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CONTACT MODELS FOR ROTOR/STATOR INTERACTIONS

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Abstract. *Decreasing radial clearances between rotating parts and the housing of turbomachinery can increase its performance; however, this increases the risk of contact interaction between the rotor and its nonrotating surrounding. This contact generates very complex rotor vibrations, which may lead to a catastrophic failure of the machine in just a few rotations. Gaining increased knowledge about rub phenomena is important for improving the ability to protect a machine from damage. Quantitative analyses of contact interaction between the rotating and stationary parts in turbomachinery are relatively scarce and incoherent. This is probably due to the fact that the real data needed for such analysis usually are not known or are difficult to determine. Theoretical studies dealing with these problems therefore usually concern studies of phenomena. This paper is no exception. It reflects the results of a survey of literature on rotor/stator contact interaction.*

Keywords: *Rubbing, Turbomachinery, Whip, Whirl.*

1. INTRODUCTION

Rub is the occasional or continual physical contact between rotating and stationary parts in a rotating machine due to its malfunction condition, which, during normal operational conditions, should not take place.

Two different time scales should be considered: one during the free motions of the rotor when there is no contact (global motion), and the other during the contact period (local motion). The complexity of the phenomena and their local/global nature causes some difficulties for the authors in treating general cases and therefore they discuss some particular aspect of rotor/stator rubs. At the extremes of the rub-related steady-state vibrational response are “partial rub” and “full annular rub” which is characterized by “dry whip”. The first type of rub means that the rotor enters into contact with its nonrotating surrounding only occasionally and maintains this contact during a small fraction of its precessional period. Hence, the most important physical phenomenon is impacting, followed by rotor-free vibration. During the full annular rub dry whip, the friction and vibrating systems stiffness modifications are considerable. The steady-state full annular rub regime cannot be maintained for a prolonged period of time, as it rapidly leads to surface damage and changes in rub conditions, followed by subsequent dynamic transients. During each local rotor/stator rub event several phases can be distinguished: (i) no rub, (ii) rub initiation with impact, (iii) rub interaction in the form of stick-slip chattering (transition phase), (iv) rub interaction in the form of a sliding/rolling contact, (v) separation. At each phase, the contribution of rub-related physical phenomena is different. Each rub event may comprise all or only a few phases.

Causes of rubbing can be imbalances, gravity force, thermal expansion, misalignment, rotor/stator relative motion, and fluid-dynamic forces producing instabilities and self-excited vibrations. Rub occurs when the sum of the shaft central line displacement plus vibration amplitude exceeds the available clearance within the stator or seal. Once the rotor starts rubbing, the system becomes modified and the vibration level usually increases.

In order to predict and prevent rubbing in rotating machinery, modelization of the rub conditions attracted the attention of many authors and researchers (Abdul Azeez & Vakakis, 1997), (Beatty, 1985), (Black, 1968), (Childs, 1979), (Choy *et al*., 1989), (Choy *et al*., 1990), (Choy & Padovan, 1987), (Crandall, 1990), (Dubowsky & Freudenstein, 1971), (Ehrich, 1969), (Ehrich, 1988), (Ehrich, 1992), (Ehrich & O'Connor, 1967), (Flowers & Wu, 1996), (Fumagalli & Schweitzer, 1996), (Johnson, 1962), (Liebich, 1998), (Lingener, 1990), (Muszynska, 1984), (Muszynska, 1989), (Piccoli & Weber, 1998), (Yanabe, 1998), and others.

In this paper the literature survey on rotor/stator contact interaction in rotating machinery is outlined, with an emphasis on the different models for describing the contact. The classification is based on the practical occurrence and their contact condition.

2. NORMAL-TIGHT RUBBING WITH DRY FRICTION

The rotor/stator rub is an abnormal situation that (among other phenomena) provides an increase in the vibrating system stiffness. This effect is often referred to as "normal-tight" situation. There exists a "twin brother" effect consisting of a decrease in the system stiffness, when a normally contacting element of the rotating system becomes loose. The latter situation often occurs when the bearing clearance becomes too large and is usually referred to as "normal-loose" phenomena. The effect of vibrating system stiffness modifications in both normal-tight and normal-loose situations is very similar. In the normal-loose phenomenon the friction contribution is usually much lower. The analysis for the normal-loose motion is mathematically equivalent to a normal-tight condition if Coulomb damping is neglected. Both cause periodically variable stiffness; thus they provide conditions for classical parametric excitation, which may lead to rotor instability.

Childs (1982) has published an analysis of fractional-frequency rotor motion due to nonsymmetrical clearance effects. These theoretical effects are parallel to the experimental results of Bently (Childs, 1993), who demonstrated that large and potentially damaging vibrations which are an exact fraction, generally 1/2, and occasionally 1/3 or 1/4 speed whirling motions. This can result from either excessive bearing clearances or rotor rubbing contact over a portion of a rotor's orbit.

Figure 1, illustrates the steady-state for elastic rotor with static deflection *D* and with contact during a fraction of the orbit**.** The rotor's radial stiffness is increased during contact over a fraction of the orbit, if the orbit radius *A* is slightly larger than the static deflection *D*.

An indication for the severity of contact is the parameter $e\beta/\pi$, where e is the restitution coefficient for impacts.

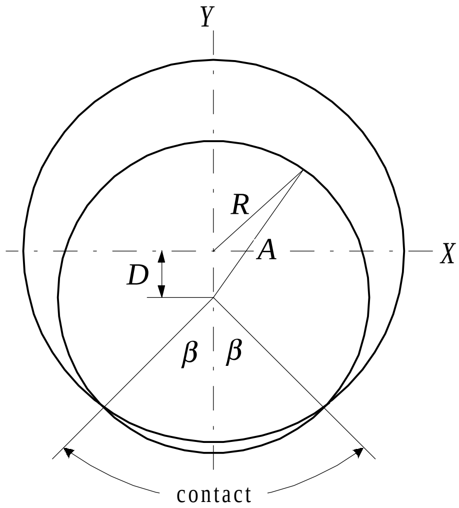


Figure 1- Rotor with static deflection and with contact during a fraction of the orbit.

The partial change in stiffness leads to a parametric excitation. Instability can occur at a range of rotor speed around $\omega = 2\lambda$ where λ is the first natural bending frequency of the shaft (half frequency whirl). Rotor damping decreases and coulomb friction increases the unstable speed range. For high friction any speed above $\omega = 2\lambda$ can be unstable.

3. ROTOR-HOUSING RESPONSE ACROSS AN ANNULAR CLEARANCE

Consider the simple shaft-disk system, with an annular clearance C_r, illustrated in Fig. 2. The massless shaft supported by ideal bearings has effective transverse stiffness k_r at its midpoint, where the rigid disk of mass m_r and radius R is mounted. Concentric with the disk there is an annular stator of mass m_s an inner radius $R + C_r$. The stator is elastically supported by a symmetrical set of springs with isotropic radial stiffness k_s.

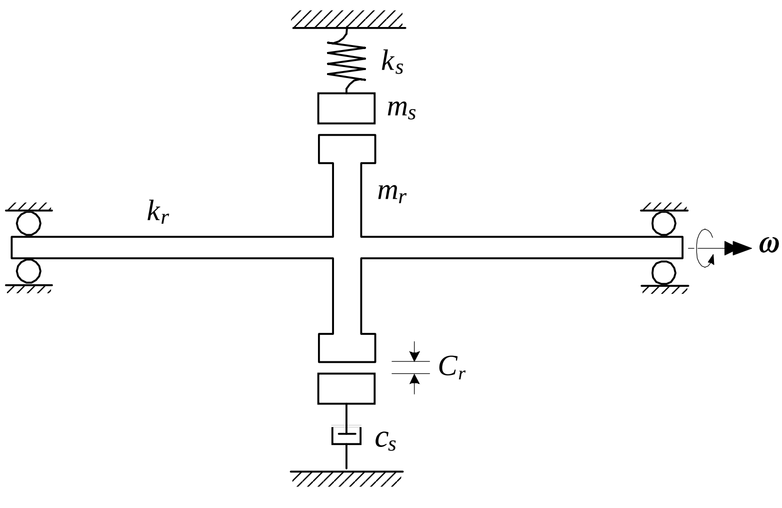


Figure 2- Ideal rotor/stator interaction.

The disk imbalance excites synchronous motion and the resulting precessional motion is forward. As the speed increases, the possibility arises for contact between the shaft and the stator with a consequent coupled motion of the shaft and stator across the annular clearance. When driven by Coulomb friction forces, the precessional motion is backward and supersynchronous. Its precessional frequency is, at first, a fixed factor the times running speed (dry-friction whirl); however, above a limiting running speed the precessional frequency “locks in” to the coupled rotor/stator natural frequency (dry-friction whip). Dry-friction whirl and whip (Black, 1968), are only likely to cause problems when contact arises between a small-diameter rotor and a stator across a large clearance.

Although unlikely to occur, dry-friction whirl and whip can be extremely destructive. The contact is supposed to be continuous and three cases can result from this contact:

3.1 Synchronous rubbing due to imbalance

Figure 3 illustrates general positions for the rotor, $\boldsymbol{r}_r$, stator, $\boldsymbol{r}_s$, and radial clearance vectors, $\boldsymbol{C}_r$, for the idealized model of Fig. 2. The normal contact force $\boldsymbol{N}$ between the rotor and stator is collinear with $\boldsymbol{C}_r$; the friction (tangential) force $\boldsymbol{F}_t$ is normal to $\boldsymbol{N}$ and $\boldsymbol{C}_r$. The rotor response is the response due to imbalance (without contact) minus the response due to the contact force. The stator response is entirely due to the contact force. Because synchronous motion is assumed, slipping occurs continuously at the point of contact; hence, $\boldsymbol{F}_t = \mu_m N$, where μ_m is the Coulomb friction factor.

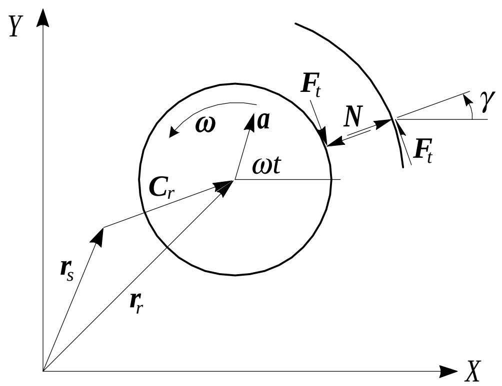


Figure 3- Rotor/stator interaction motion.

The disk imbalance excites synchronous motions, and as the speed increases towards a natural frequency, the response amplitudes will increase and contact may occur. Then the motion of the rotor and the stator will be coupled across the annular clearance. If contact is established it will be maintained throughout one orbit. Coupled synchronous motion continues with increasing speed until the running speed exceeds the coupled rotor-stator natural frequency. The speed ranges or interaction zones where contact will occur depend on the natural frequencies of rotor, stator, and the combined rotor-stator-system, and the response functions connected to these frequencies. Such regions may differ for run-up or run-down.

Synchronous, forwardly precessing motion can occur which is driven by the rotor imbalance. In the frequency range, various rotor-stator interaction zones are possible, and they arise when the response function becomes so large that contact occurs. The frequency ranges for which engagement is possible are only slightly reduced by damping or Coulomb friction.

3.2 Dry-friction whirl

Consider the ideal rotor shown in Fig. 2, the rotor is perfectly balanced and gravity is neglected. The possibility of steady whirling of this system at rate ω when the rotor rotates at the uniform speed Ω .

On the synchronous rotor motion, which is characterized by equal precession and rotation rates for the rotor, large rotor deflections (Childs, 1993) could be anticipated at rotor critical speeds, conditions that arise when the running speed coincides with a rotor natural frequency. At an undamped rotor critical speed, the rotor amplitudes are predicted to grow linearly with time. Flexible rotors are subject to a destructive motion, which has the following characteristics: (a) below a given operating speed, “the onset of instability”, denoted by ω_s, the rotor’s motion is stable and synchronous. Above this speed, there is a subsynchronous component to the rotor’s motion. The onset speed of instability always exceeds the rotor’s first critical speed. (b) For running speeds above the onset speed of instability, the subsynchronous component diverges exponentially with time. The precessional motion associated with the subsynchronous component is in the same direction as the rotor’s rotation. (c) The occurrence (or absence) of rotor instability is largely independent of the state of the rotor balance.

This type of motion is referred to as “whirling”. If a sufficiently large disturbance of either rotor or stator occurs and they are forced into contact above a certain limiting lower speed, a supersynchronous, reverse, precession with the frequency Ω arises then there are several possibilities. The attention is confined to the case where the rotor remains in contact with the stator and rolls backward (Bartha, 1998). The contact friction force has to be large enough to prevent slipping. The no-slip condition requires the rotor to precess at a frequency Ω , which is opposite in direction to the shaft rotation ω, seeing Fig. 4.

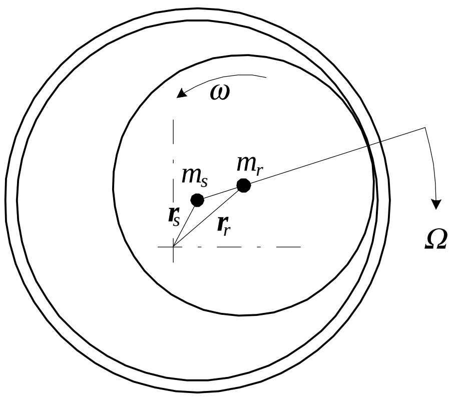


Figure 4- Kinematics for reverse whirl due to Coulomb friction without slipping.

This whirl is purely kinematically determined by $\Omega = r\omega / C_r$. The response functions have to be such that they support sufficiently large orbits, i. e., the rotor speed has to be high enough and the phase between the rotor and the stator motion has to fulfil certain conditions to really lead to dry-friction whirl.

3.3 Dry-friction whip

Friction at the contact point is large enough to allow backward precessional motions, but this whirl will exist with slipping. When the whirl occurs with the natural frequency it is called whipping. In this mode the shaft rolls, while sliding against the seal, in the direction opposite the direction of rotation and maintains contact with the seal.

High normal forces and corresponding friction forces at the contacting surfaces may lead to extremely severe damage in merely a few seconds. Experiments indicate that the amount of Coulomb friction is not the critical parameter. The whirl seems to be insensitive to the addition of lubricants, too. However, the gap ratio C_r /r has to be sufficiently large to induce friction whirl. Dry friction whip or whirl is only likely to occur for contact of a small diameter shaft at a large clearance. Additionally, triggering of contact normally requires an outside disturbance (Lingener, 1990).

When Ω reaches the natural frequency of the coupled rotor-stator system, the motion persists at this limiting frequency even for further increases of the rotor speed ω. It is then called dry-friction whipping.

4. SPIRAL VIBRATIONS FROM HOT-SPOTS

Most of literature is confined to the mechanical effect of rubbing. The influence of the rubbing induced heat on the dynamical behavior of rotors is seldom described in literature. The rub causes friction-related heating and local thermal expansion. The local heat source (hot spot) leads to an asymmetric, thermally induced bow of the rotor. Due to local expansion, the shaft bows, causing an additional imbalance in the system as shown in Fig. 6.

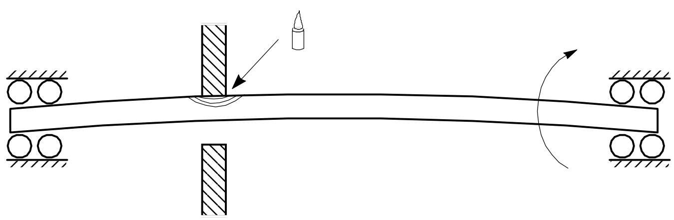


Figure 5- Heat generation through local rubbing leading to thermal bow.

The earliest publications on thermal effect of rub phenomena is often referred to as "Newkirk effect" (Muszynska, 1989). Newkirk pointed out that when a rubbing rotor is running below its first balance resonance speed, the rub-induced vibrations tend to increase with time. Other authors have studied this effect and confirmed that vibrations can grow in amplitude and phase, resulting in "Spiral vibrations" (Kellenberger, 1980). As in Kellenberger's words: "This thermal bowing moves gradually round the shaft circumference and can steadily increase in magnitude". Kellenberger developed a simple linear rotor/stator interaction model that couples the thermo-elastic bow with the rotordynamics

The thermal bow is determined by the balance of heat flow at the local contact point of the rotor. Kellenberger proposed a model of the rub thermal phenomenon assuming a one-dimensional heat flow. The flow of heat into and out of the shaft is only considered. Heat that enters also into the stator is not considered because it does not influence the rotordynamics. A possible thermal deformation of the stator is negligible.

5. IMPACTS

Impact is the sudden physical contact between elements of the system accompanied by characteristic local phenomena, followed by global motion changes. Impact occurs when the low normal force contact of a rotor/stationary part occurs instantaneously with relatively high incoming speed (precessional speed). An impact generates a wide frequency spectrum of exciting forces. The system response then contains components with natural frequencies. Repetitive periodic impacts can result in a definite spectrum of periodic excitation and periodic response of the system.

Rotor and boundary are considered rigid, then the impact is considered elastic or partially elastic. Figure 6 shows possible trajectory of a rotor after the first contact, the gliding phase and finally the rolling. The sequence of impacts dies out in finite time.

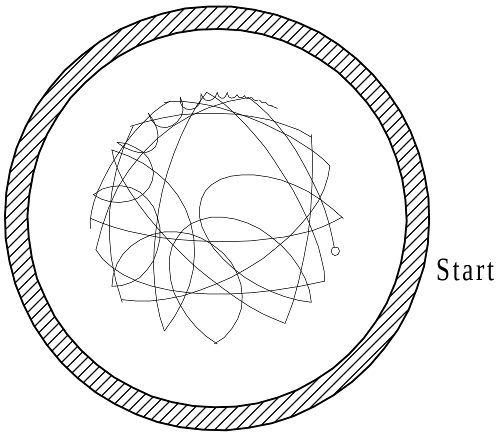


Figure 6- Possible trajectories of a rotor after a touchdown its housing.

The impacting phase is most often modeled by applying straight impact theory with the restitution coefficient that quantifies the loss of energy (Muszynska, 1984). The advanced model for impacting motion of high-speed rotors with significant elastic properties, together with high friction at the contacting surfaces promotes an “adhesive mechanism”. The rotational energy becomes transferred into vibrational energy of the rebounding motion (precessional energy), and the resulting tangential momentum creates conditions for a “super ball effect” (Szczygielski, 1987). A “tangential coefficient of restitution” representing the adhesive friction, is used to model the super ball impact loses.

Accuracy in the description of the looseness and/or rub-related phenomena in mechanical structures with intermittent interelement contacts depend mainly on the adequacy of the impact model (Goldman & Muszynska, 1994).

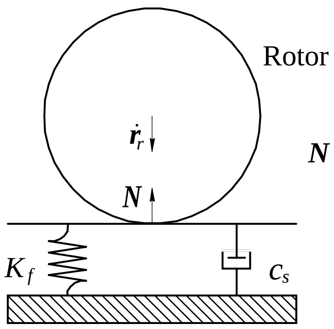
The local/global effect of the impact have one major source of a strong nonlinearity: transition from no contact to contact state between mechanical elements, one of which is rotating, resulting in variable stiffness and damping, impacting, and intermittent involvement of friction. Depending on initial conditions and parameters the energy flow is very different and the generated vibrational response can be subharmonic damped, stable, with limit cycles, as well as by a chaotic pattern, unstable.

There are three approaches employed in the description of impacts between rotational and stationary parts of the mechanical system. The first is based on the classical restitution coefficient model in which an impact is considered instantaneous and elastic (Szczygielski, 1987). The second approach considers nonelastic impact with a zero restitution coefficient; the impact is followed by a sliding stage (Muszynska, 1989). The third approach considers the mechanical system as having discontinuous piecewise characteristics with additional stiffness and damping of the stator at the contact period (Ehrich, 1988). This model seems more accurate, but it creates certain numerical problems, since two different time scales should be considered: one during the free motion of the shaft when there is no contact (global motion), and the other during the contact period by much higher stiffness (local motion). This causes some difficulties in the description of the behavior of the system.

Experimental and simulation results for the case of a rotating disc impacting rigid housing will be shown during the presentation.

6. CONTACT COMBINATIONS

In this model the forces in the radial direction during the impact is considered, as shown in Fig. 7. When the rotor touches the stator, the normal force depends on local stiffness and damping. The tangential force has a mixed viscous/dry pattern, depending on the relative velocity at the point of contact. The spring and the damper are considered linear elements (Fumagalli & Schweitzer, 1996).



$$\boldsymbol{N} = k_s\left|\boldsymbol{r}_r\right| + c_s\left|\dot{\boldsymbol{r}}_r\right|$$

for $\boldsymbol{r}_r > 0$

Figure 7- Model for forces and geometry.

All the contact phenomena introduce nonlinearities, and individually or interactively, they can contribute to chaotic behavior.

7. CONCLUSION

This paper presents survey on contact models of rotor/stator rub. It was shown for different amounts of the rub-related friction, impacting and system stiffness modification it results in significantly different steady-state vibrational patterns. In order to investigate the consequences of potential contacts, one of the key aspects is the realistic modeling of the contact itself. Most of the available contact models are based on simplified assumptions about the geometry of the contact or even about the resulting rotor motion in order to explain certain dynamics phenomena. The papers concentrate on the stationary aspects of the contact-excited rotor vibrations, and these appear to be a general lack of contact models for describing the onset phase of such vibrations. It is this onset phase, which, from the very beginning, defines the character of the contact-excited vibrations. The contact models they are using are simple, mainly using global parameters such as the coefficient of restitution for impact. They do not take into account the finite time of the impact available for the energy transfer between various rotor motions, and they do not give information on the contact forces. It is difficult to integrate the simple models into simulation routines for turbomachinery rotors.

Synchronous motion causes a rubbing over a portion of the rotor's orbit. This results in modifications of the machine motion, due to nonsymmetrical clearance effects. This can result from either excessive bearing clearances or rotor imbalance. Normal-loose variations result from bearing clearance effects; normal-tight variations result from rubbing over a portion of the orbit of a rotor. Dry-friction whirl and whip are only likely to occur for contact of a small-diameter shaft with a (very) large clearance. In addition, triggering of contact normally requires an external disturbance. Although unlikely to occur, dry-friction whirl and whip can be intensely destructive. The permanent slipping will yield wear.

Friction at rubbing surfaces generates heat. A phenomenon arising from rubbing is "spiral vibration". Rubbing in this case causes a thermal bow in the rotor.

Modeling impacts by simple models is a questionable endeavor, as impact physics is quite complex. The shaft incoming speed (precessional speed) is less important than the rotation speed, the tangential effects or super ball. In a short time of adhesive contact (with no relative motion), the rotational energy becomes transferred into vibrational energy of the rebounding motion (precessional energy). The direction and velocity of rebounded motion depends on the amount of tangential moment generated by shaft rotation. Impact create an "after impact" instantaneous response—most often, a rebounding motion of the rotor (separation from the stator) with the initial direction depending on impact conditions and relative tangential velocities in particular.

With the assistance of computers, more complex rotor system configurations with rub effects could be modeled using both finite element and/or modal synthesis techniques. Nonlinear effects can be included in both schemes.

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