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VIBRATION ENGINEERING AND TECHNOLOGY OF MACHINERY

Proceedings of The Fifth International Conference on VETOMAC-V

27-28 , August, 2009, Wuhan, P.R. CHINA.

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PREFACE

International Conferences on Vibration Engineering and Technology of Machinery (*VETOMAC*) are a series of international conferences being held as an effort to bring together engineers, scientists and mathematicians from academia and industry and to provide a platform for the presentation of creative and novel research and technological findings. It was initiated by Professor J.S. Rao, President of The Vibration Institute of India and Chief Editor of *Advances in Vibration Engineering*, an international journal aimed at providing most recent information for researchers, faculty, students and practicing engineering in these specializations on a continued basis with contributions from all over the world.

There have been four *VETOMAC* conferences successfully organized previously: *VETOMAC I* was organized by the Indian Institute of Science, Bangalore in 2000; *VETOMAC II* was organized by the Bhabha Atomic Research Centre, Mumbai in 2002; *VETOMAC III* was organized by the Indian Institute of Technology, Kanpur in 2004; and *VETOMAC IV* was jointly organized by the University College of Engineering, Osmania University, India, R&D Division Bharat Heavy Electricals Limited, India and The Vibration Institute of India, Hyderabad, India in 2007. As an effort to bring *VETOMAC* conferences international, Huazhong University of Science and Technology (HUST) and City University of Hong Kong (CityU) have been assigned the duty of organizing *VETOMAC V* to be held at HUST, Wuhan from 27-28 August 2009. The main theme of *VETOMAC V* is to provide directed focus on the state-of-the-art theoretical, experimental and computational methodologies in vibration engineering and technology in mainland China in particular and internationally in general. The associated discussions will faciliate exchanges of ideas and provide guidance for future research directions and vibration engineering solutions.

We would like to record our appreciation of the effort and assistance rendered upon by many parties without which *VETOMAV V* will be impossible, in particular to The Indian Institute of Technology (TVII), The Chinese Society of Theoretical and Applied Mechanics (CSTAM), ALTAIR China, the many colleagues and students of HUST, and last but not least, your earnest and sincere participation.

We wish the success of *VETOMAC V* and hope all participants a pleasant stay in Wuhan, P.R. China.

Professor Zeng He, Huazhong University of Science and Technology
Ir Dr C.W. Lim, City University of Hong Kong

August 2009

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CONTENETS

The Fifth International Conference on VIBRATION ENGINEERING AND TECHNOLOGY OF MACHINERY
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FATIGUE LIFE ESTIMATION OF TURBOMACHINERY BLADES

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ABSTRACT After a brief review of development of rotors and the need to perform life estimation of the modern machinery, this paper presents a comprehensive method of life estimation of advanced high speed rotating machinery concerning stress based, strain based and linear elastic fracture mechanics methods. A novel feature presented here includes a complete analytical procedure to determine a nonlinear damping model for material and friction effects without depending on costly experimental procedures. Using this nonlinear damping model, a method to determine resonant stress and stress distribution around the critical is also illustrated. The effect of acceleration on the blade resonant stress is included in the cumulative damage while crossing a critical speed. To help the designers speed up the simulation and assess life, a process simulation of all the above effects are integrated into one single platform and the way it works is illustrated.

I. UNDERSTANDING OF FATIGUE IN 19TH CENTURY

With the ushering of Industrial Revolution in the second half of 18th century, thanks to James Watt, see Dickinson and Jenkins (1989) and his reciprocating steam engine, it became the main work horse replacing the good old horse drawn mining pumps and wagons. Though sudden fatigue failures of axles in horse drawn wagons was known through occasional incidents, the rail road axles fatigue failures became a problem. Albert Wilhelm (1838) is attributed to be the first person to record observations of metal fatigue. While working in the Mining and Forestry Office in Clausthal, Germany, in 1829, he observed, studied and reported the failure of iron mine-hoist chains arising from repeated small loadings, the first recorded account of metal fatigue, see Stephens (2001).

Jean-Victor Poncelet, a French military engineer and later professor at the École d'Application in Metz published his monograph on *Introduction à la mécanique industrielle* in 1829. He described in his classes around 1837-1839 for the first time that metals as being tired in his lectures at the military school at Metz. He introduced the notion of fatigue of metals characterized by the drop in durability of steel products under repeated variable loads; see Tóth and Yarema (2006).

W. J. M. Rankine (1842) was one of the first engineers to recognize that fatigue failures of railway axles were caused by the initiation and growth of brittle cracks. In the early 1840s he examined many broken axles, particularly after the Versailles train crash in 1842 when a locomotive axle suddenly fractured. He showed that the axles had failed by progressive growth of a brittle crack from a shoulder or other stress concentration source on the shaft, such as a keyway. He was supported by similar direct analysis of failed axles by Joseph Glynn (1844), where the axles failed by slow growth of a brittle crack in a process now known as metal fatigue.

British Rail experienced series of fatigue failures of the railway axles; therefore, The Railway Inspectorate was formed in 1840 to investigate the accidents. The report was made in 1848. Their first investigation concerned the derailment of a train caused by the fall of a large casting from a wagon on a passenger train. The Howden rail crash on 7 August 1840 killed four passengers. After submission of the report in 1848, Eaton Hodgkinson in 1849 qas granted a small sum of money to report to the UK Parliament on his work in ascertaining by direct experiment, the effects of continued changes of load upon iron structures and to what extent they could be loaded without danger to their ultimate security. This is the first attempt to understand fatigue phenomenon in a scientific manner. Braithwaite (1854) reported on common service fatigue failures and coined the term *fatigue*.

Systematic fatigue testing was undertaken in 1860 by William Fairbairn and August Wöhler. Fairbairn (1864) built large-scale testing apparatus for the studies, partly funded by the Board of Trade. He studied the effects of repeated loading of wrought and cast iron girders, showing that fracture could occur by crack growth from incipient defects, a problem now known as fatigue.

Wöhler (1858 to 1870) summarized his work in several papers on railroad axles; see also Wöhler (1867). He concludes that cyclic stress range is more important than peak stress and introduces the concept of endurance limit. His work on fatigue marks the first systematic investigation of *S-N* Curves, also known as Wöhler curves, to characterize the fatigue behavior of materials. He showed clearly that fatigue occurs by crack growth from surface defects until the product can no longer support the applied load. The history of a fracture can be understood from a study of the fracture surface. He developed apparatus for repeated loading of railway axles, mainly because many accidents were caused by sudden fatigue fracture. The presentation of his work at the Paris Exposition in 1867 brought world wide attention.

Towards the end of 19th century, it has been realized that reciprocating steam engine causes lot of vibrations and fatigue failures and people began to look at pure rotating machines in the hope that there will be no vibration, noise and fatigue. Within hundred years of existence, the reciprocating steam engine was challenged.

During second century BC, Hero, see Pederson (1993) demonstrated the principle of a reaction turbine, but couldn't realize any useful work. Despite the scientific revolution followed by industrial revolution, James Watt tried to build a steam turbine and came to conclusion that it could not be built given the state of contemporary technology.

Nearly hundred years after Watt built his steam engine, De Laval of Stockholm succeeded in building the first steam turbine (impulse turbine), see Smil (2005). Once a rotating machine was achieved with steam as motive force, there was a tremendous expansion in the capacity of power generation. Just one year after Laval's turbine, Charles Parsons in 1884 came up with the first reaction turbine, see Parsons (1911). Because of heavy vibrations in reciprocating machines with severe torque and speed fluctuations, these turbines are hailed "vibration free engines" for that time at least.

With the invention of Dynamo in 1878 by Thomas Alva Edison and installation of Pearl Street Electric Power Station in 1882, the path has been cleared to produce electricity in an unprecedented scale which brought in a phenomenal expansion of the steam turbine; Early part of 20th century has seen 2 MW turbines, by 1920 the first 50 MW machine was made and by end of the II World War, 100 MW machines began to produce power. The capacity rose to 1000 MW by 1970 and in 1980 a single machine produced 1500 MW electricity, see *A Century of Progress* (1981).

II. DEVELOPMENTS IN 20TH CENTURY

Sir Frank Whittle (1907—1996) was a Royal Air Force Officer, proposed in a thesis that planes would need to fly at high altitudes, where air resistance is much lower, in order to achieve long ranges and high speeds, see St. Peter (1999). Piston engines and propellers were unsuitable for this purpose, so he concluded that rocket propulsion or gas turbines driving propellers would be required: jet propulsion was not in his thinking at this stage. In 1929, Whittle had considered using a fan enclosed in the fuselage to generate a fast flow of air to propel a plane at high altitude. A piston engine would use too much fuel, so he thought of using a gas turbine and patented his idea.

Whittle began constructing a test engine in July 1936, but it proved inconclusive. Whittle concluded that a complete rebuild was required. By April 1941 the engine W.2, was ready for tests and it produced 1600 lb thrust. The first flight Gloster E.28/39 took place on 15 May 1941. General Electric worked quickly and their XP-59A Aircomet was airborne in October 1942. In just six decades later, General Electric GE-90 115-B engine is designed for a thrust rating of 115,000 pounds (511 kN).

The rapid developments of rotating machinery, steam or gas turbines in power generation, centrifugal compressors and pumps in oil and gas industry and aerospace applications and finally aircraft engines, particularly defense applications have pushed the technology beyond recognition during the later half of 20th century.

Whittle faced several fatigue failure problems in the development of his W.2 engine, in his own words, *Frequency of turbine blade failures was becoming the latest technological barrier to overcome*, see St. Peter (1999). QE2 9th Stage Starboard HP Turbine Rotor experienced blade fatigue failures on 24 December 1968, in its maiden voyage, see Fleeting and Coats (1970). Bladed Disks are most flexible elements in high speed rotating machinery. Due to rotation, the blade root gets tightened in the disk slot and transmits the centrifugal load. The mating contact surfaces could be just 2 for high pressure turbine blades a few centimeters long and may increase to six or more for low pressure 1m long turbine blades. While the average stress in the mating areas is fully elastic and well below yield, the peak stress at singularities in the groove shape can reach yield values and into local plastic region. Last stage LP turbine blades are the most severely stressed blades in the system. Usually these are the limiting cases of blade design allowing the peak stresses to reach yield or just above yield conditions.

Failures can occur with crack initiation at the stress raiser location and propagation, two cases can be cited. The last stage blades in an Electricite de France B2 TG Set failed in Porcheville on August 22, 1977 during over speed testing, see Frank (1982). On March 31, 1993 Narora machine LP last stage blades suffered catastrophic failures, see Rao (1998). The left (No.1) engine of Boeing 777-300 A6-EMM failed because of blade fatigue on 31 January 2001 at Melbourne Airport. American Airlines Boeing 767 doing a high power GE CF6 engine run at Los Angeles airport had a #1 engine HPT failure on June 2, 2006. HPT let go and punctured left wing, #2 engine, peppered fuselage and set fire to the aircraft. The turbine disk exits the engine and slices through the aircraft belly and lodges in the outboard side of the #2 engine. Thus turbine blade failures continue to occur despite best design practices.

With the advent of high speed computers, the classical methods of stress and vibration analyses have given way to finite element methods that enable static and dynamic stress calculations at each and every discontinuity in the structure very accurately. These stresses are employed in assessing fatigue life and as per the current trend, simulation methods are advanced to speed up the life estimation and bring the products to market in minimum time.

III. WHAT IS INVOLVED IN LIFE ESTIMATION?

Historically, lifing calculations addressed what we call today High Cycle Fatigue, usually for more than 10^7 cycles, at least 10^4 cycles of loading. Wöhler *S-N* curves of 19th century still form basis for HCF calculations.

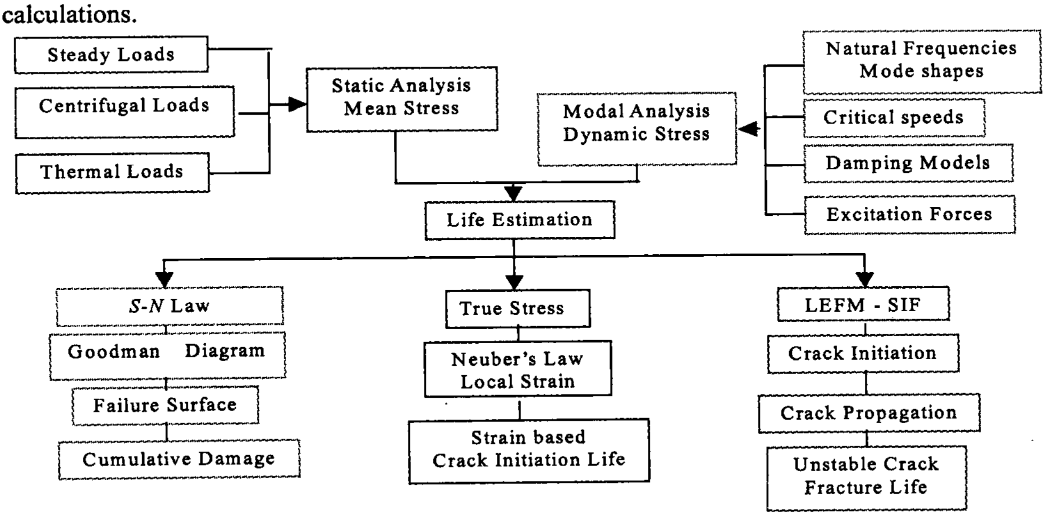

Fig. 1 Outline for Life Estimation

3.1 Mean Stress Field

Blades are subjected to steady stress fields due to gas loads, thermal loads, and centrifugal loads under normal conditions of operation. The gas loads are determined from CFD analysis of the gas path, which have a steady part and an unsteady part at nozzle passing frequencies. Because of compression in the compressor flow path or the hot gas path in turbine blades, they are subjected to thermal loading during the transient period of start-up and shut-down, they will form a mean load at a given steady operation or an overall cyclic load for each start-up and shut-down operation. The blades are also subjected to mean loads due to centrifugal loading which could be substantial in low pressure compressor and turbine blades that will push the structure into globally elastic and locally plastic conditions. Cyclic symmetry can be utilized in assessing these stress fields. The steady stress field determination is well established and can be directly imported by the user at the start of life estimation to a recently developed tool, *Altair TurboManager*, Rao, Ratnakar, Suresh and Rangarajan (2009).

Of particular interest is the result at the stress raiser location. Usually, the centrifugal loads lead to local plastic conditions in the root; an elasto-plastic analysis can give a near true picture in the stress raiser location.

3.2 Modal Analysis and Dynamic Stresses

Determination of natural frequencies and mode shapes as a function of speed is well known. The results are imported into *TurboManager* which plots the Campbell diagram.

The Campbell diagram identifies critical speeds the resonances (natural frequencies crossing with excitation frequencies) through which the blade passes.

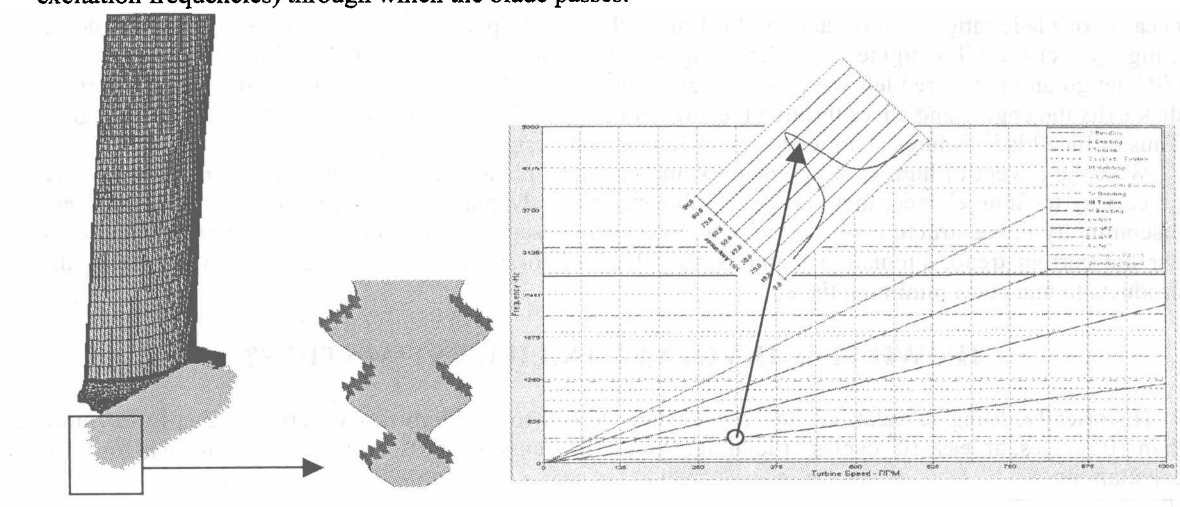

Fig. 2 FE Model of a Blade and its Campbell Diagram showing its I Critical Speed

The resonant stress at critical speed depends on two factors.

1. Excitation strength

The excitation strength is determined from a transient CFD analysis. FFT of the time domain signal identifies the resonant excitation. We will not discuss this topic here excepting to give two useful references that describe the basic flow path interference between the stator and rotor stages, see Rao (1994), Rao and Saravana Kumar (2008).

2. Damping

Damping has been identified long ago as a key parameter in blade design. Rowett (1914) conducted tests on elastic hysteresis in steel. Effects of friction and loose mounting were studied by Hansen et al. (1953). Sinha and Griffin (1984) studied analytically the effects of static friction on the forced response of frictionally damped turbine blades. Usually simple viscous damping model is used with equivalent damping determined from a test. Such a linear model is inadequate since material damping is highly dependent of the state of stress in the blade.

A nonlinear damping model was quantified through experiments by Rao, Vyas and Gupta (1986); the equivalent viscous damping is expressed as a function of strain amplitude at a reference point in a given mode of vibration at a given speed of rotation. Rao and Saldanha (2003) developed an analytical procedure using Lazan's hysteresis law (1968) for damping energy $D = J(\sigma/\sigma_e)^n$, where J and n are the coefficient and exponent and σ_e is the endurance limit. Briefly the steps followed are:

The total damping energy D_0 (N.m) is given by

$$D_0 = \int_0^v D\mathrm{d}v \tag{1}$$

where v is the volume. The Loss factor η is

$$\eta = \frac{D_0}{2\pi W_0} \tag{2}$$

where W_0 is the total strain energy (N.m). Then, Equivalent Viscous Damping C (N-s/m) is

$$C = \frac{\eta K}{\omega} \tag{3}$$

where ω is the natural frequency (rad/s) and K is the modal stiffness (N/m).

For increased strain amplitudes, the orthonormal reference strain amplitudes, stress and strain energy are multiplied by a factor F to obtain the equivalent viscous damping C_e at various strain amplitudes as given below.

$$\left.\begin{aligned} &\varepsilon' = \varepsilon F \\ &W_0' = W_0 \cdot F^2 \\ &\eta' = \frac{D_0'}{2\pi W_0'} \\ &C_e' = \frac{\eta' K}{\omega_n} F^2 \\ &\xi = \frac{C_e}{2\sqrt{Km}} F^2 \end{aligned}\right\} \tag{4}$$

A plot of equivalent viscous damping ratio as a function reference strain amplitude in the chosen mode of vibration defines the nonlinear damping model. Orthonormal modal stress field for each critical speed is imported from a parent solver and a nonlinear modal damping model for each critical speed as a function of reference strain value is developed.

The advantage of this procedure is it determines damping analytically avoiding expensive and time consuming tests at the design stage. *TurboManager* determines this nonlinear damping model and uses it in obtaining resonant stress.

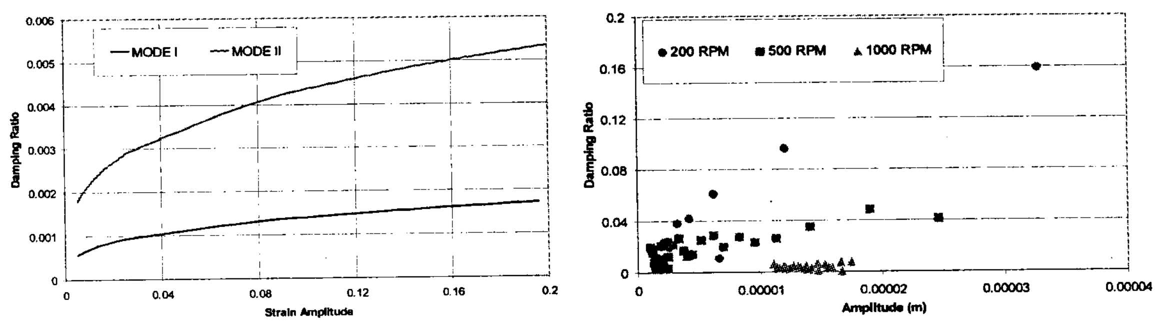

Fig. 3 Material and Combined Material and Friction Damping

Rao, Narayan, Ranjith, and Rejin (2008), determined the material damping and combined material friction damping of a bladed disk as a function of reference strain amplitude in a given mode of vibration at an operating speed as shown in Fig. 3. An alternate method is to compare the material damping and friction damping separately and retain which ever contributes maximum value as shown in Fig. 4.

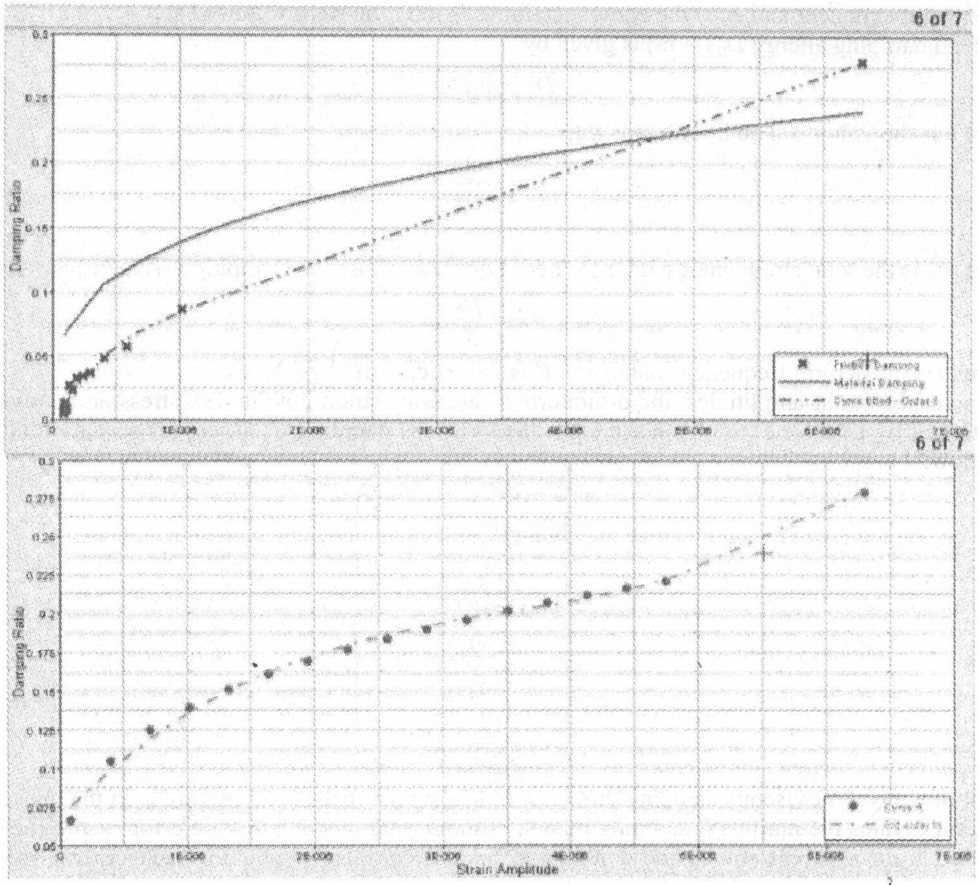

Fig. 4 Material and Friction Dampings and the resultant

3.3 Determination of Stress Distribution around Resonance

Now, the question is how to determine the stress rise and fall at critical speed and the resonant stress magnitude. The best approach is to assume the unsteady force field to be steady and obtain the equivalent static stress field first. This equivalent static stress distribution is then multiplied by the quality factor $1/2\xi$, see Rao and Gupta (1984) to obtain correct resonant stress; the only catch is we don't have the damping value. The general practice so far is to assume this damping value or determine the average damping value from an experiment. Here we have a nonlinear damping model as described in 3.2.

A nonlinear approach is used to determine the correct value of ξ by an iteration process to determine the resonant stress at all the critical speeds, see Rao, Pathak and Chawla (2001), Vyas and Rao (1994) and Rao and Vyas (1994).

The approach to solve the nonlinear damping problem is outlined below:

- Assume an equivalent linear viscous damping initially $\xi(0)$.
- Quality factor = $Q(0) = 1/[2\xi(0)]$.
- For this damping value ξ (0) we simply read of the strain amplitude at reference node from the analytically evaluated damping relation.
- Let this strain amplitude be $\varepsilon(0)$.

- Let the reference nodal strain with alternating load treated as steady be ε_s.
- Then, the resonant strain amplitude ε_r is ε_s times Q (0).
- Compare this result ε_r with assumed strain at the reference node ε (0).
- Obviously, they will be different and we determine in the second iteration a strain amplitude obtained by averaging the starting value and first iterated strain value is used to repeat the process until convergence is achieved.

We can state the i^{th} iteration steps as below in equation (5).

- Damping value $\xi^{(i)}$
- Quality Factor $Q^{(i)} = 1/[2\xi^{(i)}]$
- Reference resonant strain $\varepsilon_r^{(i)} = \varepsilon_s\, Q^{(i)}$
- New Damping Value for $\varepsilon_r^{(i)}$ $\xi^{(i+1)}$
- Compare $\xi^{(i+1)}$ with $\xi^{(i)}$ and check for convergence.

If convergence has not been achieved, take average damping value

$$\tfrac{1}{2}\,\{\xi^{(i)}+\xi^{(i+1)}\} \tag{5}$$

and repeat the above steps.

This procedure is implemented in *TurboManager* so that the designer can determine the dynamic stress accurately for a given mode experiencing resonance at a critical speed.

The stress response in this resonance region is then obtained by the dynamic magnifier relation $H(\omega)$ (with $r = \dfrac{\omega}{\omega_n}$).

$$H(\omega) = \frac{1}{\sqrt{(1-r^2)^2 + (2\xi r)^2}} \tag{6}$$

It may be noted here that the phase angle of the excitation force may vary from point to point on the blade surface and this phase should be accounted in the stress estimation, see Rao, Peraiah and Uday Kumar Singh (2007). In those cases it may be easy to perform a forced vibration analysis with a frequency very close to zero, e.g., 0.01 rad/s. The result will be same as the equivalent static stress field.

IV. HIGH CYCLE FATIGUE (HCF) LIFE ESTIMATION

In Fig.1 the life estimation is divided into three separate modules. All these three modules need mean and dynamic stress fields and the location where peak values occur.

Historically it was HCF that was first developed as the failures at that time occurred after sufficiently long hours of operation lasting months or years of rail road axles. That's how Wöhler (1867) developed *S-N* curves.

4.1 *S-N* Curves

The simplest form of an *S-N* curve is expressed in a log-log plot that defines *Fatigue Strength* or *Endurance Limit* of the material. Rao, Pathak and Chawla (2001) discussed other definitions. TurboManager determines the *S-N* curve first from given material properties as indicated in Fig. 1.

Fatigue testing is done under controlled laboratory conditions; the actual condition of a mechanical component is far from the ideal conditions, e.g., surface finish, specimen size effect, stress concentration, temperature conditions. We also have to account for the reliability as the fatigue tests are statistical in nature besides any special effects such as corrosion. These factors are discussed in Barsom and Rolfe (1987) and Rao (2000). *TurboManager* updates the material *S-N* curve to the component curve.

4.2 Effect of Mean Stress – Goodman Diagram

While *S-N* curve deals with purely alternate load, almost all structures are subjected to mean working load and stress above which the alternating load is applied. The mean stress has significant influence on fatigue and *S-N* curves. Goodman (1899) suggested a linear relationship using Endurance limit and Ultimate tensile strength of the material. A parabolic relation was suggested by Gerber (1874) which predicts no significant effect when the mean stress is very small. Soderberg (1930, 1933) suggested a linear relationship again but used yield value rather than ultimate tensile strength. Marin (1962) proposed an elliptical relation. Kececioglu, Chester, and Dodge (1974) used an exponent 2.6 instead of 2 for mean