N Curve

The expression linking N and □□$_R^m$ can be plotted on a logarithmic scale as a straight lineand is referred to as an S-N curve. The relationship holds for a wide range of endurance. It is limited at the low endurance end by static failure when the ultimate material strength is exceeded. At endurances exceeding about 5-10 million cycles the stress ranges are generally too small to permit propagation under constant amplitude loading. This limit is called the non-propagating stress. Below this stress range cracks will not grow.

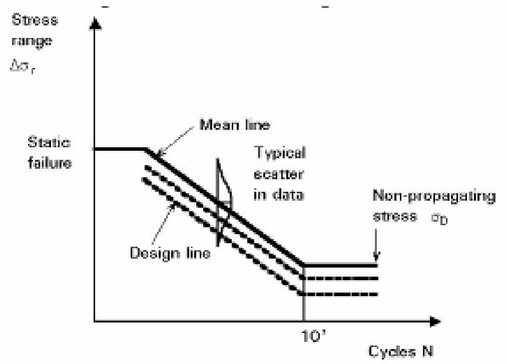


Figure 4. Typical S-N curve for constant amplitude test

For design purposes it is usual to use design S-N curves which give fatigue strengths about 25% below the mean failure values, are used to define these lines.

7. Effect of Mean Stress

In non-welded details the endurance is reduced as the mean stress becomes more tensile. In welded details the endurance is not usually reduced in those circumstances. This behavior occurs because the weld shrinkage stresses (or residual stresses), which are locked into the weld regions at fabrication, often attain tensile yield. The crack cannot distinguish between applied and residual stress. Thus, for the purposes of design, the S-N curve always assume the worst, i.e. that the maximum stress in the cycle is at yield point in tension. It is particularly important to appreciate this point as it means that fatigue cracks can grow in parts of members which are nominally 'in compression'.

8. Effect of Mechanical Strength

The rate of crack growth is not significantly affected by variations in proof stress or ultimate tensile strength within the range of low alloy steels used for general structural purposes. These properties only affect the initiation period, which, being negligible in welds, results in little influence on fatigue life. This behavior contrasts with the fatigue of non-welded details where increased mechanical strength generally results in improved fatigue strength, as shown in Figure 5.

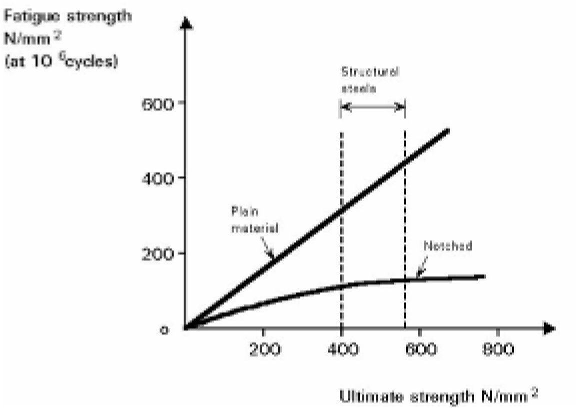


Figure 5. Effect of Mechanical Strength

9. FEM Analysis

Finite element technique involves element-modeling discretion, which is defined through a displacement function of each node.

$$\{F\} = [k]\{D\} \tag{1}$$

Modeling used is rectangular trilinear element. As the result, when the load is occurred to the journal bearing, then the out coming strain and stiffness matrix are,

$$\{\phi^{(e)}\} = [N_1 N_2 \ldots N_n] \{\Phi^{(e)}\}$$
$$[K^{(e)}] = \sum_{i=1}^{N} [k] \tag{2}$$

Next, we can determine the possible outcoming stress by,

$$\{\tau^{(e)}\} = [B] \{\Phi^{(e)}\} \tag{3}$$

Finite element analysis is supported by *FAST* Software and structure analysis. The boundary conditions of Journal bearing, give it a fatigue load of 0 to 30000 N and ultimate axial load of 40000 N. This is a simple case. However, we will use a 3D FEA model to make a point. After a linear-static stress analysis, result shows the maximum stres. Because it is a linear analysis, we can easly scale the stresses for different loads. There for, the maximum stress is:

$$\sigma_{\max} = \frac{F_{tu}}{F_{fatigue}} (PeakStress) \tag{4}$$

The margin of safety at the ultimate load is base on maximum stress value from Finite Element Analysis.

$$M_{safetyUlt} = (S_{tu} / \sigma_{\max}) - 1 \tag{5}$$

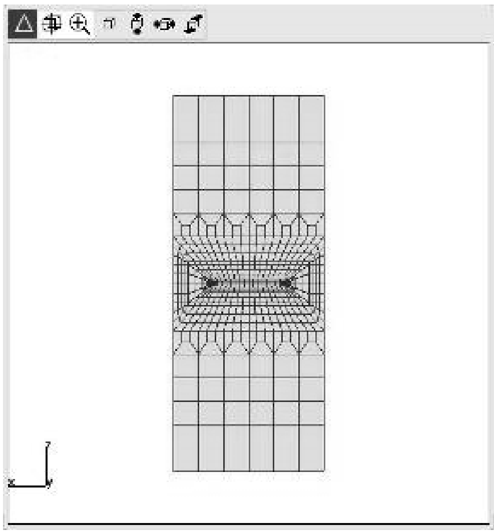


Figure 6. FEA Modelling for Journal Bearing housing by *Fast* Soft.

Handling Finite Element Analysis stress requires a good understanding of the stress-concentration effect, quantified as a factor K_t. The theoritical stress-concentration factor is based on a theoritical elastic, homogeneous, isotropic material and can be expressed as:

$$k_t = \sigma_{\max} / \sigma_{nom} \tag{6}$$

Where:
s_{max} = Maximum (peak) stress, and s_{nom} = nominal or average stress.
Handling FEA fatigue stresses correctly also requires good understanding of fatigue stress-concentration factor, *Kf*. It's found from.

$$K_f = S_{nf}/S_f \qquad (7)$$

where S_{nf} = fatigue stress at K_t = n, and S_f = fatigue stress at k_t = 1.
The relation between the *fatigue* stress-concentration factor and the stress-concentration factor is,

$$K_f = q(K_t - 1) + 1 \text{ or}$$
$$q = \frac{(K_f - 1)}{(K_t - 1)} \qquad (8)$$

where *q* is the fatigue-notch sensitivity and $0 \le q \le 1$. Here, *q* = 0 for no notch and *q* = 1 for a full notch. Average fatigue-notch-sensitivity values for some typical materials can be found in Figure 1.31 in *Peterson's Stress Concentration Factors*.

The relationship between *Kf* and *Kt* shows that *q* plays the important roll in the fatigue-stress-concentration factor. It should be obvious that $Kf \le Kt$. When *q* is not available, conservative results come from using *Kf* = *Kt* or *q* = 1. S-N curves with *Kt* = 1 are typically applied to FEA results. By knowing the *q* effect, it can be shown that S-N curves with *Kt* = 1 still produce conservative fatigue calculations for FEA applications because it assumes *q* = 1 or *Kf* = *Kt*. That's why the surface factor is usually ignored in FEA for average or machined surfaces.

By Finite Element Analysis, it is not so easy to determine fatigue life. When we find the first yield point, it means this point is in the highest stress state. Then we can refer S-N curve.

10. Results and Discussion

From this research we can get effect of concentration stress or Kt on strength fatigue with S-N method. On this method only count fatigue life or endurance limit from Journal Bearing.

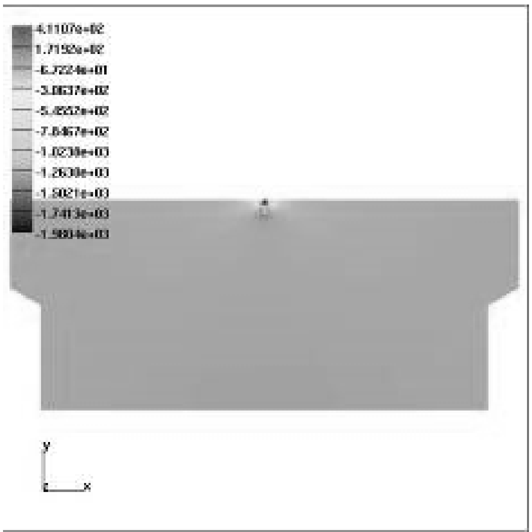
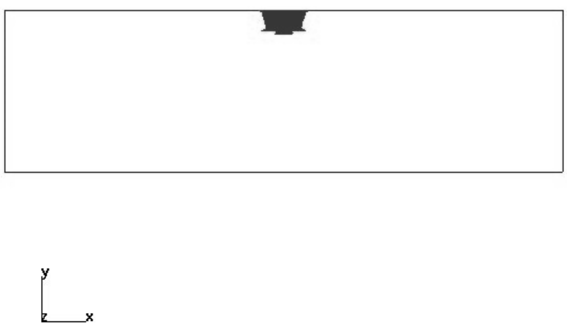


Figure 7. Picture shows the stresses distribution



Figutre 8. The first yield point occurs

Table 1. Fatigue Test for Journal Bearing

Stress (Mpa)	Fatigue life (cycle)
13,872	48671
21,047	20518
27,728	15191
35,312	10513
40,128	5732

From the data we can draw S-N curve for Journal Bearing.

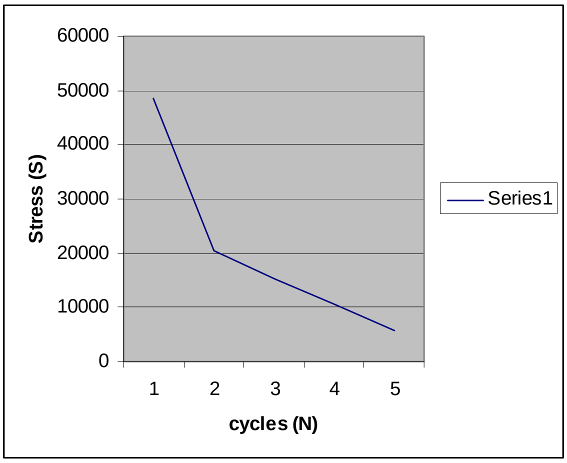


Figure 9. S-N Curve

11. Conclusions

A general approach to modelling the durability of Journal Bearing has been developed.

The approach removes the requirement of rebuilding FEM models in order to capture the important stress raising features which significantly affect fatigue life predictions.

The method is ideally suited for predicting data for fatigue life calculations in Journal Bearing.

Example applications have been presented demonstrating some of the capabilities of the method.

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