iJRASET
International Journal For Research in
Applied Science and Engineering Technology

INTERNATIONAL JOURNAL FOR RESEARCH

IN APPLIED SCIENCE & ENGINEERING TECHNOLOGY

Volume: 7 **Issue:** III **Month of publication:** March 2019

DOI: http://doi.org/10.22214/ijraset.2019.3020

International Journal for Research in Applied Science & Engineering Technology (IJRASET)
ISSN: 2321-9653; IC Value: 45.98; SJ Impact Factor: 6.887
Volume 7 Issue III, Mar 2019- Available at www.ijraset.com

A Review on Vibration Theory and Friction Damping of Gas Turbine Blade Friction

Shailendra Kumar Bohidar
PhD Research Scholar, Kalinga University, Raipur INDIA

Abstract: This paper reviews recent research on analyzing friction dampers to reduce turbine blade vibration. Methods of analysis and attempts to indicate some of the limitations and theory of Gas turbine and considerations in gas turbine modeling. In order to reduce the high fatigue risk of blade wheel cycles, the prediction of vibration levels is at an early stage of the design process important. Therefore, the different sources of damping must be modeled accurately. Conclusions are drawn concerning the current state-of-the-art and recommendations are made regarding directions for future research and development.

I. INTRODUCTION

A. Basics Gas Turbines

Gas turbines work dependent on Brayton cycle. Figure 1 demonstrates a commonplace single-shaft gas turbine and its principle segments including blower, burning chamber (combustor), and turbine. The arrangement of these segments is called motor center or gas generator (GG). Blower and turbine are associated by the focal shaft and pivot together. Figure 2 shows standard Brayton cycle in weight volume (P-V) and temperature-entropy (T-outlines individually. Air enters the blower at segment 1 and is compacted through passing the blower. The hot and compacted air enters the burning chamber (combustor) at area 2.

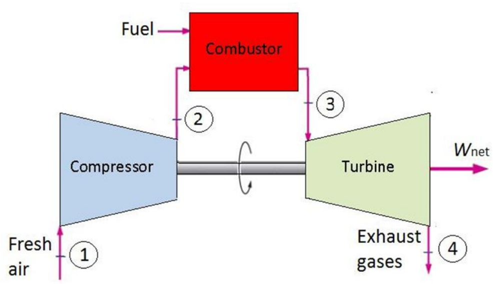


Fig. 1: Schematic of a typical single-shaft gas turbine.

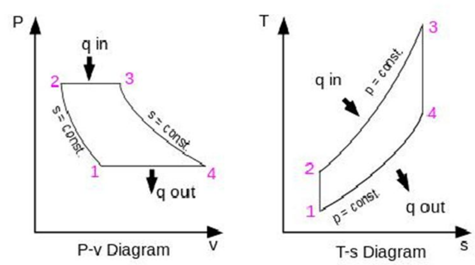


Fig. 2: Typical Brayton cycle in pressure-volume and temperature-entropy frames.

B. Gas Turbine Type

As the initial step of demonstrating, it is important to get enough data about the kind of gas turbine which is to be displayed. As it was at that point expressed, GT can be an air or stationary gas turbine. In spite of the fact that there are distinctive kinds of GT dependent on their applications in industry, they have a similar principle regular parts including blower, ignition chamber and turbine. Figure 3 demonstrates an ordinary single-spool air gas turbine motor.

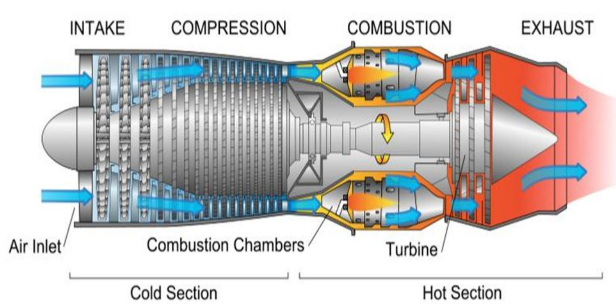


Fig. 3: A typical single-spool turbojet engine.

C. Gas Turbine Configuration

Setup of a gas turbine is another essential standard in GT displaying. Albeit all gas turbines almost have a similar fundamental structure and thermodynamic cycle, there are extensive qualifications when they are researched in subtleties. For example, to upgrade gas turbine cycle, framework proficiency or yield control, through various strategies, for example, warming, between cooling or warmth trade, specific GT designs are used. Gas turbines can be likewise ordered dependent on the kind of their poles. They might be single-shaft or split-shaft (twin-shaft, triple-shaft). In a solitary shaft gas turbine, a similar turbine rotor which drives the blower is associated with the power yield shaft through a speed decrease. In a split-shaft gas turbine, the gas generator turbine and the power turbine (PT) are mechanically separated. Gas generator turbine, additionally called blower turbine (CT) or high weight (HP) turbine, is the segment which gives expected capacity to driving the blower and embellishments. Be that as it may, control turbine, additionally called low weight (LP) turbine, does the usable work. Figure 4 demonstrates an average twin-shaft gas turbine motor

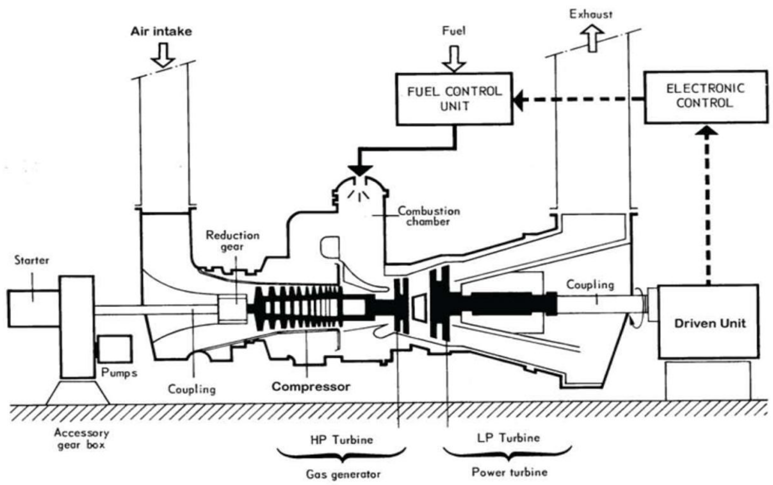


Figure.4: A typical twin-shaft gas turbine engine .

II. THEORY OF VIBRATION

Any motion, which repeats itself after an interval of time period, is called vibration. The study of vibration is concerned with the oscillatory motions of bodies and forces associated with them. All bodies possessing mass and elasticity are capable of vibration. Most of the Engineering machines and structures experience vibration to some degree and their design generally requires consideration of their oscillatory behavior. Due to the faulty design and manufacturing defects, the high-speed engines are subjected to vibrations. The vibrations lead to excessive and undesired stresses in the rotating system,

A. Types of Vibrations

1) *Free Vibration:* Free vibration is a vibration in which energy is neither added to nor removed from the vibrating system. It will just keep vibrating forever at the same amplitude. Except from some superconducting electronic oscillators, or possibly the motion of an electron in its orbit about an atomic nucleus, there are no free vibrations in nature. They are all damped to some extent. The equation of free vibration is as

$$m\ddot{x} + Kx = 0$$

2) *Free-Damped Vibration:* Damped vibration is one in which there is an energy loss from the vibrating system. This loss may be in the form of mechanical friction or in the form of turbulence as the vibrating system disturbs its surroundings. The amplitude of a free-damped vibration will eventually decay to zero. The diminishing of vibrations with the time is called damping. External damping can be increased by using dashpots or dampers.

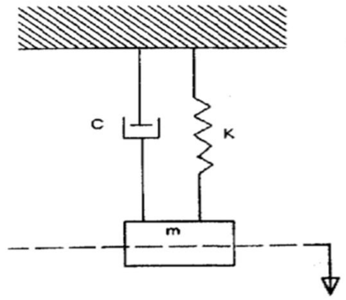


Fig 5 Free-damped vibration system

Consider the forces on the mass m when it is displaced through a distance below the equilibrium position during the vibratory motion.
Let,

K= stiffness of the spring.
ç= damping coefficient (damping force per unit velocity)
0, = frequency of natural undamped vibrations
X =displacement of mass from mean position at time t.
u = r= velocity of the mass at time t
f= x acceleration of the mass at time t.

When the mass moves downwards, the friction force of the dashpot acts in the upward direction as shown in Fig. 5.

3) *Forced-Undamped Vibration:* Forced vibration is one in which energy is added to the vibrating system, as for example in a clockwork mechanism where the energy stored in a spring is transferred a bit at a time to the vibrating element. The amplitude of a forced, undamped vibration would increase over the time until the mechanism was destroyed. Thus the equation of motion will be

$$m\ddot{x} + Kx = F_0 \sin \omega t$$

4) *Forced-Damped Vibration:* The amplitude of a forced, damped vibration will settle to some value where the energy loss per cycle is exactly balanced by the energy gained.

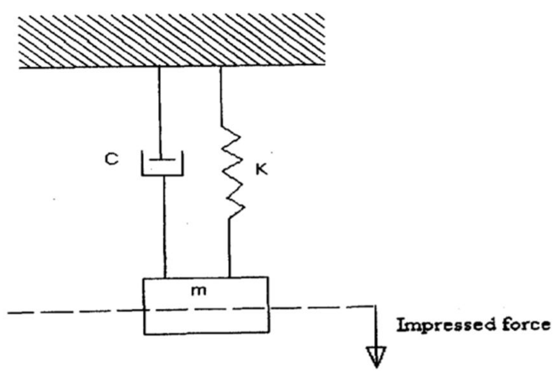


Fig. 6 Forced-damped vibration system

Thus the equation of motion will be

$$m\ddot{x} + c\dot{x} + Kx = F_0 \sin \omega t$$

Complete solution of this equation consists of two parts, the complimentary function (CF) and the particular integral (PI)

$$\text{CF}= Xe^{-\zeta\omega_n t}\sin(\omega_d t+\varphi_1)$$

$$PI = \frac{F_0}{\sqrt{(K-m\omega^2)^2+(c\omega)^2}}\sin(\omega t-\varphi)$$

$$x = CF + PI$$

$$= Xe^{-\zeta\omega_n t}\sin(\omega_d t+\varphi_1)+\frac{F_0}{\sqrt{(K-m\omega^2)^2+(c\omega)^2}}\sin(\omega t-\varphi)$$

The first part, complimentary function becomes negligible with time as €* = 0. Then the steady state response of the system is the second part of the solution (particular integral).The amplitude of the steady state response is

$$A = \frac{F_0}{\sqrt{(K-m\omega^2)^2+(c\omega)^2}}$$

$$= \frac{F_0/K}{\sqrt{\left(1-\frac{m\omega^2}{K}\right)^2+\left(\frac{c}{K}\omega\right)^2}}$$

$$= \frac{F_0/K}{\sqrt{\left[\left(1-\frac{\omega}{\omega_n}\right)^2\right]^2+\left(2\zeta\frac{\omega}{\omega_n}\right)^2}}$$

B. Magnification Factor

The ratio of the amplitude of the steady state response to the static deflection under the action of force (Fo) is known as magnification factor (MF).

$$MF = \frac{F_0/\sqrt{(K-\omega^2)^2+(c\omega)^2}}{F_0/K}$$

$$= \frac{K}{\sqrt{(K-m\omega^2)^2+(c\omega)^2}}$$

$$= \frac{1}{\sqrt{\left(1-\frac{m}{K}\omega^2\right)^2+\left(\frac{c}{K}\omega\right)^2}}$$

$$= \frac{1}{\sqrt{\left[1-\left(\frac{\omega}{\omega_n}\right)^2\right]^2+\left(2\zeta\frac{\omega}{\omega_n}\right)^2}}$$

The magnification factor depends upon

1) The ratio of frequencies
2) The damping factor.

III. THEORY OF FRICTION

Friction is the force between surfaces in contact that resists their relative tangential motion. "Relative tangential motion" is a fancy way to say "slipping". Its direction is opposite to the relative velocity (or intended velocity). Usually, three kinds of friction, depending upon the conditions of surfaces are considered.

1) *Dry Friction:* Dry friction is said to occur when there is relative motion between two completely unlubricated surfaces. Dry friction is subdivided into two types.
2) *Static Friction:* Occurs when the two surfaces in contact are not in relative motion; that is, when one surface is stationary relative to the other surface.Varies in strength from zero (when no external force is trying to force slippage) to some maximum value (just before slippage occurs).
3) *Kinetic Friction:* Occurs when two surfaces in contact are in relative motion; that is when one surface is slipping or sliding across another surface. Is always weaker than the maximum static friction.
4) *Skin or Greasy Friction:* When the two surfaces in contact have a minute thin layer of lubricant between them, it is known a skin or greasy friction. Higher spots on the surface break through the lubricant and come in contact with the other surface. Skin friction is also termed boundary friction.
5) *Film Friction:* When the two surfaces in contact are completely separated by a lubricant, friction will occur due to resistance of motion between the lubricant and the surfaces in contact with it. This is known as film friction or viscous friction.

A. *Laws of Dry Friction*

Experiments have shown that the force of solid friction is

1) Directly proportional to the normal reaction between the two surfaces
2) Opposes the motion between two the surfaces.
3) Depends upon the materials of the two surfaces.
4) Independent of the area of contact.
5) Independent of the velocity of sliding,

B. *Coefficient Of Friction*

An accurate estimate of friction forces induced at interfaces in relative Vibratory motion depends on the accuracy with which the coefficient of sliding friction can be estimated. The simple law of friction established by Leonardo da Vinci assumed the friction force to be proportional to the normal load acting on the interface. The apparent simplicity of this relationship has been the primary reason for its use in most studies of dry friction.

When the applied force (F.) overcomes the frictional force (f) between two surfaces then the surfaces begins to slide relative to each other as shown in Fig.7 The sliding frictional resistance is normally different from the static frictional resistance. The coefficient of dynamic friction is expressed using the same formula as the coefficient of static friction and is generally lower than the static coefficient of friction. Whether u equals to Hd or to his depends on applied force.

Coefficient of friction,

$$\mu = \frac{f}{R}$$

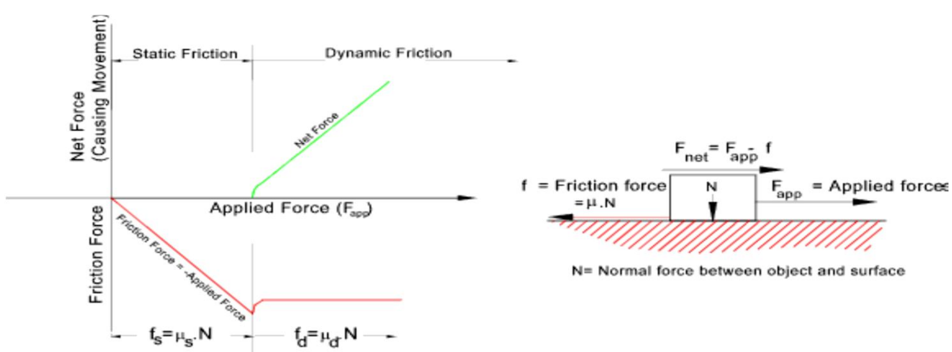


Fig.7 Forces acting on two contacting bodies

IV. VIBRATION IN PARTS OF TURBINE

When elastic members or bodies such as a spring, a beam and a shaft are displaced from their equilibrium position by the external forces, and then release, they execute a vibratory motion. This is due to the reason that, when a body is displaced, the internal forces present in the body act in the form of elastic or strain energy. After these forces the body reaches its original position. When the body reaches the equilibrium position, the whole of the elastic or strain energy is converted into kinetic energy due to which the body continues to move in the opposite direction. The whole of the kinetic energy is again converted into strain energy due to body again returns to the equilibrium position. In this way, vibratory motion is repeated.Any motion, which repeats itself after an interval of time period, is called vibration. The study of vibration is concerned with the oscillatory motions of bodies and forces associated with them. All bodies possessing mass and elasticity are capable of vibration. Most of the Engineering machines and structures experience vibration to some degree and their design generally requires consideration of their oscillatory behavior.

A. Turbine Blade Vibration

Vibrations in a turbine blade are developed due to steady and unsteady loads acting on the blade. The steady aerodynamic load and the centrifugal force in rotating parts, causes a static displacement and steady stresses. Unsteady load can lead to blade vibration and thereby cause alternating stress. Various mechanisms are commonly defined to describe the vibrations of turbo- machinery blades. They are usually classified according to the origin of the excitation and can be of mechanical or aerodynamic in nature. Mechanical excitations include blade tip and casing contact. Aerodynamic excitations includes blade row interactions (forced response), self excitation (flutter), impact of air jets and as well as turbulence of gas.

When vibrations are self-excited (flutter) the vibration motion of the blade itself causes an unsteady pressured field around the blade sustaining the vibrations. Such behavior is usually started by small aerodynamic or mechanical disturbances. It can lead to drastically increasing blade vibration amplitudes and rapid blade failures, if the mechanical damping is too low to dissipate the aerodynamic energy put on the blade. Long and slender structures are more prone to flutter.

B. General Ways To Damp Vibrations

As the vibrations in turbo engines have been a problem since the early beginning of their development, there exist numerous empirical and semi empirical methods to suppress vibrations. There are usually some devices to modify the mechanical behavior of the structure to avoid resonance. And also the adaptation and control of aerodynamic parameters has become a possibility to optimize the vibration behavior of engine components, which in turn requires the knowledge of the unsteady flow phenomena in the machine. The most important measures to control the blade vibration behavior are damping, mistuning and unsteady aerodynamic measures.

The following are some the general methods used to reduce or eliminate vibrations removing external excitation if possible.

1) Using shock absorbers and dynamic absorbers resting the system on proper vibration isolators.
2) Using Damping (Friction damping, Viscous damping, Structural damping)

V. FRICTION IN DYNAMIC CASES AND INDUCED VIBRATION DAMPING

Gas turbine blades are manufactured with high-strength materials offer low inherent damping. The ability of the rotor to withstand vibratory stresses due to resonance or flutter depends almost entirely on the extent of damping due to friction at rubbing interfaces, which are designed to minimize vibratory amplitudes of turbine blades.

The prevailing mechanism is one of dynamic friction at the interfaces, and certain relevant characteristics of the phenomenon are discussed below. The phenomenon of friction between contacting surfaces is an elusive physical mechanism that defies clear comprehension.

The complexity of friction damping arises from variations in the type of time-dependent motions developing at an interface. These variations in the relative motion at the contacting surfaces span the extremes between micro slip and gross motion (macro slip) and include local slip, stick-slip motion, chatter, etc.,

The parameters that control the resulting motion include surface characteristics, normal forces holding the surfaces together and their distribution, properties of materials in contact, surface treatment, temperature, frequency of vibration, and coefficient of dynamic friction. It is assumed that a sub layer of decreased resistance is formed at the interface between rubbing components. One speaks of dry friction forces when the sub layer is in a solid phase. The phenomenon is clearly nonlinear and the feasibility of linearization needs to be established in each application.

A. Modified Mechanical and Damping Devices

The older method to tackle vibration problems of turbo machine blades, once they are detected, is to change the vibration characteristics of the structure by modification of damping and stiffness of the system. Damping is inherent in the aero elastic system in the form of friction damping, material inherent damping (usually small) and aerodynamic damping. It decreases the vibration amplitude whereas a stiffness modification shifts the resonance frequency. However, stiffness modification may not be sufficient for engines with varying operating conditions, i.e. aero-engines. One straightforward and classical approach of mechanical modification is to connect the blades within a blade row with wires or laces. Besides the added weight another drawback of this method is the weakening of the blades due to the necessary holes to fasten the wire or lace. More advanced is the use of part span shrouds, which is common practice to stabilize fan blades at a certain blade height. This measure adds weight to the engine and can be problematic as it disturbs the main flow introducing increased risk of flow separation. The design of the shrouds is an active research area. In turbine engines the use of tip shrouds is possible.The vibration characteristics of the turbine blades can be modified with "blade-to-ground dampers" and with "under-platform dampers". These dampers do not disturb the flow, but modify the vibration characteristic of the blades by changing the friction contact between neighboring blades. Currently, considerable research work is done regarding the design of these dampers but still it is more active area to do research.

B. Friction Dampers

Gas-turbine blade failures can be attributed in many cases to high cycle fatigue caused by large resonant stresses. To avoid such failures, designers of aircraft engines frequently incorporate friction devices into turbine designs in order to increase damping and reduce vibratory stresses, called friction dampers. Damper is a simple metallic piece as shown in Fig. 8 and a blade with damper system is also shown in Fig. 9. In an operating engine, centrifugal forces acting on the damper and lift it gradually, at a certain speed, it fully engages the platform or cover plate at a pre-designed location. The concept, which has been developed over the past three decades or so, is now commonly employed industry-wide and is based on the principle that relative motion can take place as the blade vibrates. This relative motion can dissipate vibratory energy and prevent blade amplitudes from escalating to undesirable levels. The principal parameters that govern the performance of such a damper are: contact load, roughness of rubbing surfaces, level of external excitation, location of the contact region, and mass and stiffness of the damper.

The damping devices are to be of three types :

1) Blade-to-ground dampers
2) Blade-to-blade damper
3) Rectangular contact damper
4) Wedge contact damper
5) Part-span shroud damper

The blade-to-ground (B-G) damper, as shown in Fig. 10 that provide a link between a vibrating point on the blade and a relatively rigid structure such as a cover plate. The blade-to-blade (B-B) dampers, as shown in Fig.11 and Fig.12 it provide a link between neighboring blades. In both cases the damper transmits a load through a friction contact which dissipates energy, When slip occurs.

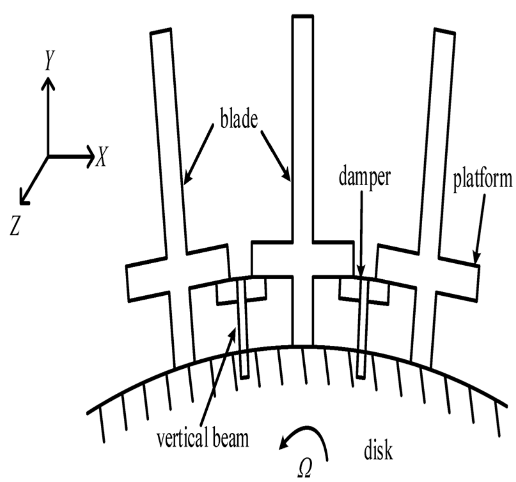


Fig.8 Friction damper

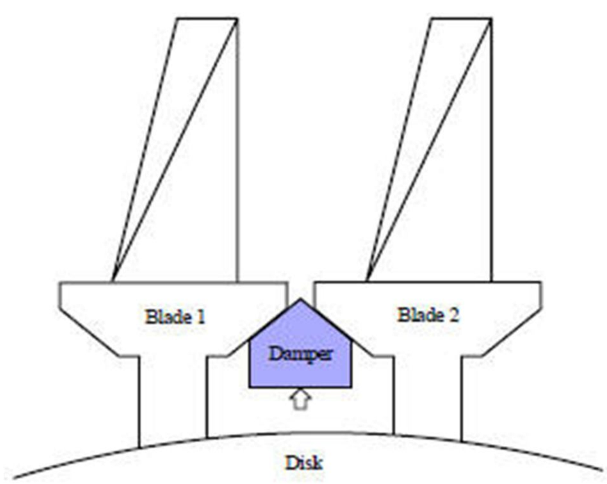
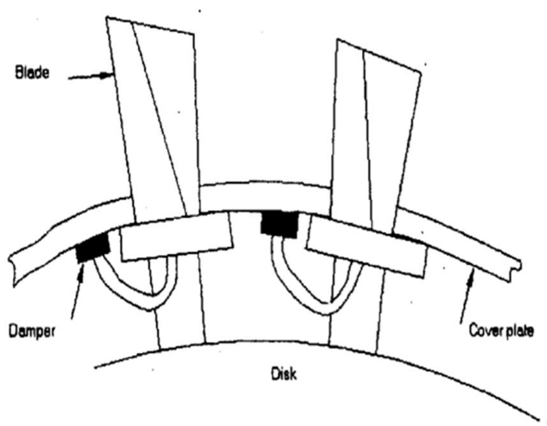


Fig. 9 Blade with damper

International Journal for Research in Applied Science & Engineering Technology (IJRASET)
ISSN: 2321-9653; IC Value: 45.98; SJ Impact Factor: 6.887
Volume 7 Issue III, Mar 2019- Available at www.ijraset.com



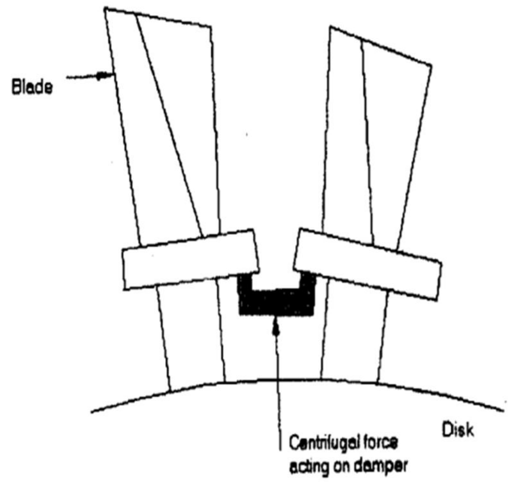
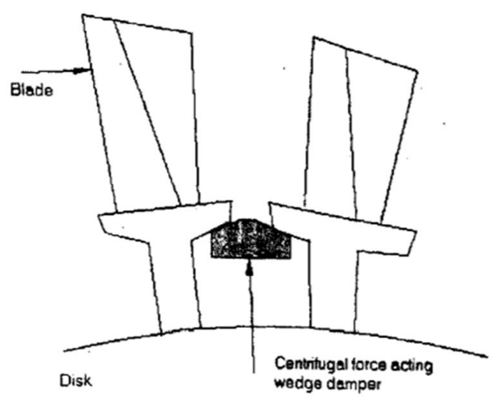


Fig.10 Blade-to-Ground damper Blade Disk Fig.11 Blade-to-Blade damper of rectangular contact type Centrifugal force acting on damper



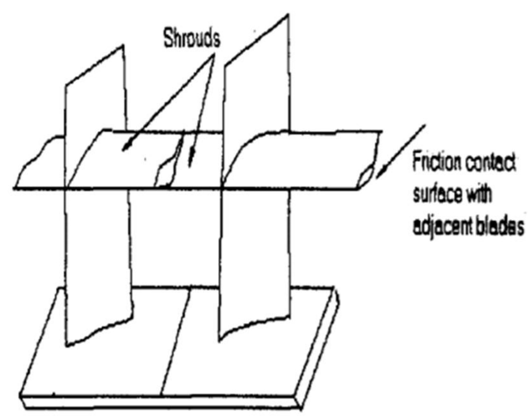
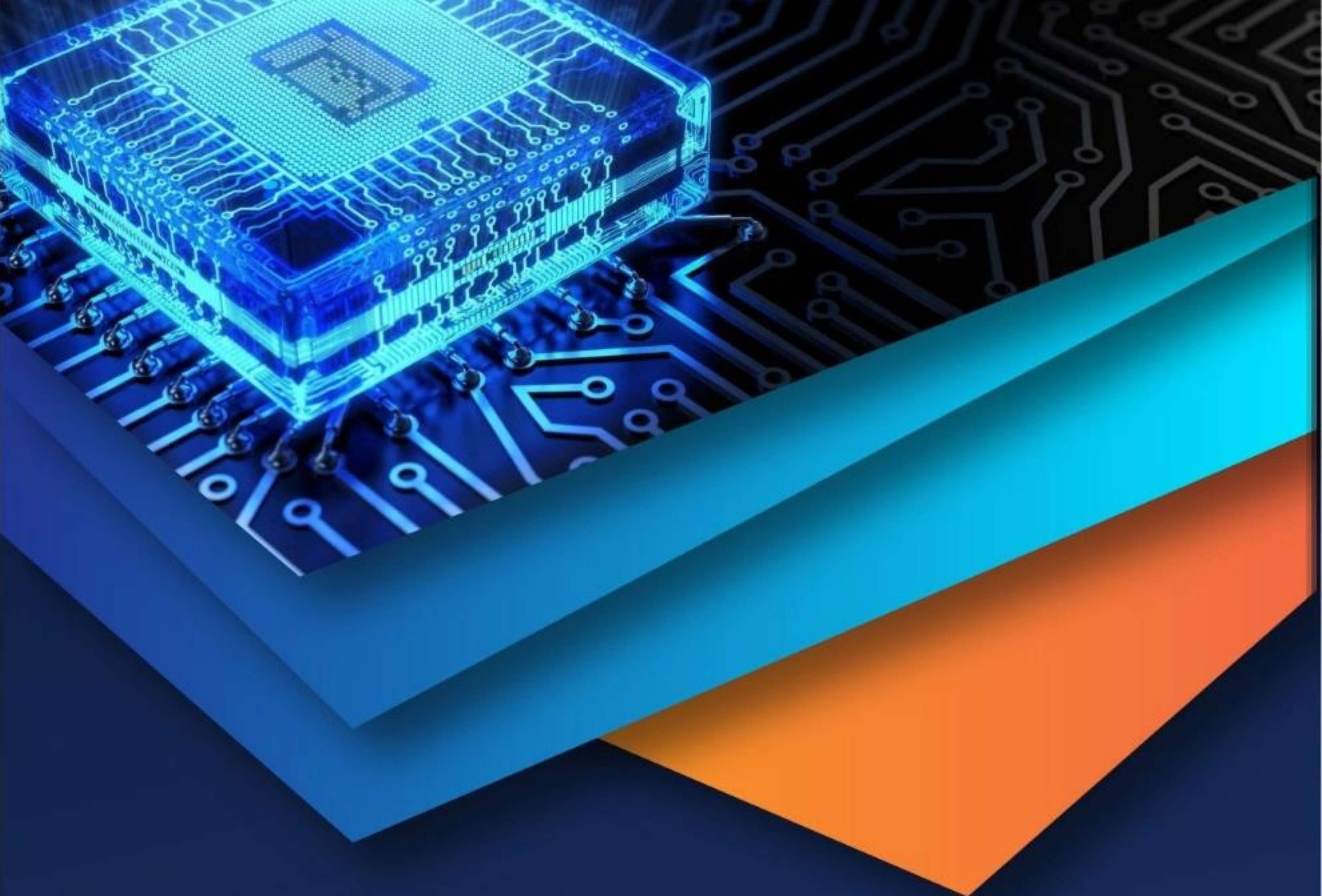


Fig.12. Blade-to-Blade damper of wedge contact type Fig.13 Part-span shroud damper arrangement

REFERENCES

[1] Ewins, D. J., Structural dynamics characteristics of bladed disks AGARD Manual on aeroelasticity in axial-flow turbomachines, Vol. 2, 1988
[2] Ewins, D.J. ,The effect of blade mistuning on vibration response – a survey IFToMM 4th Int. Conf. on Rotordynamics, Prague, Czechoslovakia, 1991
[3] Srinivasan, A. V., Flutter and Resonant Vibration Characteristics of Engine Blades,ASME Journal of Engineering for Gas Turbines and Power, Vol. 119, 1997, p. 742-775.
[4] Slater, J.C., Minkiewicz, G. R., Blair, A. J.,Forced response of bladed disk assemblies – A survey The shock and vibration digest, Vol. 31, No. 1, January 1999, p.17-24
[5] Moyroud F., Fransson T., Jacquet-Richardet G. A comparison of two finite element reduction techniques for mistuned bladed disks ,Journal of Engineering for Gas Turbines and Power, Vol. 124, No. 4, p.942-952
[6] Rivas-Guerra, A. J., Mignolet, M. P. ,Local/Global effects of mistuning on the forced response of bladed disks , Proceedings of ASME TURBO EXPO, 2001-GT-0289.
[7] Judge, J., Pierre, C., Mehmed, O. ,Experimental investigation of mode localization and forced response amplitude magnification for a mistuned bladed disk ,ASME Journal of Engineering for Gas Turbines and Power, Vol. 123, 2001, p. 940-950.
[8] Myhre, M., Moyroud, F.,Fransson, T. H. , Numerical investigation of the sensitivity of forced response characteristics of bladed disks to mistuning ,Proceedings of ASME Turbo Expo 2003, June 16-19, 2003, Atlanta,USA

[9] Kenyon, J.A., Griffin, J. H. Mistuned bladed disk forced response with frequency veering Proceedings of 8th National Turbine Engine HCF conf., 14-16 April 2003, California
[10] Bladh, R., Castanier, M. P., Pierre, C. Component-mode-based reduced order modeling techniques for mistuned bladed disks – Part I: Theoretical models ASME Journal of Eng. for Gas Turbines and Power, Vol. 123, 2001a, p. 89-99.
[11] Jacquet-Richardet, G., Ferraris, G., Rieutord, P. Frequencies and modes of rotating flexible blade disk-shaft assemblies: A global cyclic symmetry approach ,Journal of Sound and Vibration, Vol.191 (5), 1996, p.901-915
[12] Yang, M.-T. Griffin, J. H. A reduced order model of mistuning using a subset of nominal modes ,ASME Journal of Engineering for Gas Turbines and Power, Vol. 123, 2001, p. 893-900.
[13] Feiner, D. M., Griffin, J. H. A fundamental mode of mistuning for a single family of modes, Proceedings of ASME TURBO EXPO, 2002, GT-2002-30425.
[14] Petrov, E. P., Sanliturk, K. Y., Ewins, D.J. , A new method for dynamic analysis of mistuned bladed disks based on the exact relationship between tuned and mistuned systems ASME Journal of Engineering for Gas Turbines and Power, Vol. 124, 2002, p. 586 -597
[15] Lim, S., Castanier, M. P., Pierre, C. , Reduced order modelling and analysis of bladed disks with large mistuning , Proceedings of 8th National Turbine Engine HCF conf., 14-16 April 2003, California
[16] Mignolet, M. P., Rivas-Guerra, A. J., LaBorde, B. , Towards a comprehensive and direct prediction strategy of the effects of mistuning on the forced response of forced response of turbomachinery blades Aircraft Engineering and Aerospace Technology, Vol. 71, 1999, No. 5, p. 462-469.
[17] Mignolet, M. P., Lin, C.-C., LaBorde, B. H. A novel Limit Distribution for the analysis of randomly mistuned bladed disks , ASME Journal of Eng. for Gas Turbines and Power, Vol. 123, 2001a, p. 388 -394.
[18] Petrov, E., Ewins, D. J. Analysis of the worst mistuning patterns in bladed disk assemblies Proceedings of ASME TURBO EXPO, 2001, 2001-GT-0292, Orleans, Louisiana, USA



10.22214/IJRASET

INDEX COPERNICUS

45.98

IMPACT FACTOR: 7.129

TOGETHER WE REACH THE GOAL

IMPACT FACTOR: 7.429

INTERNATIONAL JOURNAL FOR RESEARCH

IN APPLIED SCIENCE & ENGINEERING TECHNOLOGY