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The Phenomena of Oil Whirl and Oil Whip

1. Introduction

Large machines mounted on fluid film bearings are exposed to many vibration problems including those generated due to resonances. Fluid film bearings contribute in the dynamic characteristics of these systems by influencing the natural frequencies of the rotor system as a whole. Unlike the well-known mechanical natural frequency, the fluid natural frequency is dependent upon operating factors such as shaft speed and shaft eccentricity in the bearing.

Whirl and whip represent the major examples of fluid film resonances. They are characterized by forward precession in a circular orbit with subsynchronous frequency. Journal bearings (which involve radial forces) are the normal location where these instabilities develop. Orbit and frequency spectrum plots can be used to investigate the existence of whirl or whip vibrations.

From the theory of *Complex Dynamic Stiffness*, the interaction among shaft, oil film and bearing plays a major role in the stability. Controlling oil whirl/whip characteristics is achieved through adapting this interaction. The methods used to overcome these instabilities are many and they include: rotor-related solutions, bearing-related solutions and lube oil-related solutions.

In this article, a simplified physical description of the fluid induced instabilities and the associated vibrations is given and supported with illustrative figures. More concentration is paid for oil whirl and oil whip. Also, a basic comparison of the two is given in the last section of this article. The theory of the rotor dynamics behind this phenomenon and the practical approaches to control the oil whirl and whip are left for another article.

2. Fluid Induced Vibrations

Fluid Induced Instabilities (FII) are damaging problems faced in many rotating machinery, e.g., large turbines and compressors. The vibration resulting from these problems limits safe and efficient equipment operation because these problems are directly related to the machine speed and unfortunately may exist over a range of speeds.

Fluid Induced Vibrations (FIV) are described as a special type of self-excited resonance vibration (refer to Figure 1). They are induced by an internal mechanism (oil film bearing, this particular case) that transfers part of shaft rotational energy back to the shaft as a lateral vibration. This mechanism is very much related to fluids, hence, sometimes called Fluid Generated or Fluid Related Instability.

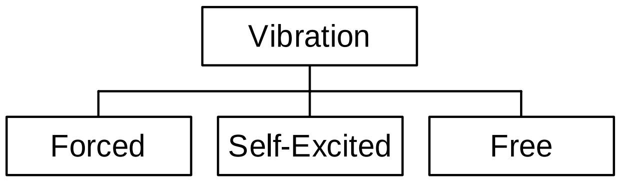

Figure 1 Vibration classes based on source of excitation

Examples of the FIV are: *Oil Whirl, Oil Whip, Subsynchronous Resonance* and *Stall*. FIV could be generated in different fluids. Figure 2 below categorizes the instabilities based on fluid at which the instability is generated.

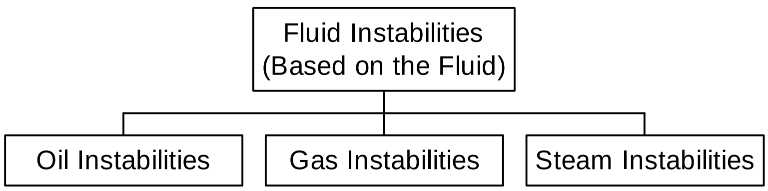

Figure 2 Instability classification based on fluid

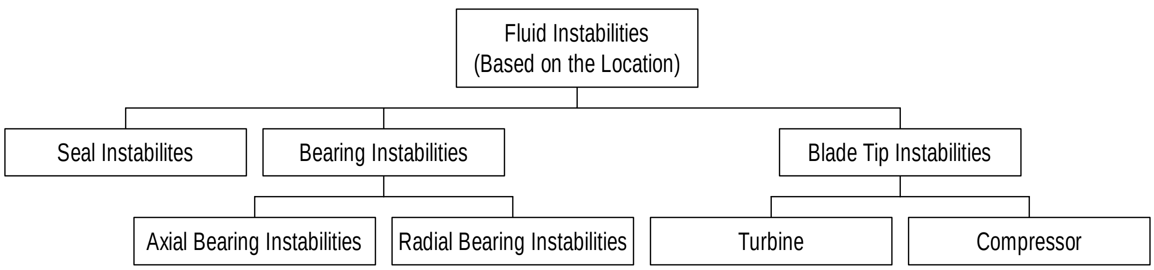

Figure 3 Instability classification based on location

Moreover, FIV could be encountered in many locations in the equipment itself like the tips of the blades, bearings and seals. Figure 3 classifies these instabilities based on the location.

3. Rotor-Bearing: the Interaction and Effects

Oil whirl and oil whip arise when both *Direct* and *Quadrature Stiffness* reduce to zero (for more information, refer to complex dynamic stiffness articles). For an actual rotor, the formulas describing a rotor model, with whirl or whip vibration, are very complicated. Machines mounted on radial fluid film bearings will have two types of resonance: mechanical and fluid. The natural frequency of the whole system (not only the rotor) will be affected by the interaction of the three elements: shaft, oil film and bearing. These elements are partially mechanical (solid materials) and partially fluid (hydraulic oil).

Complex Dynamic Stiffness
Direct Stiffness ≈ Mechanical Based Stiffness
Quadrature Stiffness ≈ Fluid Based Stiffness

In the literature, if resonance is not explicitly specified, usually the mechanical resonance is meant.

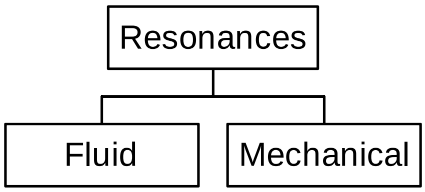

Figure 4 Types of resonance

Notice that the natural frequency of a machine found by the impact test will not be accurate for such machines because it gives you only the natural frequency at zero rpm with minimum fluid film thickness. This explains why natural frequency value collected by an impact test varies sometimes from that observed while the machine runs during start up. To compensate this deficiency, *Frequency Interface*

Charts are developed. These charts provide the machine natural frequency as a function of the rotor speed and with normal fluid film thickness. The simplified figure bellow is an example.

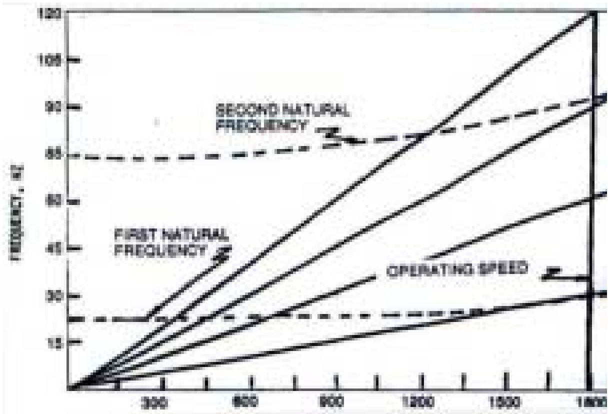

Figure 5 Frequency interference chart

It is important to keep in mind that:

1. The mechanical characteristics (related to the rotor, e.g. shaft mass) are generally speed independent while the fluid characteristics (related to the fluid film and bearing e.g. oil film thickness) are generally speed dependent.

2. In the transient state, where speed changes, don't confuse Vibration *frequency* (in cpm) and *frequency order* (in nX). It is possible have one of them fixed while the other one is variable, however, it is impassible to have both of them constant.

Oil whip is mainly influenced by the mechanical characteristics which are, again, generally independent on shaft rpm. Notice in the cascade plot (Figure 5) the whip frequency is almost constant at about 1400 cpm. This value did not vary even the shaft speed has been increased from 3000 rpm to 6000rpm. The whip frequency order, however, is about 0.47X when the speed is 3000 rpm then the order starts to decrease as the shaft speed increases.

On the other hand, oil whirl is mainly influenced by the fluid-related characteristics that are, again, generally speed dependent. Notice in the cascade plot (Figure 5) the whirl frequency changes with speed. Whirl frequency at 1400 is about 770 cpm then it starts to increase till it reaches 1150 cpm at 2800 rpm. The whirl frequency order, however, is constant throughout this speed range. It is about 0.47X.

3. Whip/Whirl Vibrations: Snapshot

Whirling and whipping vibrations are similar oil induced vibrations encountered with shaft speeds above its first critical speed. Only fluid film bearings (axial or radial) suffer from this instability (being *hydrostatic* or *hydrodynamic lubricated bearing).*

Typically during start up, instability starts with oil whirl and as speed increases whirl continues and then disappears. In most cases oil whip starts thereafter (see Figure 5). From cascade plots, one can notice that whirl (and similarly whip) starts at a certain rpm and continues to survive till it reaches a certain higher rpm where it diminishes. Many research works have been done to predict the starting rpm. The *Instability Threshold* is a common term used for the instability starting rpm. For example, in the cascade plot (Figure 7), the instability threshold of the whirl is about 1550 rpm. Another example is Figure8, where the instability threshold is 6300 rpm.

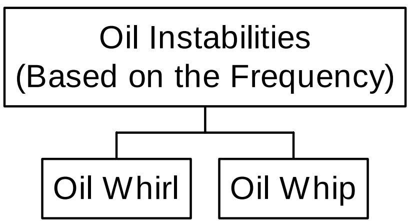

Figure 6 Oil instability classification

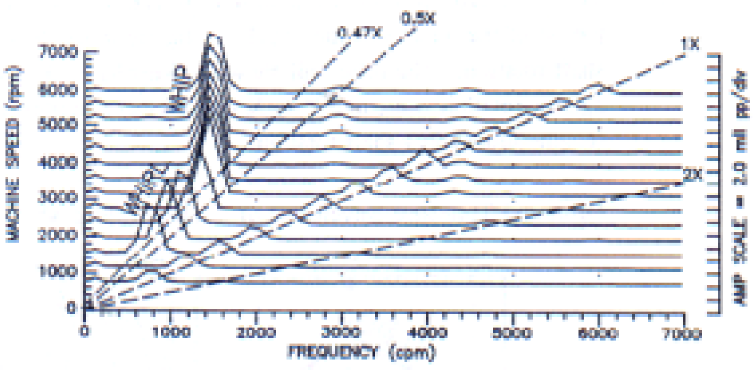

Figure 7 Cascade plot illustrating whirl and whip

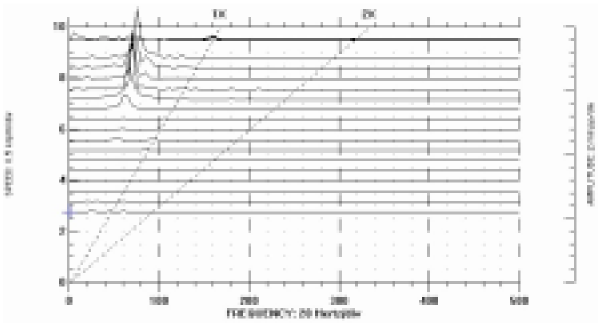

Figure 8 Cascade Plot of FIV

4. Oil Whirl/Whip Symptoms

The typical symptoms of the whir/whip vibrations can be summarized as follows:

1) Subsynchronous frequency (noticed in the frequency spectrum)

2) High amplitudes (reaching to machine's alarm limits)

3) Circular or nearly circular orbits (noticed in the orbit plot)

4) Forward precession (noticed in the orbit plot)

4.1 A subsynchronous frequency, ω, is the one with a value less than the value of the shaft running frequency, Ω, i.e., $\Omega > \omega$. Whirl and whip frequency order is approximately equal to the *Fluid Circumferential Average Velocity Ratio* of the lubrication oil. It is typically between 0.42X to 0.48X. For example, at 2400 rpm, whirl frequency is 1200 cpm or whirl order is 1200/2400 =0.5 X.

4.2 The actual value of the high amplitude generated by oil whirl/whip problems are very much dependent on the machine design and dimensions. In practice, the maximum allowable vibration amplitude of subsynchronous vibration is, usually, set less than the synchronous maximum allowable vibration amplitudes. For example, $Limit_{whirl} = 0.25\ Limit_{Unbalance}$.

4.3 Circular orbits reflect that the shaft whirls inside the bearing clearance without being preloaded in one direction more or less compared to the other direction.

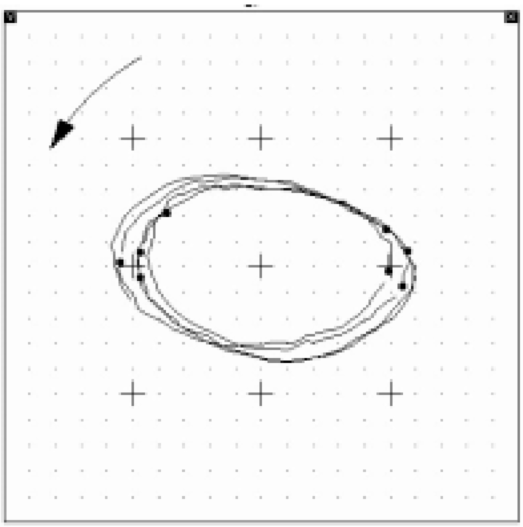

Figure 9 Orbit plot illustrating a fluid induced vibration

4.4 Forward Precession is an orbiting of the shaft about an outside center (e.g. center of the bearing) in the same direction of the shaft rotation. This can be noticed from either an orbit or a *full spectrum* plots. Some rubbing problems have similar symptoms except that they undergo a reverse precession.

4. Comparison Between Oil Whirl and Whip

Whirl	Whip
Forward precession.	Forward precession.
Compared to whip, instability threshold	Occurs under more specific conditions,

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usually occurs with lower rpm.	typically with higher rpm
It depends on rotative speed to start and end.	It depends on rotative speed to start and end.
During its existence, 1. Frequency order is almost constant even with rpm increase. 2. Frequency changes (increases) with rpm-increase.	During its existence, 1. Frequency order changes (decreases) with rpm-increase. 2. Frequency is almost constant with rpm increase.
Circular or almost circular orbit.	Circular or almost circular orbit.

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