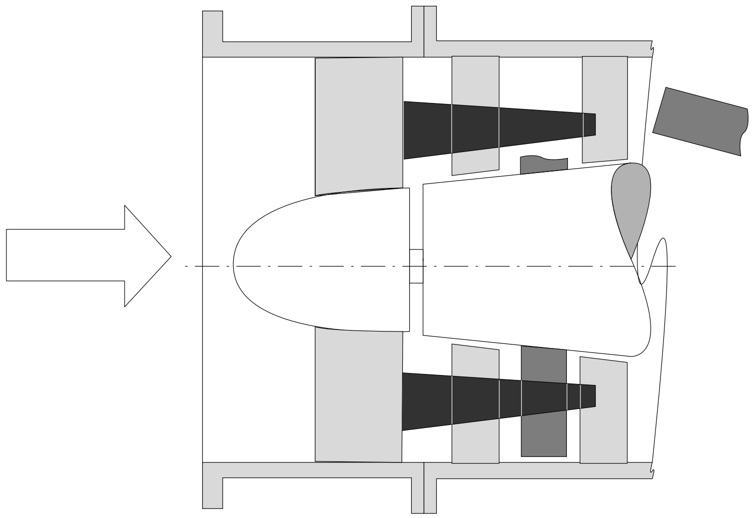
# 5.4.3 Dynamic Fatigue in the Fatigue Limit Range (HCF)



## 5.4.3.1 Vibration Excitation and Vibration Load.

**Dynamic stress in the HCF range** is stress that causes **dynamic fatigue cracks** after roughly **$10^5$ load cycles** (Ill. 5.4-1 and Ill. 5.4.1.1-2). At this dynamic load level, crack initiation occurs without noticeable plastic deformations.

The characteristic constructive design of engines has many thin-walled light structures under high static and dynamic stresses. They are subjected to **high-frequency vibration excitements** such as air and gas flows, mechanical vibrations of contacting parts, imbalances, combustion vibrations, and tooth forces from gears. The design-specific high utilization of the strength leads to corresponding fatigue sensitivity if there are additional, unexpectedly powerful dynamic loads, especially those caused by resonance. The following text focuses on the possibilties of damage-causing vibration excitement in commonly affected **parts** (Ref. 12.6.3.1-5).

HCF fatigue fractures occur at a multitude of different machine elements. To these belong:
- Anti friction bearings (fatigue pittings at the race).
- Blading of turbo engines.
- Components in motors like connection rod, piston, piston pin.
- Gears.
- Bolts.
- Pipe lines.

In the following **components of gas turbines will be exemplary discussed**. The discribed effects can also be found on comparable elements of the mechanical engineering.

Especially modern means of transport like motor vehicles, trains and aircraft show in its design many thin walled, light, statically and dynamically structures highly used to capacity. These are subject of **high frequent vibration excitations** like air and gas flow, mechanical vibrations of parts in contact, unbalances, combustion fluctuations and tooth forces from gears. A high dimensioning caused use of the strength leads to a correspondent fatigue sensitivity. This is especially true, if additional, unexpected dynamic loads trigger resonances. In the case of resonance also little forces are in the position to generate large vibration amplitudes with correspondent high dynamic load. Lacks a noteworthy damping, the amplitudes get theoretical infinite. However this extreme case can be excluded in practice because during exceeding the yield point, **plastic deformations act damping**. In the following shall be pointed at the possibilities of failure causing vibration excitations at especially concerned components/**parts** (Lit 5.4.3.1-5).

**Compressor blades:**

Damage is caused by **flexural modes** of the 1st and 2nd order, as well as torsional vibrations of the 1st order (Ill. 5.4.3.1-4). A tendency to thin cross sections respectively profiles with sharp edges promotes torsion vibration modes of higher orders. So vibration cracks at the corners and parallel to the edges develop (Ill. 5.4.3.3-10). The trend to **integral design promotes coupled vibrations**. An example are so called **blisks** (Ill. 3-16 and Ill. 5.4.3.3-6). Typical is the coaction of blades and disk from the rotorstage of a turbo engine. Vibration modes of higher order are because the little deflection less damped by the surrounding air stream. These are only 'protected' against vibration overload by mechanical friction damping. With this **integral designs** like **brazed or welded stators** and blisks are especially prone for vibration, because they lack from design the friction damping (Ill. 5.4.3.1-9).

**Labyrinths:**

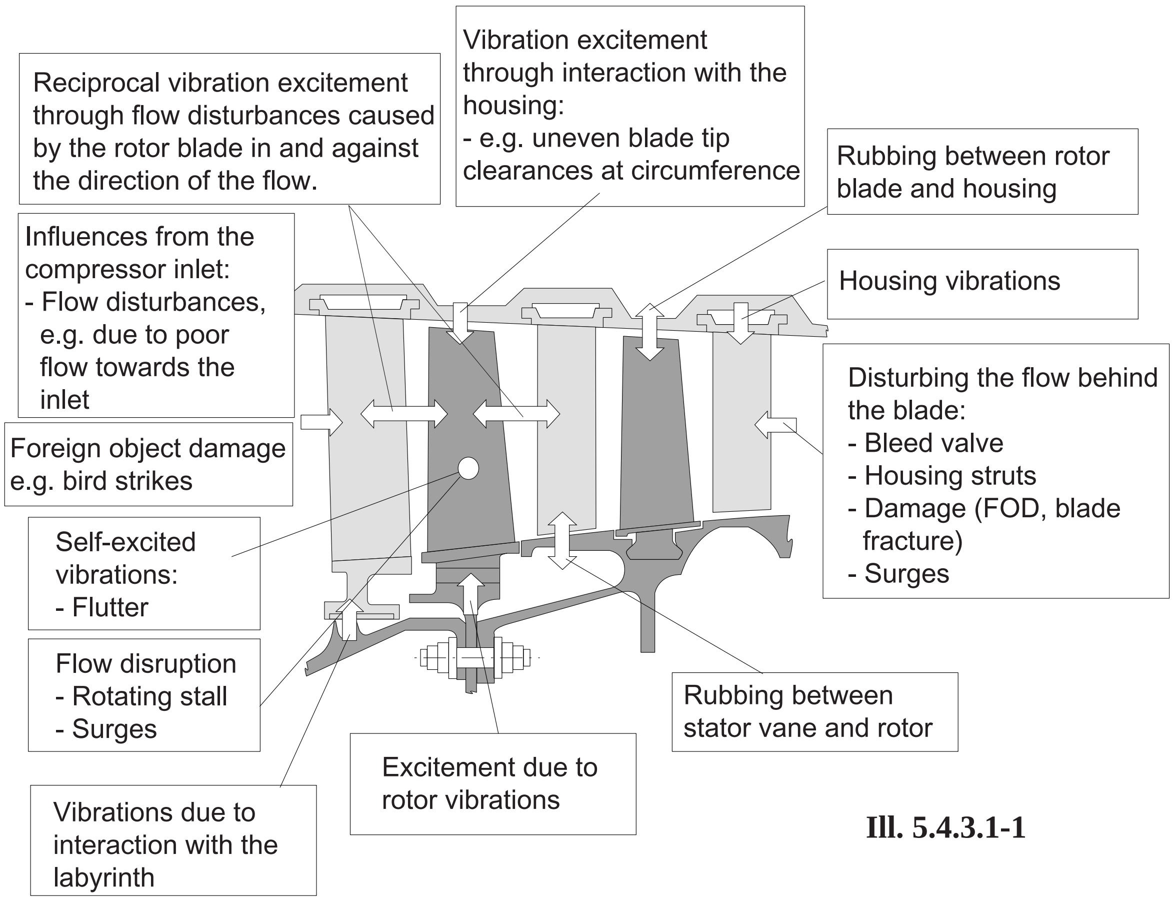
Labyrinths are vibration-sensitive assemblies. Dynamic fatigue fractures are promoted by cracks in rubbing areas (hot cracks).

**Rotors and shafts:**

The excitement mechanisms of these rotating systems are especially diverse (Ill. 12.6.3.1-10). They are based on mechanical, aeromechanical, and aerodynamic effects.

**Disks:** Disk vibrations (Ill. 5.4.3.1-6 and Ill. 5.4.3.1-9) also occur coupled with the blade annulus/rim. Even seemingly **massive parts** (e.g., casted turbine wheels) can be from experience astonishing sensitive for high frequency vibrations.

The blading of compressors and turbines can be excited to dangerous vibrations in many different ways.




**Ill. 5.4.3.1-1**

***Ill. 5.4.3.1-1:*** *This primarily concerns* ***four types of vibration excitement. Recognizing the excitement and type of vibration is a prerequisite for targeted solutions****/remedies and elaborate verification tests in compressors.*

***Wake vortex excitement*** *(Lit. 5.4.3.1-5): This is an excitement that occurs frequently in flows. A zone of low flow speed forms* ***ahead of and behind disturbances*** *of blades and braces. The wake vortex behind the stator vanes reduces the aerodynamic forces on the following running blade in the interference zone. The static pressure builds ahead of the inlet edge of the stator vane. Running blades of the upstream stages pass through these interference zones. The stator vanes are also struck by the flow disruption of the running rotor blades and thus excited. In some cases, this impulse frequency or the harmonic frequency are equal to the resonant frequency of the blade. This results in resonance (Ill. 5.4-4 and Ill. 5.4-5.1) and the danger of high vibration amplitudes.*

*The tendency to* ***continually reduce the axial spacing of the stages,*** *in order to reduce compressor length and save weight, allows the disturbrances to act even more effectively.*

***Flutter excitement:*** *Flutter vibrations (Ill. 3-9.2 and Ill. 3-9.3) can occur as flexural modes (fundamental flexural modes) and/or as torsional vibrations. This is a self-inciting process. It is determined by the ascending force and point of attack of the aerodynamic forces. If flutter occurs, it is difficult to escape this condition. Blade fractures can occur in seconds.*
*Flutter can also be excited by other self-energizing influences. So wheels can flutter under forces of the road surface (Ill. 3-9.1, Ill. 5.4-6 and Ill. 5.4-7.2 „F“).*

***Non-systematic excitements:*** *Experience has shown that flexural modes and torsional vibrations with dangerously high amplitudes can occur in compressor blade rows even without any of the described excitement mechanisms. It is assumed that these vibrations are due to sporadic mechanical excitements and/or random disturbances and vortices in the flow.*
*These excitements include surge shocks, which can cause extreme blade deflection together with high-frequency vibrations (LCF at high-frequency).*

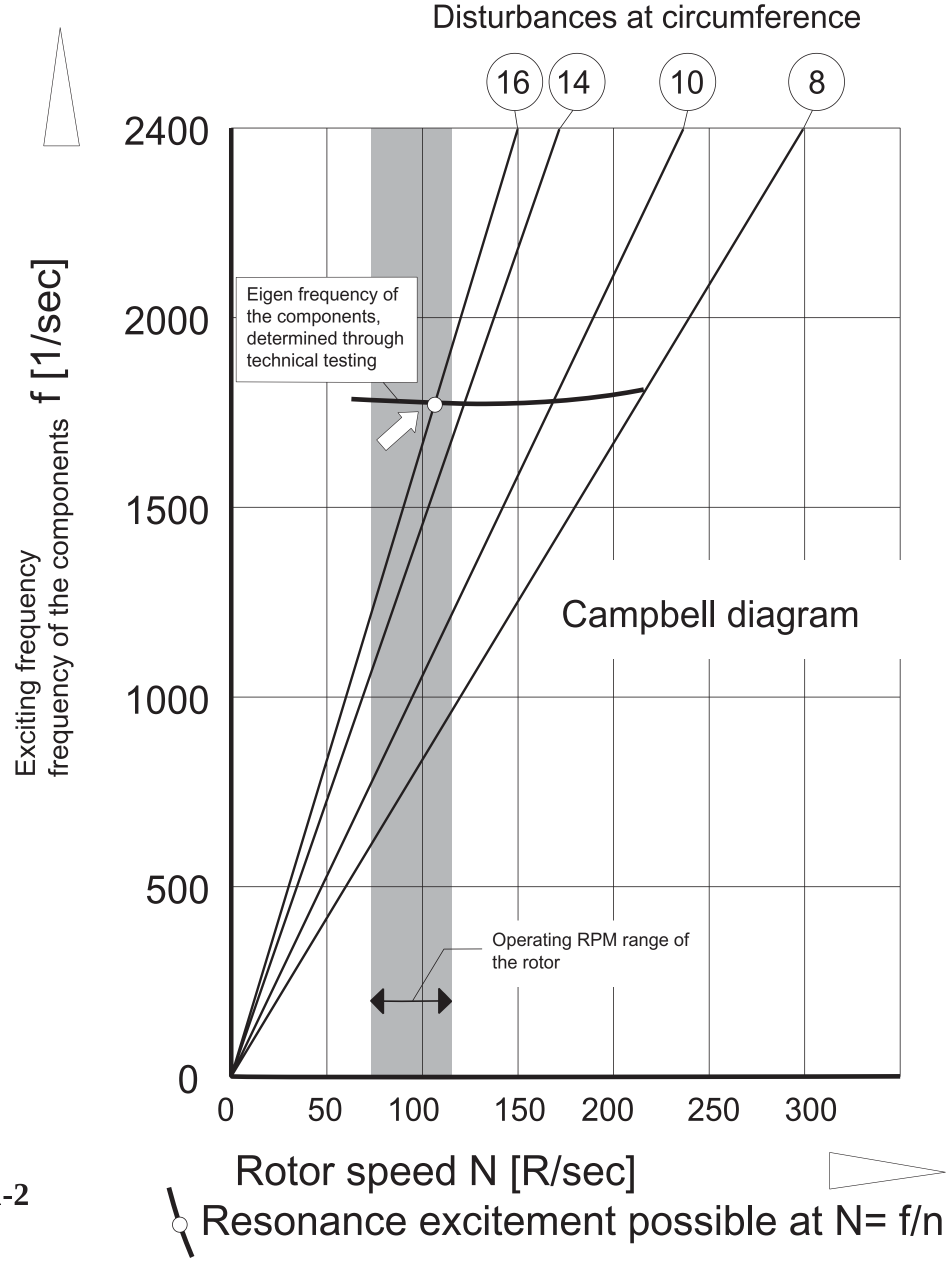
***Unfavorable inlet flow:*** *Nonuniform pressure, velocity and temperature in the intake flow lead to changing aerodynamic loads at rotor blades which run through these zones.*
*Does a* ***ground vortex*** *develop, it disturbes massiv the flow in a narrow limited area during entrance into the compressor. The consequence is a vibration excitation of the* ***following blade stages*** *(not only the first). This happens also during the suction of hot gases. This leads to temperature nonuniformities in the intake flow, which can trigger in an extreme case dangerous blade vibrations.*

***Mechanical excitement:*** *Blades can be dangerously mechanically excited. Mechanisms include vibrations coupled with the disk (Ill. 5.4.3.1-9) and excitements of stator vanes via the housing. The housing/casing is subject to high-frequency excitements from the* ***blade passing frequency****, due to the pressure differences on the pressure and suction sides of the blade tips. These vibrations are transmitted into the stator assemblies, which are fastened in the housing. If the blades are axially fixed by labyrinth rings, then labyrinth vibrations (Ill. 5.4-7.1 „B“ and Ill. 5.4-7.2 „D“) can excite blade vibrations.*
*A unique type of excitement occurs during* ***blade tip rubbing****. This can cause dynamic overstress in a very short time (Ill. 5.4-7.1 „C“). In order to prevent this from occurring, the rubbing systems must be optimized accordingly. Typical influences on the blade loads during rubbing are:*

*- Rotor bearings (location, type),*
*- damping and stiffness of the system (rotor, blades, housing, bearings),*
*- blade fastening in the rotor (integral or inset: jammed, loose, glued-in),*
*- blade bracing (e.g. with/without shrouds, closed shroud, braced shrouds, etc.),*
*- feed motion (size, speed),*
*- contact areas (number, length),*
*-gaps (symmetry, width, length),*
*- run-in behavior of the tribo-system (coating in housing, blade tip).*

*Possible consequential damages include extreme blade vibrations with fatigue fractures within seconds. This is the case, for example, if a fractured blade lays against the housing and is run over by the rotor blades. These experience LCF flexural modes in the plastic zone.*

As far as possible, vibrations are prevented during engine development, but in this case, as well, "the engine will tell us" about the success.




Ill. 5.4.3.1-2

***Ill. 5.4.3.1-2** (Lit 5.4.3.1-3): The so-called **Campbell diagram** makes it possible to **recognize resonances and identify sources of vibration excitement**, not only during investigation of causes following damage, but also much earlier, during the design phase (also see Ill. 5.4.3.3-2 uand Ill. 5.4.3.3-3). In this diagram, the **rotor frequency** (Hz), i.e. the number of rotations per* *second, is plotted on the abscissa, and the* ***resonant frequencies of the vibrating components*** *and their incitations are plotted on the ordinate. For every disturbance, a line can be drawn in this diagram that corresponds to the disturbances around the circumference. The curve of the resonant frequency (not a horizontal line because it also contains the centrifugal force influence and the temperature dependency of the E modulus) intersects with these lines in possible resonance points in the operating RPM range (here, in a single point, see arrow). Typical disturbances in the gas flow include braces, stator vanes, and bleed valves (vent openings). In the case of damage, experimental* ***vibration analyses*** *(e.g.* ***modal analysis****) are used. The damage-causing* ***mode of vibration*** *is most likely to be the one in which the highest loads occur in the crack initiation zone of the damaged part. This is the location with the greatest surface strain i.e.smallest radius of curvature between the* ***nodal lines*** *(Ill. 5.4-5.2 and Ill. 5.4.3.1-5)). Measurements with strain gauges make verification possible. The influence of operation on the actual height of the occurring loads is estimated on the basis of experiential values. Naturally, during design, one will attempt to place the resonance possibilities of* ***important components outside of the operating RPM range*** *of the rotor. During engine development, the resonant frequencies of the components are designed in a way that prevents dangerous resonance from occurring. However, since it is impossible to prevent all potential resonances between startup and full power due to the large number of flow-influencing components, engine operation will ultimately have to provide definitive information. In other words,* ***the engine will tell us.***

***Ill. 5.4.3.1-3: Minor damping*** *can enable dangerous vibration excitements and therefore also lead to dynamic fatigue fractures.*
*Even if no specific damping measures (chapter 5.4.3.3) are implemented, there are many protective damping effects acting on engine parts, such as* ***air damping.*** *These effects are usually* ***rubbing processes between contact surfaces*** *of the root fastening, shroud- and root platforms, and* ***clappers.*** *Another damping effect is provided by coatings that are actually used for a different purpose (e.g. erosion or corrosion protection). If these damping effects, the use of which is generally not a conscious decision, are lost, it can result in dynamic fatigue fractures. Typical mechanisms that break down damping effects include:*

*- Mechanical* ***jamming*** *of socket connections.*

*-* ***Cold welding*** *(****galling/seizing,*** *chapter 5.9.2) of contact surface under fretting stress.*

*- Brazuing or* ***fusing-together of parts by melted*** *contaminants (dust, detached silver, Ill. 5.3-8).*

*- Aging and/or* ***spalling of coatings****.*

*-* ***S****of plastic sleeves from friction bearings.*

*These mechanisms can be attributed to very different causes, including :*

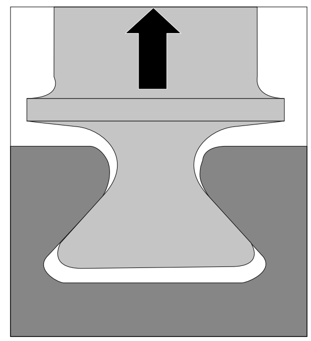
*- Construction/design..*
*- Assembly.*
*- Repair.*
*- Operation.*

If the damping necessary for safe operation is missing or lost during operation, the risk of dynamic fatigue fractures increases.

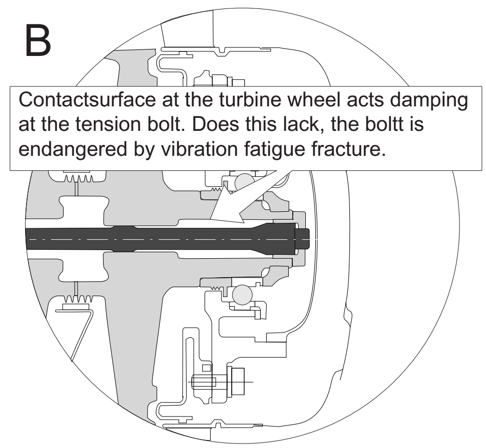
Causes for insufficient damping of vibrating structures, especially stator and rotor blades:

- **Design:**
  - Jamming due to unsuitable tolerances.
  - Strain (temperature, mechanical loads).
  - Geometry (e.g. angle of dovetail connections "**A**")
- **Assembly:**
  - Contact-free positioning of (unrecognized) damping surfaces ("**B**").
  - Use of unsuitable auxiliary materials (e.g. lubricants in areas in which the surfaces are intended to be unlubricated).
- **Repair:**
  - Use of coatings that have rubbing behavior that is considerably different from that of the orginal part surfaces ("**C**").
- **Maintenance,**
  - Use of unsuitable auxiliary materials such as cleaning agents and lubricants ("**C**").
- **Operation:**
  - Jamming due to:
    - Unusual strain.
    - Corrosion, oxidation.
    - Dust deposits.("**A**", "**D**").
  - Macroscopic or microscopic fusing due to fretting ("**D**").
  - "Brazing/soldering through low-melting metals (e.g. silver).
  - Loss of a sufficient oil cushion (squeeze film) in dampened elastic bearings.

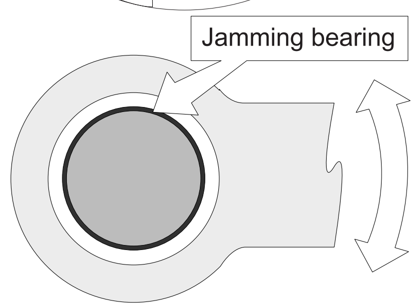
A



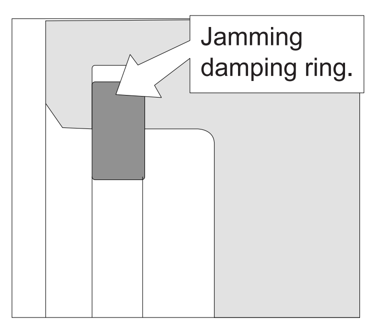
B




C

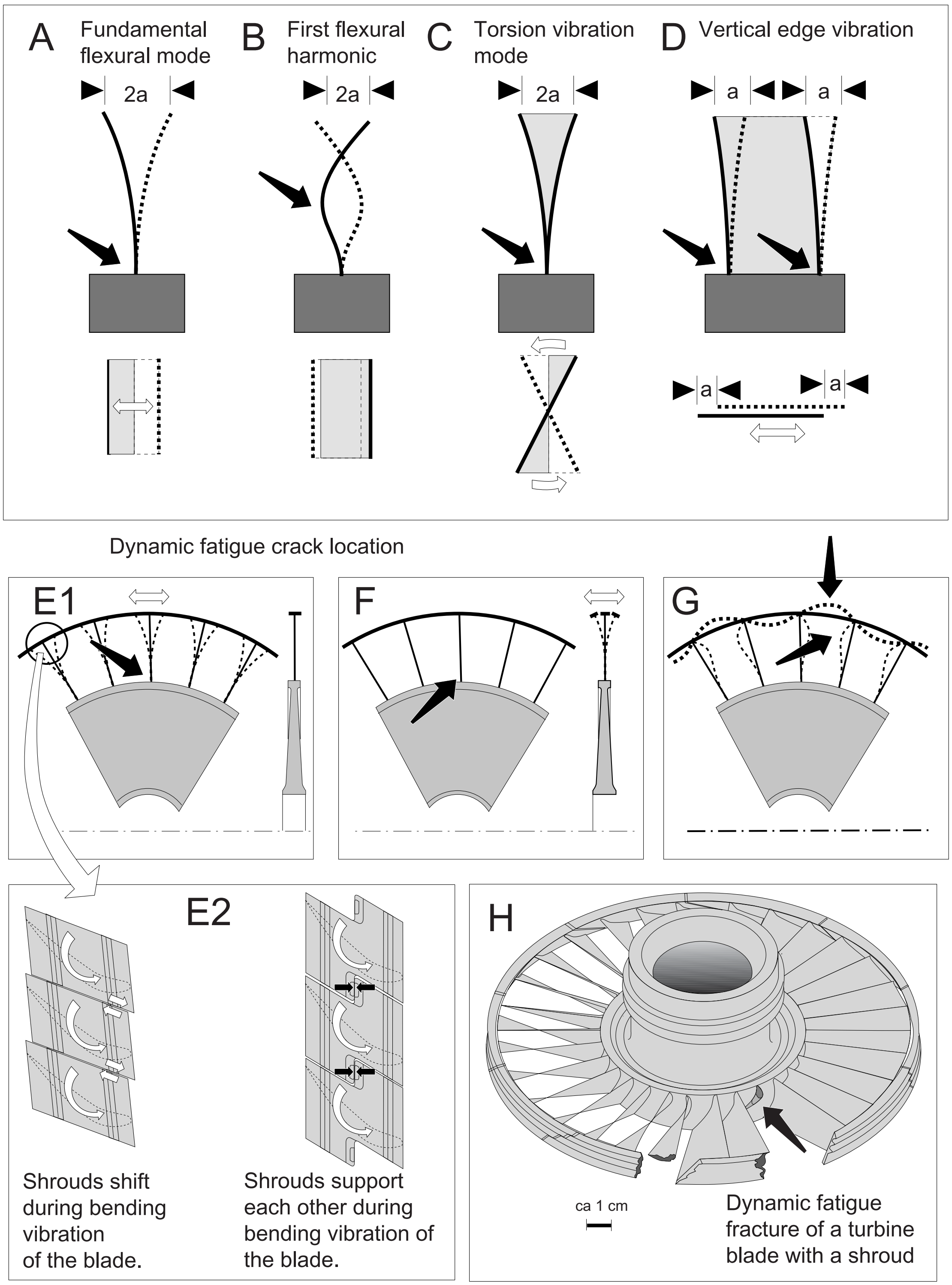



D




**Ill. 5.4.3.1-3**

# Typical blade vibrations:




**Ill. 5.4.3.1-4**

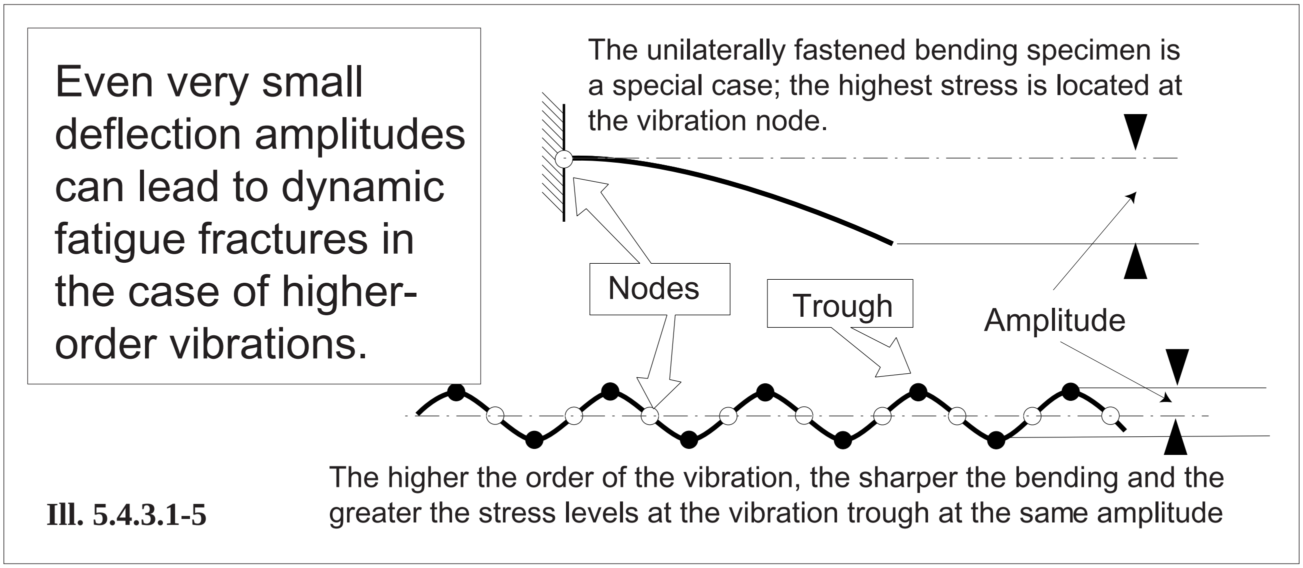
***Ill. 5.4.3.1-4:*** *The* ***fundamental flexural mode*** *(top left diagram, („****A****“) is the most common damage-causing* ***dynamic mode*** *in blades without shrouds. It causes fatigue cracks near the root platform. The first flexural harmonic („****B****“) is in the second diagram from left. Depending on the blade profile, a dynamic fatigue crack is located in this area closer to the center of the blade. The first* ***torsion vibration mode*** *is shown in the third diagram („****C****“) from left. However, there are also a large number of vibration forms of higher orders with especially complex* ***nodal line patterns*** *that can lead to dynamic fatigue fractures ( „****D****“). The top right diagram symbolizes a flexural mode around the vertical axis (vertical edge vibration), which puts especially large amounts of stress on the edges of dovetail roots. In addition, the blade is bent due to its twist.*

***Bracing the blades*** *at the tips (shrouds) and or middle (clappers, snappers) stiffens them and raises the resonant frequencies.However, important factors include the contact surfaces of the shrouds, and whether these are twisted against one another(„****F****“). Experience has shown that these measures are no guarantee for the prevention of dangerous blade vibrations. For example, if a flexural mode causes all blades to be deflected in unison (“****Spike field vibration****“ (****E****“); bottom left diagram), then an axially* ***angular segmented shroud*** *(„****E2****“) is evidently not very effective. (segmented) shrouds will not have a significant effect in the case of coupled blade/disk vibrations („****F****“) in which the tip deflection is primarily axial (bottom right diagram („****E3****“). Here the additional danger of a ‘shingling’ exists In contrast effective is a ‘****Z shroud****’ („****E2****“ right). In this case the support of the shrouds hinders of the twisting a bending vibration of the blades..*

*In a “soft“ shroud, a higher-order flexural mode can develop in the blading and cause cracks below the shroud („****G****“) and/or in the upper half of the blades (Lit. 5.4.3.1-26).*

*The diagram bottom right („****H****“) shows a cast turbine disk from a small shaft-power engine in which many different variations of a cast shroud (non-segmented, segmented with various segment sizes) were unable to prevent frequently repeated dynamic fatigue fractures (HCF) in the blades. The fractures were located both near the shroud and near the annulus.*

***Ill 12.6.3.1-7:*** *The highest stresses in a vibrating part occur in the region of the maximum of the* ***antinodes.*** *If one imagines a bending rod, it is understandable that its bending radius becomes smaller as the bending force increases. The antinodes experience a similar deflection and are highly stressed. The nodes/****nodal lines*** *are in zones with no deflection and* ***minor bending loads.*** *One exception is the unilaterally fixed bending beam (top diagram), in which the highest vibration amplitude occurs in the fixed end at rest, and the lowest stress is in the tip, which is the most deflected section.*




**Ill. 5.4.3.1-5**

*In general, the following is true:* ***With an equal maximum deflection amplitude, the dynamic loads on a part increase along with the order of the excited natural frequency.***

*So tension bolts can suffer dynamic fatigue fractures with minimum deflection (e.g., range 0,1 mm) during a bending vibration of a high order with* ***frequencies up to the ultrasonic range*** *(Ill. 5.4.3.1-3 „B“).*

***Ill. 5.4.3.1-6*** *(Lit 5.4.3.1-7 and Lit. 5.4.3.1-32):* ***Disk-shaped parts*** *such as rotor disks or labyrinth carriers can be excited to very different* ***vibrational forms.*** *These can be classified into three basic types:*

*- Vibrations with* ***nodal diameters (fan-shaped vibrations**, „**B**“).*

*- Vibrations with* ***node circles (umbrella vibrations**, „**A**“)*

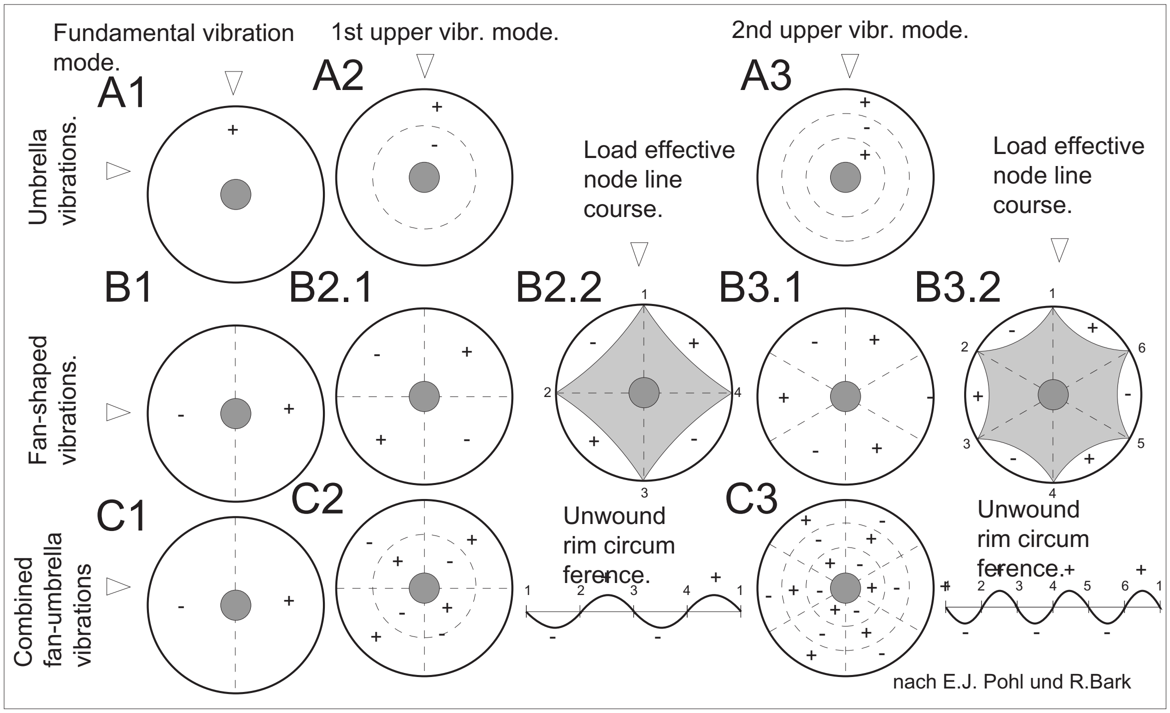
*-* ***Combinations*** *of vibrations with nodal diameters and node circles (**combined fan-umbrella vibratuions**, „**C**“).*

*The ideal nodal line patterns depicted here are changed to polygon-like shapes („**B2.2**“ and „**B3.2**“) in real parts due to the cross-section shapes and geometric characteristics (stiffness changes such as a balance band on a turbine disk, sketch bottom left) or a labyrinth carrier. This also influences the* ***location and progress of dynamic fatigue cracks*** *that are to be expected in the range of antinodes.*

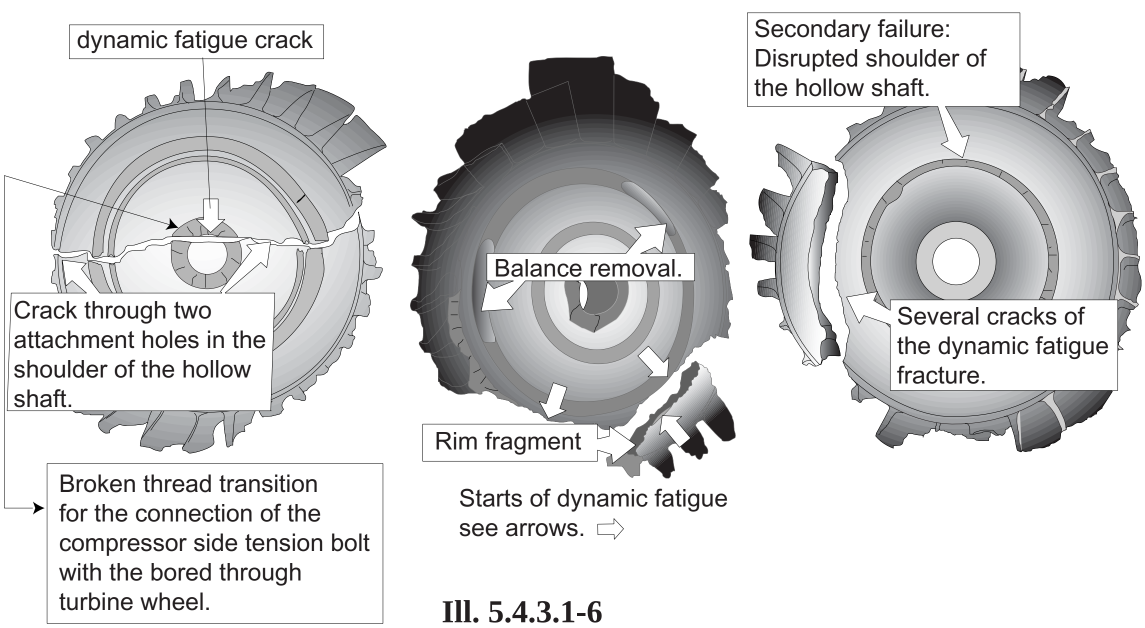
*Below there are all three vibration types shown for failures at single piece casted turbine wheels of a helicopter engine. At the left, correlation with a 1 nodal diameter vibration (**fan-shaped vibration**, „**B1**“). The failure mode below in the middle is caused by a 3 nodal diameter vibration (fan-shaped vibration „**B3.1**“). The failure down right may be assigned an* ***umbrella vibration**, (probably „**A2**“). Thereby combinations of vibration modes are not excluded.*

***Blisks*** *(Ill. 5.4.3.3-9), are not damped by friction at the blade roots, as is the case with inset blades. This means that the problem of* ***mistuning*** *is especially noticeable in blisks. This problem is caused by small production-related* ***deviations*** *within the otherwise acceptable geometric tolerances. This means that* ***unfavorable frequency combinations with neighboring blades*** *can cause individual blades to experience several times greater dynamic loads than the*

Vibrational modes at rotor disks.




Integral turbine wheels/blisks (casted in one piece) from a Ni base casting alloy.



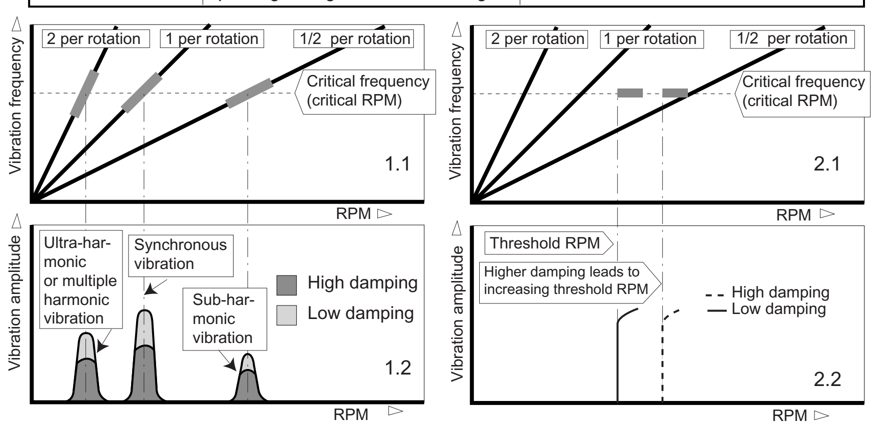

**Ill. 5.4.3.1-6**

*mean value experienced by the blading. Therefore, mistuning is problematic as a measure for preventing resonant vibrations. It can, however, be a useful remedy for* ***flutter vibrations.***

## By what characteristics can vibration excitements be identified?

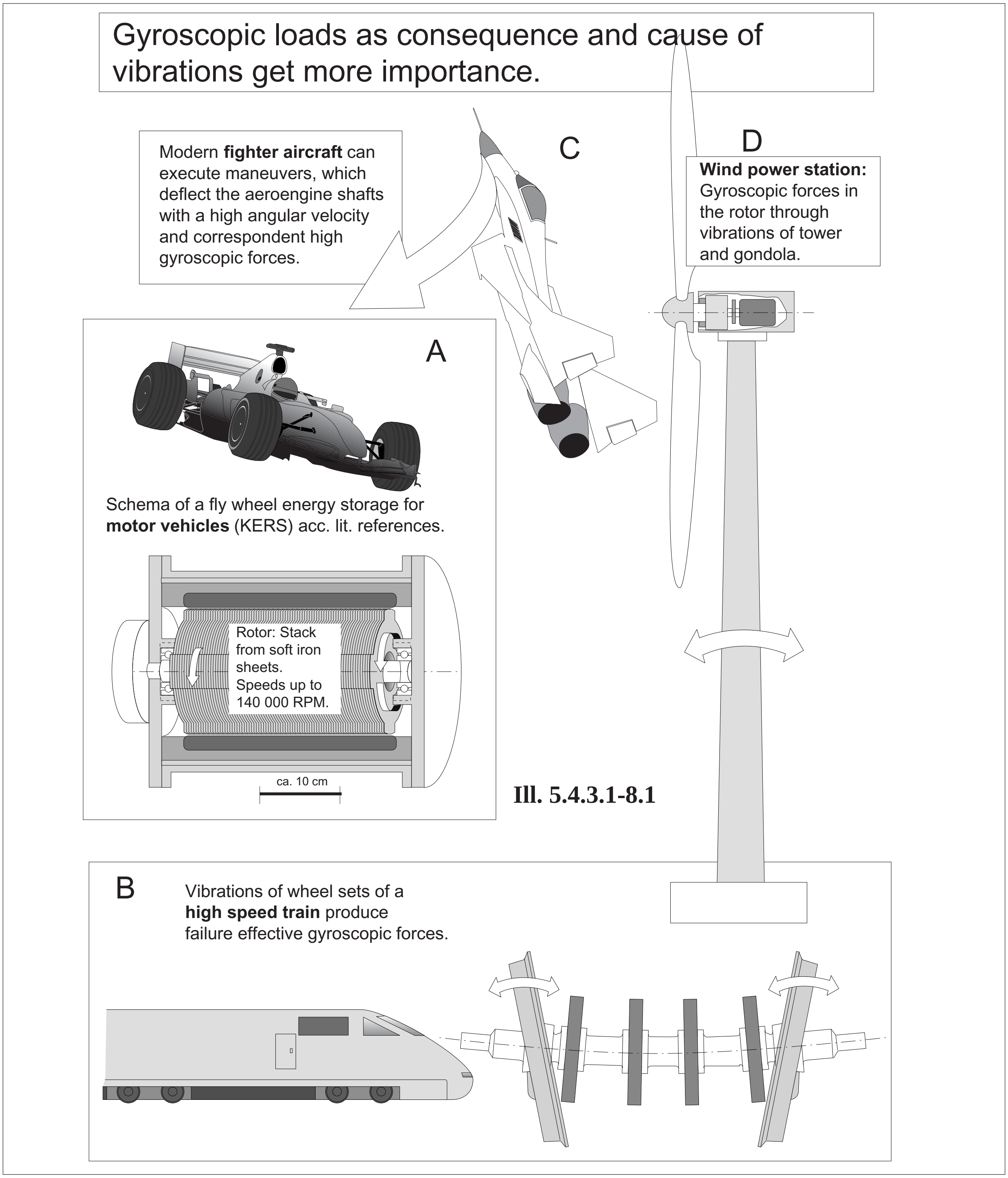
| Excitement type / Characteristic | Excited vibration or resonant vibration | Self-excited vibration or instability vibration |
|---|---|---|
| Dependence of vibration **frequency** on the RPM | The vibration frequency is equal (synchronous) to the RPM or its whole-integer components or products (diagram 1.1/1.2). | Vibration frequency is almost constant and largely independent of the RPM or an external exciter (diagram 2.1/2.2). |
| Dependence of vibration **amplitude** on the RPM | Peak of the amplitude in a small RPM range (critical RPM), or its whole-integer components or products (diagram 1.1/1.2). | The amplitude rises suddenly upon reaching a threshold RPM and then remains at a high or increasing level as the RPM increases further (diagram 2.1/2.2). |
| Influence of damping | Additional damping can reduce the peak amplitude (maximum value), but not the RPM at which the peak amplitude occurs (diagram 1.1/1.2). | Additional damping can shift the threshold RPM to higher RPM levels. Above the threshold RPM, there is no considerable influencing of the amplitude (diagram 2.1/2.2). |
| Rotor geometry | The strength of the excitement and therefore the height of the vibration amplitude depend on unevenness in the axial symmetry (mass distribution) or external forces that act on the rotor (e.g. G-forces). The amplitude can be reduced through smaller imbalances. | The vibration amplitudes are independent of the axis symmetry. A small deflection of an otherwise axis-symmetrical rotor causes a self-increasing amplitude (diagram 2.1/2.2). |
| Vibration frequency | The vibration frequency is close to the critical RPM of the rotor or its natural vibrational modes | The vibration frequency is close to the critical RPM of the rotor or its natural vibrational modes |
| Prevention of vibrations or remedies for them | Tuning the critical frequencies in order to get out of the operating RPM range<br><br>Reducing the imbalances from finishing, assembly and operation<br><br>Damping the system in order to limit the peak amplitudes when passing through critical RPM ranges | Operating RPM must be below the threshold RPM<br><br>Measures against the instability mechanism (Ill. 12.6.3.1-12).<br><br>Damping measures in order to raise the threshold RPM above the operating RPM |

re: F.F. Ehrlich




**Ill. 5.4.3.1-7**

***Ill. 12.6.3.1-11*** *(Ref. 12.6.3.1-16): If* ***characteristic effects*** *are observed within the context of a* ***failure-relevant rotor vibration****, then it becomes possible to draw* ***conclusions regarding the excitement mechanism****. This makes it possible to define specific investigations to determine the cause of damage and strategies for solving the problem.*




***Ill. 5.4.3.1-8.1**: **Failure relevant gyroscopic forces** arise in very different scenariuos. It can come from, that these are not always aware to the unexperienced designer. Here some examples are summarized.*

***Fly wheels** are tested as rotating high speed **energy storage systems** (KERS) in motor vehicles („**A**“, Ill. 5.2.2-0). They can act deteriorating and dangerous during direction changes, uneven road and vibrations. Especially a bad driving* *behaviour must be avoided (Lit. 5.4.3.1-30). Also it may be necessary to consider these during dimensioning of the rotor bearing.*

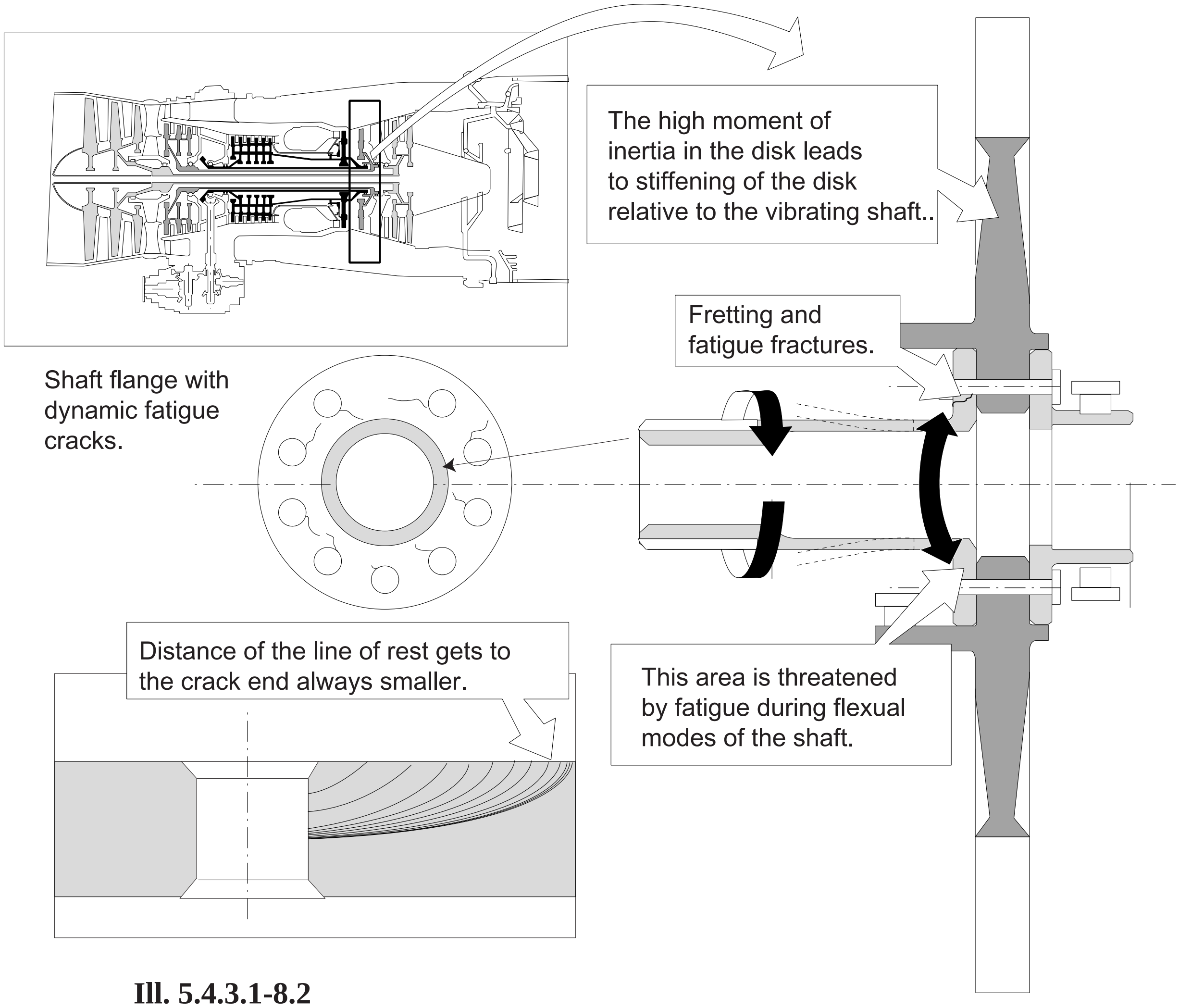
***Wheel sets of high speed trains** („**B**“, Lit. 5.4.3.1-31). This problem can also be representative for other systems at which elastic deflection through vibrations act at the rotor axis. The sketch below right is the markedly exaggerated display of the symetric **bending vibrations from a wheel set**. Also if the tilting of the wheels und with this the axis is actually very small, it may still be sufficient for the developing of deteriorating gyroscopic forces. Possibly this influences an uneven wear of the wheels running surfaces (polygonalization). This problem is extremely serious. There must be seen a connection with a catastrophic train accident. A further example is the influence of the **vibration of the wings at the aeroengines of an airliner**, which are mounted with a pylon at the nacelle. Its **rotation axis** is thereby makedly deflected. This **loads as well the mounting as also the rotor bearing** and its **support in the casings** (strength, stiffness). With these influences the dimensioning must reckon.*

***Aeroengines of fighter aircrafts**: Not, as from a layman may be supposed, a turn is from the aspect of the gyroscopic forces of crucial relevance. Here the deflection of the rotor axis of the aeroengine is relatively slow, because the physis of the pilot limits markedly the g-forces. Therefore the minimum turn radius with several 100 meters is still relatively large. Thereby the physical load at the pilot depends from the speed. As dangerous and dimensioning relevant have shown extremely maneuvers. To these belong expecially movements at which nearly during stand still a **tilting of the aircraft in a fraction of seconds** takes place („**C**“). High gyroscopic loads develop which can highly load the structure of the airoengine können. This leads to elastic deflections of the rotor. These are in the position to bridge **sealing gaps** at blade tips and labyrinths. The reaming during rubbing worsenes permanent the enging performance and the operation behaviour.*

***Wind power stations** Lit. 5.4.3.1-28 and Lit. 5.4.3.1-29): Also here vibrations trigger dimensioning relevant gyroscopic loads at the rotor. Im fact this rotates relatively slow, however its polar moment of inertia is enormous because of the size and the mass.*
*- A **bending vibration of the tower** deflects at the same time with a pitch the nacelle vertically.*
*- **Torsion vibrations** of the tower pivot the rotor axis horizontal.*
*The gyroscopic forces not only load the **bearing** and by means of elastic deformations of **shafts** and **casings/housings** additionally the **gears**. Also the blades are loaded above this by the deflection forces. These elastic deformations can get aerodynamically relevant and complicate the situation.*

***Ill. 5.4.3.1-8.2: Gyroscopic forces** usually act against a deflection of the axis of rotation. Disks with a large polar moment of inertia **stiffen** under*

## The influence of gyroscopic effects on the component loads during shaft vibrations.




**Ill. 5.4.3.1-8.2**

*the influence of gyroscopic force. This increases the* ***dynamic stress in the areas of shaft shoulders.*** *For example, it is thinkable that, at the same amplitude of flexural modes in a shaft, more dangerous dynamic loads will be induced during flight than in a* ***rotating bending test in a laboratory*** *using a solidly fastened shaft. Similar overstress can occur during deflection of the axis of rotation during a flight maneuver. So the gyroscopic forces increase the dynamic load at the transition of the shaft to the disk.*

*A damage mechanism related to gyroscopic forces is indicated by cracking in the flange region with a* ***slowing growth rate*** *(sketch below left, see also Ill. 4.4-1)****.*** *In spite of many cracks which weaken the flange cross section extremely, in several cases never a fracture of the flange occurred. Obviously a so damaged part can 'survive' even hundreds of operation hours. In this case, it can be assumed that the cracking makes the shaft connection more elastic and breaks down the effect of the gyroscopic forces.*

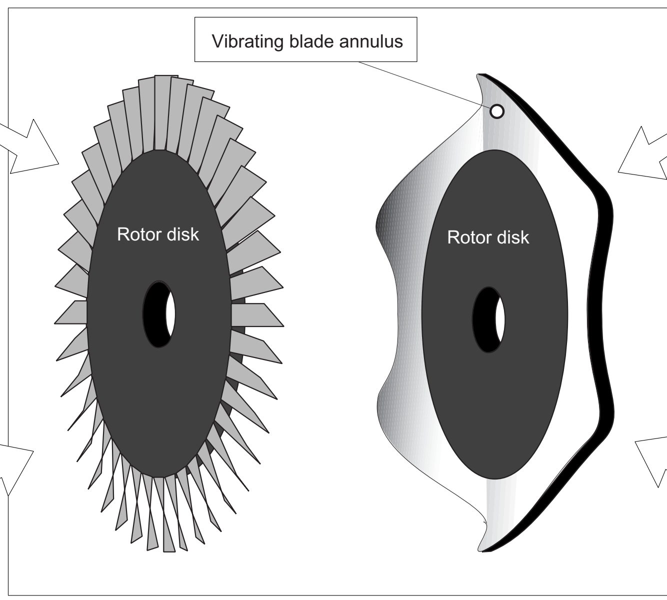
Disk vibrations have a high danger potential, and can be excited in many different ways:

**Excitement via the blading:**

- Flow disturbances
- Rubbing
- Foreign object influence
- Consequential damage
- Alford force

**Excitement through gas forces on the disk membrane:**

- Gas vibrations in ring ducts
- Hydrodynamic paradox




**Ill. 5.4.3.1-9**

**Excitment via labyrinths:**

- Labyrinth vibrations
- Gas vibrations in the labyrinth area
- Alford force

**Mechanical excitments:**

- From the shaft system
- From clamping bolts
- Tooth forces

***Ill. 5.4.3.1-9:*** *A bladed rotor disk usually forms a* ***coupled vibration system****. This is true not only for blisks, in which the blades and disk are integrally connected, but also for disks with inserted blades.Under the typical high centrifugal forces common in operation, the blades sit very firmly in the disk. Due to the lack of damping in the blade root, the exciteability of blisks to dangerous vibrations is seen as especially dangerous. At the typical high frequencies of*

*coupled vibrations, the deflections and, therefore, the **air damping**, are relatively small. The excitement possibilities that are provided as examples have already been covered in different sections:*
***Disk vibrations** are especially dangerous because the possibility, that due to dynamic fatigue cracks fragments develop, which will be „uncontained“ by casings (see also Ill. 5.2.2-0). This problem gets additionally importance in compressors of larger aeroengines with the introduction of **blisk designs**.*

***Excitement through the blading:***

*- Flow disturbances (Ill. 5.4.3.1-2).*
*- Rubbing (Ill. 5.4-6 and Ill. 5.4-7.1).*
*- Foreign object damage (e.g. bird strike).*
*- Consequential damages (e.g. after a blade fracture).*
*- Alford forces (Ill. 5.4-6).*

***Excitement through gas forces at the disk:***

*- Gas fluctuations in annular channels (Lit. 5.4.3.1-32)*
*- Cooling air supply to the rotor.*

***Excitement through labyrinths:***

*- Labyrinth vibrations.*
*- Gas vibrations in the labyrinth area (Lit. 5.4.3.1-32).*
*- Alford forces.*

***Mechanical excitements:***

*- From the shaft system (Lit. 5.4.3.1-32).*
*- From tension bolts (Lit. 5.4.3.1-32).*
*- From gear teeth.of the own gear (Lit. 5.4.3.1-32).*