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Control of vibration and resonance in aero engines and rotating machinery – An overview

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ABSTRACT

This paper presents an overview of the vibration problems which are experienced in running gas turbines, and other high-speed machinery. The primary problem is that of resonance, where response levels under dynamic loading can be 100 or 1000 times greater than the levels resulting from static loading of the same magnitude. These resonances can be caused by steady, non-oscillatory, forces being applied to a rotating disc and their prediction and observation from measurement under running conditions are essential capabilities for the machinery dynamics engineer. Additional problems can arise if instabilities are encountered, either from aerodynamic sources (flutter) or from rotor dynamics. In all cases where severe vibrations are encountered, they must be controlled by the introduction of extra damping to the critical components, usually by incorporating friction devices. The use of visual displays to illuminate and help to understand the complexities of vibration in rotating machinery structures is presented.

1. Introduction

Whenever a structure is subjected to a dynamic load, its (vibration) response in terms of deflection or stress, etc. can be many times more, or many times less, than the corresponding response that would result if the load was static. In aero engines, 'many times' means 10 or 100 times, and sometimes even 1000 times. This simple statement explains why structural vibration is such a concern, and such a critical issue for heavily-loaded structural components in gas turbines.

Dynamic loads will be generated wherever there is motion. Thus, all vehicles and machines will be subjected to dynamic loads, and, indeed, most stationary structures, such as buildings, bridges, chemical plant, will also experience dynamic loads because of the machines connected to them, or because of the environmental loading through the ground or in the air which supports and surrounds them. However, dynamic loads are a particular concern in machinery because these structures, more than the others, tend to contain very high levels of energy, and with that go large forces, and high vibration.

2. Structural sensitivity and vibration modes

Perhaps the most significant issue, however, relates to the "100-times-greater or 100-times smaller" comment in respect of the response level produced by a given magnitude of force. This comes about because of an acute sensitivity that is possessed by almost all structures to two things: (i) the frequency at which a dynamic force is applied to the structure, and (ii) the precise location at which it is applied. This characteristic is illustrated in Fig. 1 which shows the vibration response at the free end of a supported beam resulting from a unit transverse force applied at different frequencies and at different points along the beam. The resulting map shows how the response amplitude at the reference point varies over a huge range (virtually infinite, because there are some zero response points at certain conditions). Therein lies the complexity of structural vibration, the basis for the possibility of inflicting considerable damage and the potential for avoiding it. Of course, the regions of extremely high vibration are seen to coincide with specific frequencies, and these are the natural frequencies of the structure(s) in question. All the 'sharp' maxima in the plot of Fig. 1 occur at just two frequencies in this case, and these are two of the natural frequencies of that structure. In contrast, the minima in the plot occur at many different locations, and not just two points. These are determined by a second set of properties of the structure – its mode shapes, and here we see just how important these are in determining response levels alongside the more commonly-understood significance of the natural frequencies.

3. Vibration problems in aero engines

In rotating machinery, as in all structures, the adverse consequences of vibration are numerous, and range from the most

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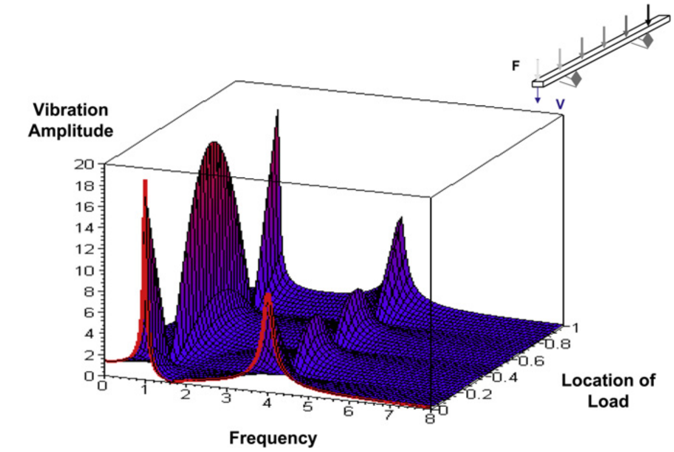

Fig. 1. Structural sensitivity to frequency and location of load.

serious of structural failure, through fatigue or overload, to malfunction of devices required to operate in a high vibration environment, discomfort for passengers, operators or passers-by, and noise. To a first-order approximation, the levels of vibration measured in m or m/s associated with each of these four types of problem can be an order of magnitude different from that which is associated with the next one (so that structural vibration levels that can cause fatigue failure might be expected to be 1000 greater than those which can cause an unacceptable noise). Thus, we see that vibration is an essentially logarithmic phenomenon, in which we must seek to reduce levels by a factor of 10, not just 2, to achieve a significant improvement.

In aero engines and other rotating machines, dynamic forces are experienced everywhere and so most components are subjected to dynamic loads that make them vibrate. Some of these components are already highly stressed because of their function or their location in the machine and, for these, adding vibration can result in a serious concern that early fatigue damage may be inflicted. In general, there are two groups of common vibration problems: those associated with the stationary parts of the engine – the casing, the pipework, the vanes ⋯ and those of the rotating parts – especially the discs and blades. The second group tend to be the more critical because the consequences of even a small failure on a rotating component are magnified by out-of-balance and secondary damage done by debris being thrown off.

4. Sources of vibration excitation (dynamic loading)

There are a number of sources of dynamic loading (excitation forces, in vibration parlance) present in every rotating machine. Perhaps the most common one is out of balance, which is – in effect – a steady force, and not an oscillatory one, that rotates relative to the stationary casing part of the engine. The rotation of this steady force about the engine axis is felt by the casing as a time-varying force which alternates at a frequency equal to the speed of the rotation and is thus experienced as a dynamic load. There is a second source of excitation, equally common and more pervasive, the so-called 'engine-order' excitation which also derives from another static (non-oscillatory) force. This engine order excitation results from the inevitability that the gas stream passing through the engine is never completely axisymmetric in its magnitude. A range of practical features of the engine design and operation, ranging from nonuniformity in the intake, to obstructions such as bearing supports, nozzle guide vanes, combustors, etc all result in the axial flow and pressure exerted by the working gas on the blading being non-uniform in intensity around the 360° of the annulus, at every stage in the engine. This nonuniformity of the steady pressure (load) impinging on the downstream blade row is felt by rotating blades as a time-varying load, and thus it generates a dynamic, vibration, response. This is illustrated by the cartoons in Fig. 2. The essential feature of this type of excitation is that if there is a cosine $n\theta$ component in the non-uniform pressure distribution around the annulus, then this will be seen as a dynamic load (force) which has a frequency of $n\Omega$, where Ω is the rotation speed. There is an added subtlety, which is very significant in its effect, and that is that the dynamic loading, in addition to having an effective frequency for the blades rotating through it, has a spatial variation as well as a temporal one, and that is an amplitude variation of cosine $n\varphi$ where φ is the angular position around the (rotating) bladed disc, while θ is the angular position around the (stationary) engine casing.

There are other sources of excitation, not linked to the rotation and often deriving from turbulence in the flow of external environment, but these do not constitute such major sources of vibration as do the two described above.

5. Descriptions of vibration characteristics in rotating machines

The actual vibration experienced in a running engine will be the 'product' of structural sensitivities and dynamic loads and this response forms the centre of our attention as machine designers and developers. Both when studying measured data, or when seeking to predict what will happen under operating conditions, the response characteristics are the essential indication of structural dynamic performance, and quality.

There is a format of presenting vibration response data from a rotating machine that is particularly useful, and this we shall introduce next. When a machine is running at a constant, or very slowly-changing, speed, it is usually assumed that steady-state conditions apply to the vibration of the various components. Thus if we take a short (duration ~1 s) sample of data from a vibration transducer (Fig. 3(a)) and perform a Fourier Transform on it (Fig. 3(b)) we then have an indication of the frequency content of the vibrations at that speed. If this is repeated every few seconds for, say 5 min, while the engine is gradually changing speed between minimum and maximum values, and we stack the spectra one behind the other, Fig. 3(c), we shall construct a simple waterfall plot of all the analysed time samples. By using the known relationship between instant-in-time and rotation speed, it is possible to construct a diagram showing the vibration response frequency and amplitude features as a function of rotor speed – the z-mod shown in Fig. 3(d).

These 'z-mod' plots contain an extraordinary amount of useful information, in addition to the short-term output of measuring the highest response level, and this can best be seen by using a logarithmic amplitude scale as this illustrates the very low-level vibration behaviour (typically that which is away from resonance regions) as well as the high vibration levels where damage may be being inflicted. Clearly seen on this diagram are a series of radial lines, emanating from the origin of the frequency-speed axes, and a less regular set of lines which are at near-constant frequencies – 'horizontal' lines[1]. The radial lines represent the existence of vibration at most of the integral engine orders, and almost certainly are the result of the various sources of engine-order excitation (static forces in one frame of reference being 'seen' by components in the other frame of reference as a dynamic loading). The 'horizontal' lines represent

[1] These lines are not strictly horizontal, but they vary only very slowly with rotating speed and are not exact multiples of rotation speed as, are the 'radial' lines.