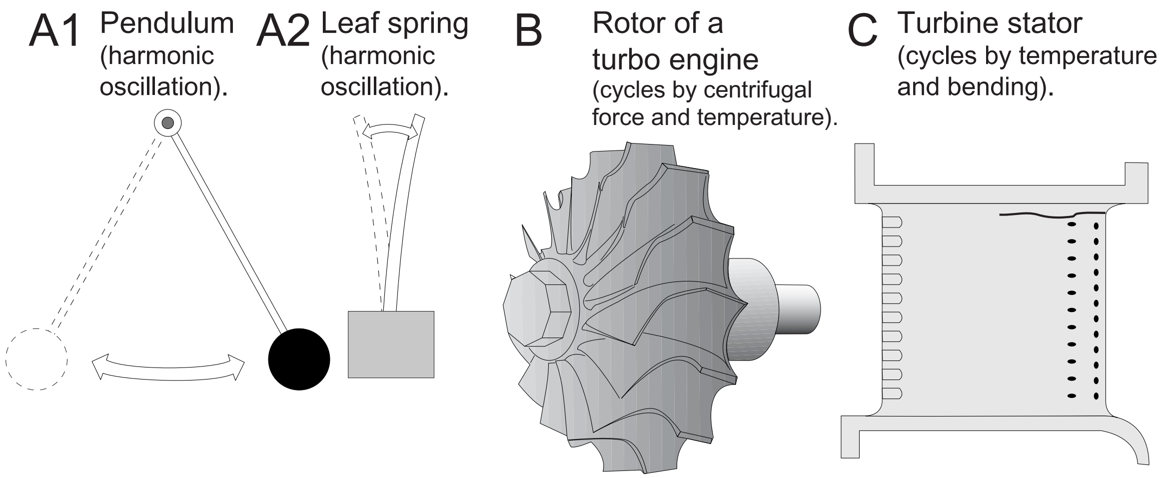
5.4 Dynamic Load and Dynamic Fatigue.



Dynamic fatigue is **strength loss in a part caused by crack initiation and growth under dynamic loads.** If the dynamic loads (Ill. 5.4-1) remain below a material-specific threshold value (fatigue limit**,** better **dynamic fatigue limit**), the number of tolerable load cycles will be sufficient to attain the prescribed life span. Some materials, such as martensitic steels, can tolerate any number of load cycles below the fatigue limit. Materials like **light metal alloys and austenitic steels show no horizontal progression** Wöhler curve in the high cycle range and with this rather **no fatigue limit.**

Dynamic fatigue is the most important life-determining load in many engines (Ill. 5.4-1). In **turbo engines** the low-frequency load cycles during the start-up/shut-down phases play an important role (LCF). Dangerous high frequency vibrations (chapter 5.4.3.1) develop usually through resonances (Ill. 5.4-4). In this case, centrifugal force and restricted thermal strain act in combination (chapter 5.4.2). The hot parts, such as turbine blades (Ill. 5.4.3.1-4) and combustion chambers (Ill. 5.4.2.1-5) are subjected to powerful low-frequency cyclical loads from restricted thermal strain (thermal fatigue, chapter 5.4.2). Of course, virtually all other engine parts are also under dynamic loads. For example, running tracks in roller bearings and tooth flanks on gears are subjected to dynamic loads from the cyclically changing pressure during force transmission. Basically **dynamic fatigue needs always a fraction of tensile stress or shear stress** in the load cycle. In some cases this is only difficult to identify, if the copmponent or the loaded cross section stands under outer cyclic compression load. Example is a notched bar under pulsating compression stress (Ill. 4.3-11 and Ill. 5.4-14) or a overrun surface (Ill. 5.4-14). Dynamic fatigue is compounded by accompanying effects such as corrosion (chapter 5.6.3.2), fretting, (chapter 5.9.3)and the influence of the surrounding atmosphere on the fracture-mechanical behavior (crack growth, chapter4.3).

The load/time graph of the vibrations during operation can vary widely and be very complex. This is true of the stress peaks, rising and falling slopes, and also any dwell times. Depending on the temperatures, materials with different dynamic strengths react to these influences with varying degrees of intensity. Titanium alloys can tolerate far fewer load cycles in the LCF range if there is a dwell time of several minutes at maximum stress level (high temperature low cycle fatigue=HTLCF; Ill. 5.4-12 and Ill. 5.4-13). Creep and fatigue damages evidently occur during the dwell time.

In this case it is rather unlikely that a failure accumulation of deteriorations from creep or fatigue during the dwell time is concerned. So this extreme behaviour can not be explained. Obviously rather the diffusion processes in connection with the production process of the semi-finished parts play a role. For this the hydrogen dissolved in sensitive structure features are made responsible.

A **dynamic fatigue fracture** is also called only **fatigue fracture**. This term does not satisfy because it is also used for creep. Additionally because fractures/cracks are concerned, deteriorations which already before took place, are not addressed. But just these have gained importance with the dimensioning limited lifetime of vibration loaded parts (see below). Therefore **dynamic fatigue** as deterioration designation should be preferred. In the term 'vibration fatigue' a vibration as cause is obvious. So the question arises what is understood under a **vibration** (oszillation, Ill. 5.4-4, Lit. 5.4-2).

Concerned is the sequence of a constitutional change of a system out of balance, which is brought back by a restoring force into the original condition and at which certain features recur. At a **periodical oscillation** a periodical energy transformation between kinetic energy and stored energy (potential energy) occurs within a **fixed time interval**. A special but also frequent case is the **harmonic oscillation.** It follows a sinus curve. A typical example from physics is the pendule („**A1**“). This definition applies to many high frequency **free and forced oscillations** (excited from the outside) in the technique. This is especially true in the resonance range (Ill. 5.4-4). To this belongs the vibration of an one-sided fixed cantilever (sketch „**A2**“).

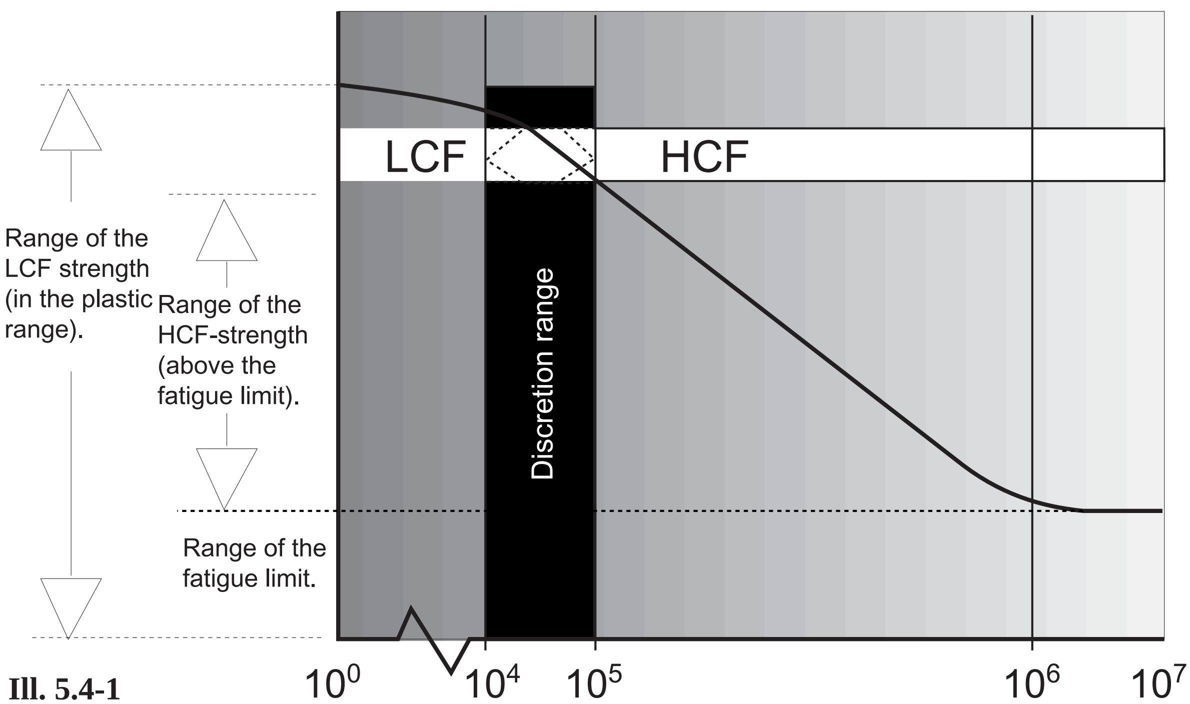
A further problem is, that many dynamic operation loads are not periodic. This is for example true for thermal fatigue and load cycles of rotating parts. With this for many dynamic technical processes the term **dynamic fatigue** is no hint at a periodic oscillation. An example is the cyclic loading of the rotors from a turbo engine with irregular start-stops and changes of the rotation speed at spontaneous power changes („**B**“). Because there are not concerned oscillations according the definition above. Every cycle can be individual in load progression and height. A repetition in a fixed time interval (periodic) is rather unlikely. Also thermal fatigue (chapter 5.4.2) which is based on thermal stresses, is not subject to a vibration/oscillation in the defined sense. So the comprehensive generic term „**dynamic load**“ seems the most unproblematic. It contains all possible loads, leading to **dynamic fatigue**.

Dynamic fatigue fracture originally correlated rather the to day understanding of a HCF failure in the range of the finite life fatigue strength (Ill. 5.4-1). So loaded systems are dimensioned by means of the fatigue limit (no fracture/crack also after arbitrary many cycles). Fractures at loads markedly above the fatigue limit are also termed as **short time fractures**. Today the materials behaviour under **cyclic plastic deformations** (LCF, chapter 5.4.1) is element of the dimensioning (damage tolerant design). With this the term **fatigue fracture** gets diffuse. It can be ambiguous. There is also a creep fatigue (statical loaded) and corrosion fatigue. To distinguish these concerned the failure mechanism, it is better to speak also about **dynamic fatigue**.

Because of the confusing terms the english identifications **HCF** and **LCF** für for the load respectively HCF and LCF fracture/crack for the failing prevailed (Ill. 5.4-1).

In **notches** (Ill. 5.4.4-1) a markedly geometry, material and temperature depending drop of the dynamic fatigue strength occurs. This is in the HCF range, compared with the LCF loading, especially pronounced (Ill. 5.4-3). The **temperature dependency** of the HCF strength shows exemplary Ill. 5.4.3.2-4.

Load/stress ranges in the Woehler diagram.



Ill. 5.4-1

Ill. 5.4-1: *Many construction materials like heat treatable steels show a dynamic fatigue limit. From this dynamic load the* ***woehler curve progresses horizontal****. In the woehler diagram the load cycles are usually logarithmic applied on the abscissia, the stress amplitude at the ordinate. Below this load respectively* ***dynamic fatigue strength*** *arbitrary many load cycles will be endured. This is naturally only true, as long as not further operation influences act like corrosion or wear. However there are many materials which show* ***no pronounced dynamic fatigue limit****, i.e. whose* ***woehler curve*** *drops further also at high load cycle numbers. To these belong light metal alloys (Al, Mg, Ti), Ni alloys and austenitic steels. Anyway there can be specified useful thresh holds for the dynamic strength, if the very high cycle numbers in the operation times will not be reached in practice. Are loads endured above the dynamic fatigue limit with more than about* 10^5 *cycles till failing, we speak about* ***high cycle fatigue*** *(HCF). Below about* 10^5 *cycles we speak about* ***low cycle fatigue*** *(LCF). This means those strenth values will be used for the dimensioning of machine components with very different dynamic loads. This is true for time periods and sequences. From this, also the loading frequency is concerned, as far to speak at all about this. So the time as distinguishing parameter is rather confusing. For LCF and HCF the term fatigue stands for the charakteristic failure mechanism.*
Therefore the terms ***low cycle fatigue (LCF)*** *respectively LCF strength and* ***high cycle fatigue (HCF)*** *have prevailed international.*

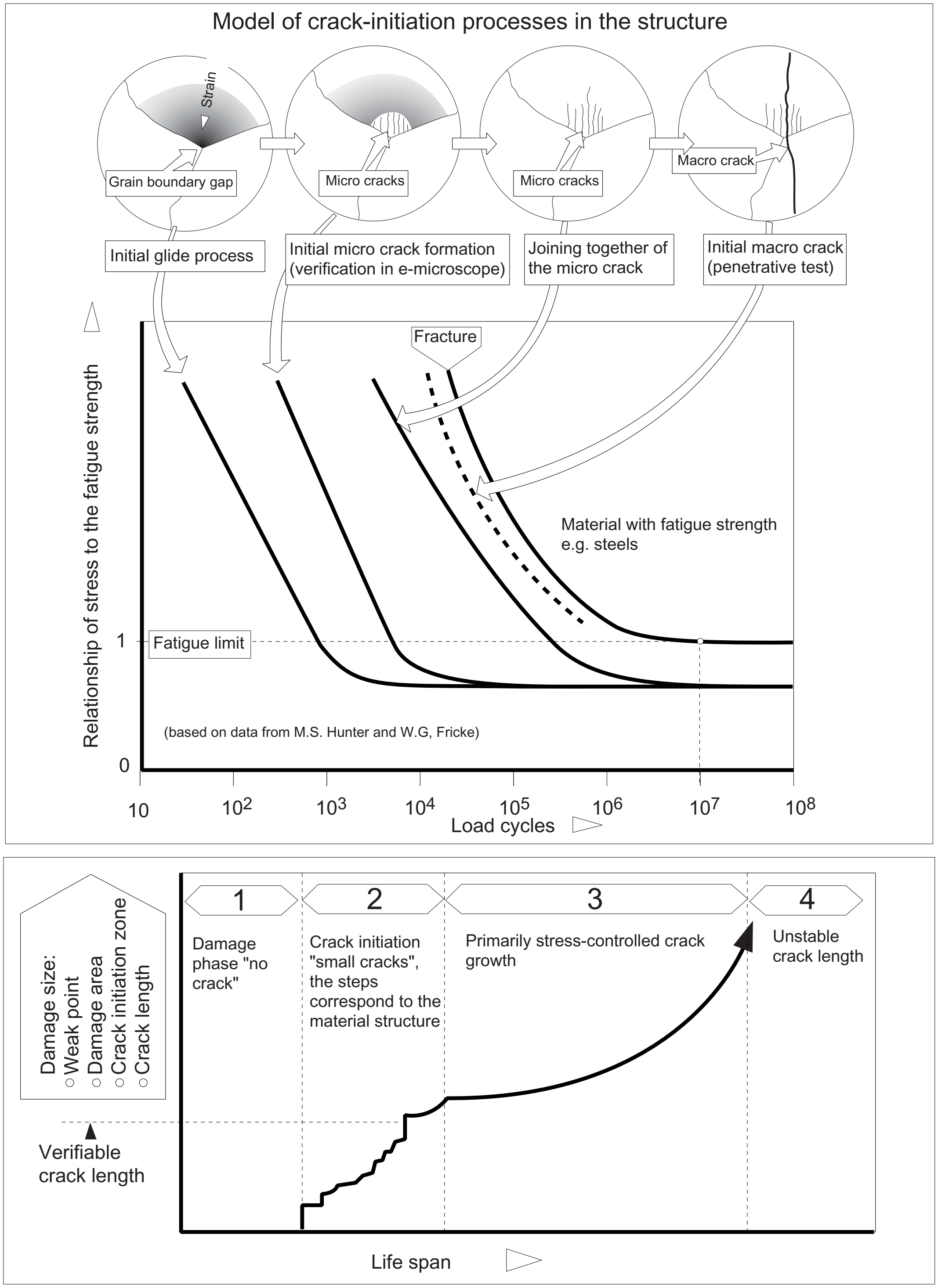
Ill. 5.4-2 *(Lit. 5.4-18, Lit 5.4.3.2.3-7): Dynamic fatigue in the* ***crack initiation phase of metals*** *can be plausibly explained with the aid of the following model (Lit. 5.4.3.2-3). Purely elastic strain of the metal lattice should be tolerated indefinitely. This means that the fatigue strength should be found in this range. However, the actual fatigue strength is considerably lower. Crystals have different elastic properties, depending on their orientation (E moduli, also see Ill. 5.4.3.2-1). A material consists of many crystals that are arranged with different lattice orientations relative to one another. If an external load acts on this structure, neighboring crystals (especially* ***grain pores****) hinder one another's elastic strain. This restriction leads to high stress levels, corresponding to the grain orientation (micro-stress). The result is local, plastically deformed zones that are strain-controlled by the surrounding grains (top left diagram). This effect appears even at relatively low macro-stresses in a hysteresis of the cyclical stress-strain curve and, therefore, at increased damping. Depending on the material, the deformation resistance increases during plastic deformations, i.e. the material becomes harder (strain hardening). This is not only true for tensile tests with large plastic strain and LCF loads, but also for HCF loads, in this case at grain pores in the micro-range and at small flaws in the notch effect range. Under vibrating loads, the hardening process occurs over a long time and a correspondingly high number of deformation cycles. Hardening and micro-stress reinforce one another in a self-energizing process called* ***reciprocal hardening*** *(also see Ill. 15.4-8). This process ends only when the entire affected zone has hardened to the point that only elastic strain occurs (stationary state).*
If the dynamic loads are sufficiently high to exceed the strength of the most highly stressed hardened zone, electron microscope-detectable cracks form and/or the structure is damaged accordingly (second diagram from left). Crack initiation leads to local resiliency and, since this process is strain-controlled, stress reduction in these areas. The notch effect of the crack tip acts against this ***self-repair*** *(self healing) process.*
If the macro-stresses are below the fatigue strength, the relaxing effect will be dominant and there will be no growth in micro-cracks above the perceptibility limit (third diagram from left). If, however, the macro-stress is greater than the fatigue strength, the notch effect of the macro-cracks will be dominant and result in crack growth and macroscopic crack initiation in the fatigue fracture (right diagram). The limiting curves of the damage/crack sizes also reveal the limits within which possible non-destructive testing methods can be used.
At the fatigue strength limit, ***cyclic hardening*** *and* ***cyclic breakup*** *just cancel each other out.*
The top graph shows the ranges of the various ***damage stages*** *(Ill. 4.3-6.1, Lit 5.4.3.2-4) in a way similar to the* ***Woehler diagram****, but with standardized dynamic loads.*

The depicted fatigue model permits the interpretation of further phenomena that can be of considerable practical importance:

"Strength training" (conditioning) for materials *(Ill. 5.4.3.2-7, Lit 5.4.3.2-3): In an* ***incremental test****, the specimen is first subjected to low dynamic stress and, after running through 10^7 load changes, the stress levels are incrementally increased several times* ***until a dynamic fatigue fracture occurs****. Specimens in this type of test, which have a sufficiently low initial stress stage, have considerably higher dynamic strength than samples that are immediately stressed above the fatigue strength. This can be explained by the fact that plastic strain and micro-cracking cause relaxation at structural weak points, mitigating their effect. Therefore, design data that are determined in this type of test* ***can be higher than the characteristic values that are actually attainable in the engine parts.*** *This is a dange-*

continued page 5.4-6

Under dynamic loads, changes occur in the material long before cracks are recognized and fracture occurs.



Ill. 5.4-2

Fortsetzung von Seite 5.4-4

rous situation because it ***increases the risk of part failure due to HCF.***

Damage due to dynamic loads: *This effect occurs when the dynamic loads are temporarily high enough to cause growth-capable micro-cracks to form below the fatigue strength. This load limit can be plotted in the Woehler diagram as a* ***damage line.***
These damages can occur, for example, due to large temporary dynamic loads on parts. Typical examples are compressor blade rubbing and loads on the blading during surges. Therefore, it must also be possible to use sufficient statistical numbers of damaged parts to verify this type of damage through dynamic tests below the fatigue strength. Experience has shown that this type of verification has often yielded ***usable results****.*

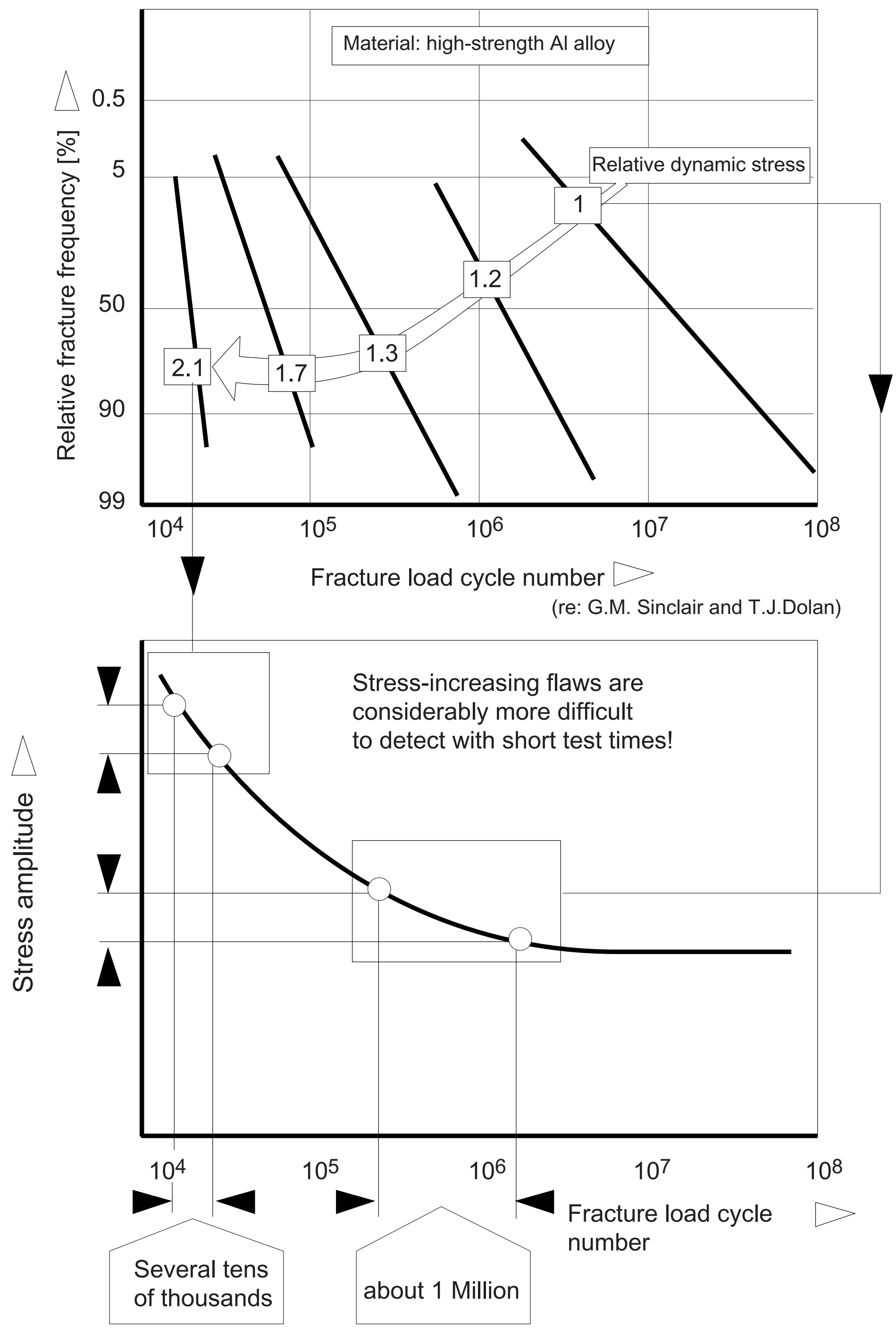
Notes: *Naturally, the described model does not satisfactorily explain the dynamic fatigue processes for cast alloys with large grains (in the centimeter range, Ill. 5.4.3.2-9), directionally-solidified alloys, or single-crystals. In these cases, cracks usually initiate at surface weak points (e.g. damaged grain boundaries) and internal flaws (e.g. pores, Ill. 5.4.3.2-8 and Ill. 5.4.3.2-9).*

The bottom diagram shows the phases of ***macro-crack initiation****, fatigue cracking to crack instability, and the eventual fracture of the part (Ill. 4.3-1). The crack size is plotted over the life span, i.e. the number of load cycles.*

Ill. 5.4-3 *(Lit. 5.4.3.2-9): The example of a high-strength Al alloy is used to show the influence of the size of dynamic loads on the* ***life span statistical spread/scatter.*** *The steeper the lines for the various loads, the smaller the scatter becomes (top diagram). One can easily recognize that the scattering is minimal at high dynamic loads (in the LCF range, bottom diagram). Loads in the range of the fatigue strength cause the scattering to increase considerably. The same is true of the* ***notch effect****, (Ill. 5.4.4-1) which decreases with increasing loads, and is therefore* ***smaller in the LCF range than in the HCF range.***
The following important conclusion must be drawn from this behavior: ***In order to investigate the influence of small flaws and weak points on dynamic strength, tests must be conducted in the HCF range.***

Accelerated/abbreviated tests *with high loads are less sensitive to notches, and are therefore not on the safe side.*

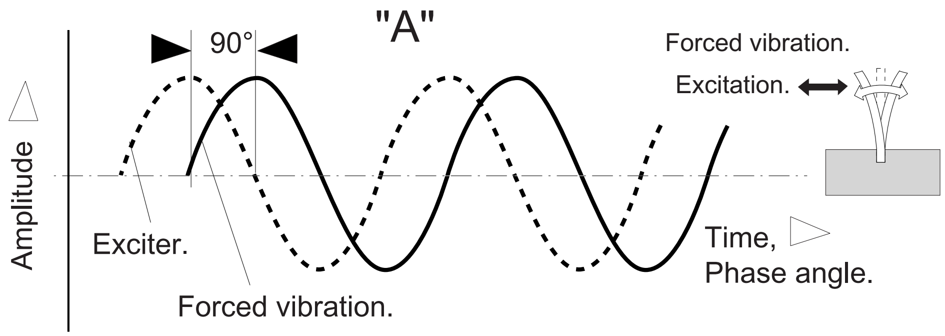
The greater the loads, the smaller the scattering of the dynamic strength, since the influence of weak points is reduced.



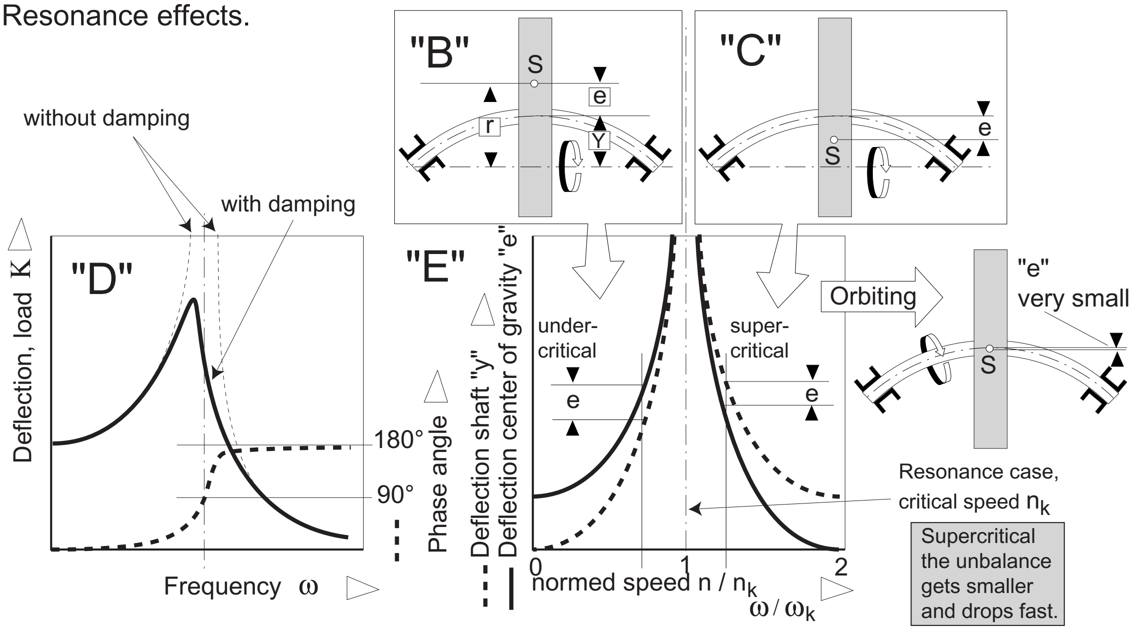
Ill. 5.4-3

Basics for the understanding of the origin of a vibration load.

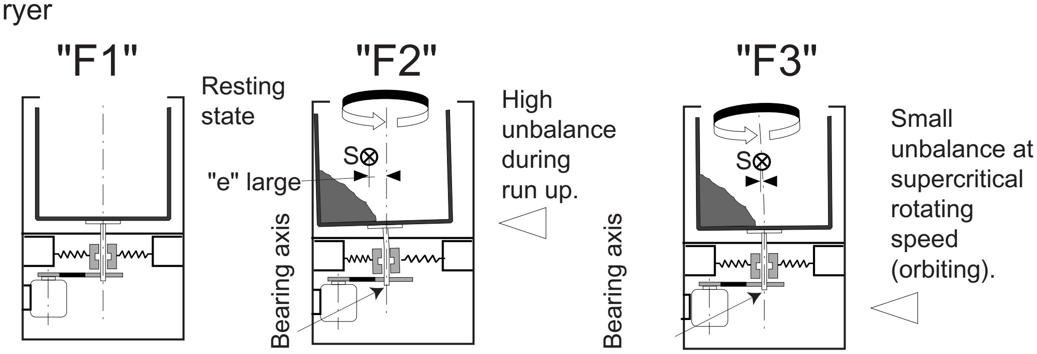
A phase shift of 90° between the exciter and forced harmonic oscillation occurs in the resonance range and provides maximum energy transfer.



Resonance effects.



Spin dryer



Ill. 5.4-4

***Ill. 5.4-4** (Lit. 5.4-2): Here the practitioner schall get a short overview about **vibration effects** in connection with problem analysis.The theoretican part can be gathered from the available extensive specialist literature.*

*The frame above shows a **harmonic sinusoidal vibration** („**A**“). In many cases however other, much more complex vibration modes are concerned.*

*It is important, that with higher- frequency self-oscillations the bending load, because of the smaller bending radii, increases in the vibration loops. So very high-frequency vibrations can even **at minimum amplitudes** (in the range of 0,1 mm, Ill. 5.4.3.1-5) trigger dynamic fatigue fractures.*

*A **free vibration** occurs after a **singular** disturbance/deflection and returns depending from the damping after a decay time back to the rest position. The vibration takes place in so called **natural modes** (eigenmodes, Ill. 5.4.3.1-4 and Ill. 5.4.3.1-6) with the associated **resonance frequencies** (natural mode).*

*We speak about a **forced vibration**, if **thereby an excitation acts from outside**. Depending from this, how near are the exciter frequency and the resonance frequency, the vibration will be intensified. Are the frequencies equal, we speak about **resonance.** In this case, depending from the drop of the **damping** of the excited system, the amplitudes of the deflection respectively of the dynamic load will increase exponential (Ill. 5.4-5.1).*

*In the frame below are indicated the relations for an **unbalanced rotating shaft** („**E**“) with a bearing at both sides (Lit. 5.4-6). Thereby the excitation conditions differ in the frequency range below the resonance (undercritical range) from those above (supercritical range).*

*In the **undercritical operation** the center of gravity from the rotation system (shaft, unbalance, persistent line) resides outside of the shaft and rotation axis („**B**“). The bending increases due to the centrifugal force and can get unrulable in the resonance range during too low damping.*

*In the **supercritical operation** the center of gravity respectively the unbalance resides between shaft and rotation axis („**C**“) and so nearer to it. With this the unbalance respectively the deflection decreases, the rum gets more smooth. We speak about „**self-centering**“. This effect is utilized at **centrifuges** like **spin-dryers** (sketches „**F1**“, „**F2**“, „**F3**“).*

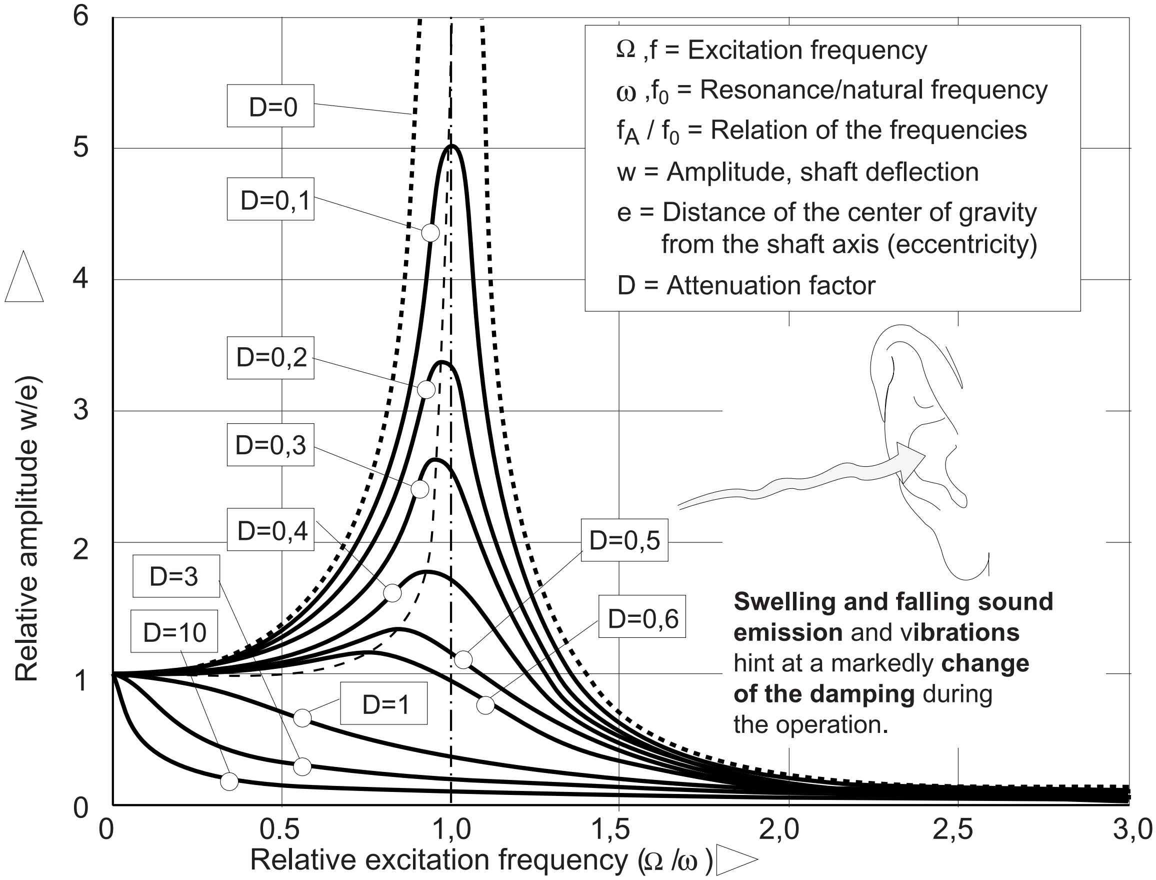
*Ar **turbo engines** we benefit from it in the case of high unbalances (e.g., blade fracture or foreigen object damages, Lit. 5.4-7).*

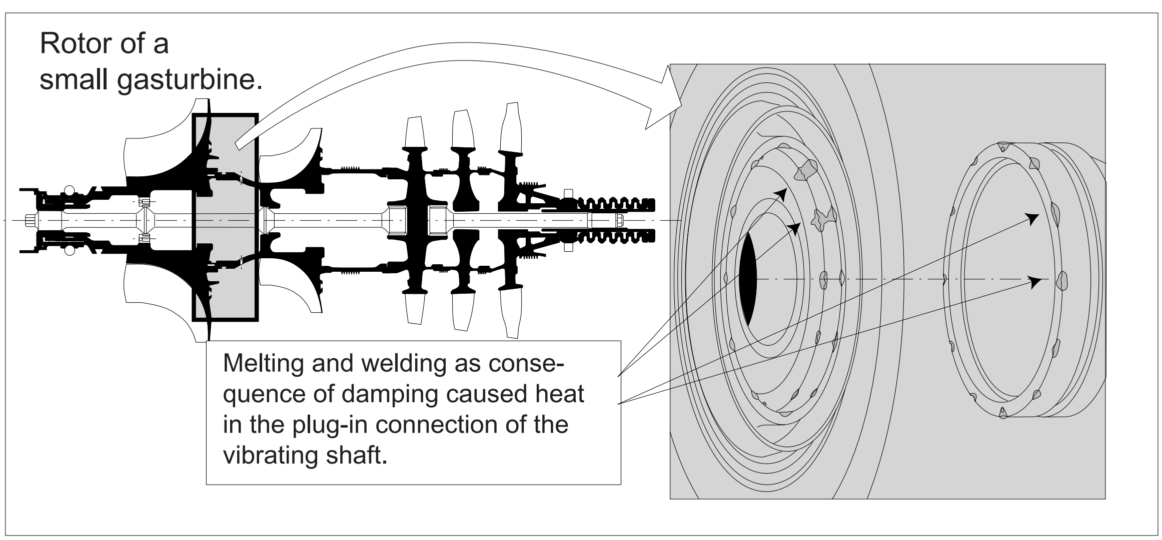
Ill 5.4-8 deals with a sketch of the kinematic changes between undercritical operation to the supercritical operation, understandable for the practitioner.

*Of special interest besides the dynamic load is the **elastic deformation of the components during vibrations**. These influence:*

- ***Sealing gaps** and with this leaks.*
- ***Rubbing processes** with*
  ***overheatings** (structure changes,*
  ***drop in strength**, formation of hot tears),*
  ***vibration excitation** of other components,*
  ***sliding wear** (size, distribution, area).*
- ***Bearing lifetimes** (contact pattern, kinematics).*
- ***Cavitation** (in the damping layer of oil dampened bearings, fuel systems),*
- ***Function of control units and valves** are disturbed.*
- ***Acoustic emission**.*
- ***Identification** through monitoring sensors.*
- ***Fretting**.*

Effect of the damping at the resonance behaviour.

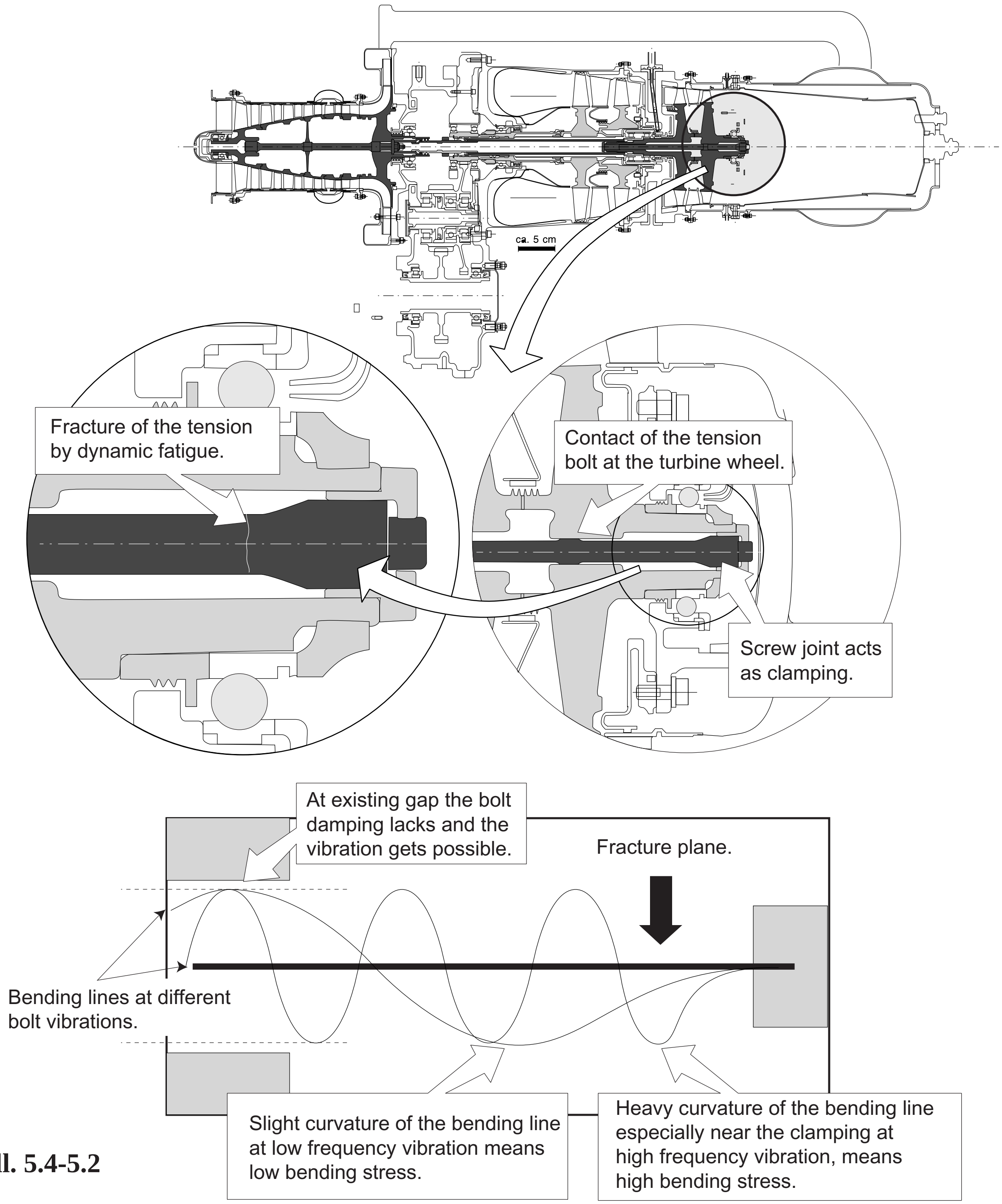




Ill. 5.4-5.1

Ill. 5.4-5.1: *The* ***damping*** *is a precondition for the control of enevitable resonances in the technique. The vibration behaviour of a system and the indication of concerned parts can enable* ***significant hints at causes and the failure development*** *of technical problems.*

The avoidance of rubbing processes can trigger failures if the damping drops unconscious.



Ill. 5.4-5.2

Already the sound emission (Lit. 5.4-8) and/or sensible vibrations can give at rotors hints at its causes. So in the displayed case below at test runs during the development markedly ***swelling and falling noise levels*** *without the change of the frequency (change of the sound) have been* *noticed. **Synchronous** to this sensible **vibrations** occurred. The engine hat to be shut down. During the following disassembly extreme **overheatings** up to melting **at the centering plug connections** of the shaft have been found. These developed through intense micro movements caused by a shaft vibration. Cause was a decrease of the pretension of the central tension bolt, triggered by thermal expansions. With this an operation of the rotor in the resonance was possible. Obviously the **damping in the plug connections** changed periodically due to the **fricttion conditions** (Ill. 5.4-6). From this the deflection ot the shaft and the energy of the sound respectively the vibration was influenced.*

***Ill. 5.4-5.2** (Lit. 5.4-7): In case of a small helicopter turbine engine (sketch above, Lit. 5.4-8) the by itself correct design principle should prevent friction spots at the high loaded hub area already during overhaul assembly of the rotor disks. The till then usual assembly lead to this, that usually **the turbine turbine sided central tension bolt with a fitting collar touches the surface of the hub bore from the wheel**. The relative movements between tension bolt and hub bore (vibrations, thermal expansions) lead to fretting spots in the hub of the turbine wheel, however without dangerous failures at these components.*
The touching of bolt and turbine wheel could be avoided with an exact centering of the bolt during the assembly. After a high number of turbine engies have been supplied, within the following years cumulated fractures occurred in the thinner waisted shank of the tension bolt, an area which suffers no fretting at all (sketch in the middle).
*Investigations allowed the conclusion, that the now no more touching **tension bolt is no more dampened by the friction process at the hub of the wheel**. So through the resonance a high frequency vibration of the bolt could be excited (probably through teeth frequencies from the gear). The amplitude at the vibration loop, necessary for the dynamic fatigue of the bolt, was during the extremely high frequencies (ultrasonic range) so little, that the relatively tiny play between bolt and wheel was no more bridged, thus no danping was effective (sketch below).*
*This case shows, how carefully a change of by itself proven engine configurations must be proceeded to avoid a **„disimprovement“**.*

***Ill. 5.4-6** (Lit 5.4-8 and Lit. 5.4-9, Lit. 5.4-19 and Lit. 5.4-20): There is a difference between **(externally excited) forced vibrations** or **resonant vibrations**, which depend on the frequency of the excitement forces that are acting on the vibrating system from the outside, and **self-excited** or **instability-vibrations**, which are independent of an external excitement or its frequency.*

***Forced vibrations or resonant vibrations:** If, in addition to restoring force and resistance, another inciting, periodically changing external force acts on a system, it is called a forced vibration, in contrast to a free vibration. If the eigenfrequency of the part and the excitement frequency are the same, resonance occurs.*
*The exciting frequency is either the rotor RPM or many times greater. A **cricial RPM** is at hand when the rotary frequency of a natural oscillation is equal to the natural frequency of the rotor. Excitement frequencies can also be several times greater than the rotor RPM (harmonic). Such high excitement frequencies originate in stators or toothed gears, for example. In the case of an externally-excited resonant vibration, as a first approximation, the critical frequency remains constant at every shaft RPM rate. Deviations from this behavior are possible if the stress in the part (e.g. centrifugal force in a rotor blade) increases with the RPM (similar to a tightened violin string) or if the temperature increases with the power output (**a drop in the E modulus** with*

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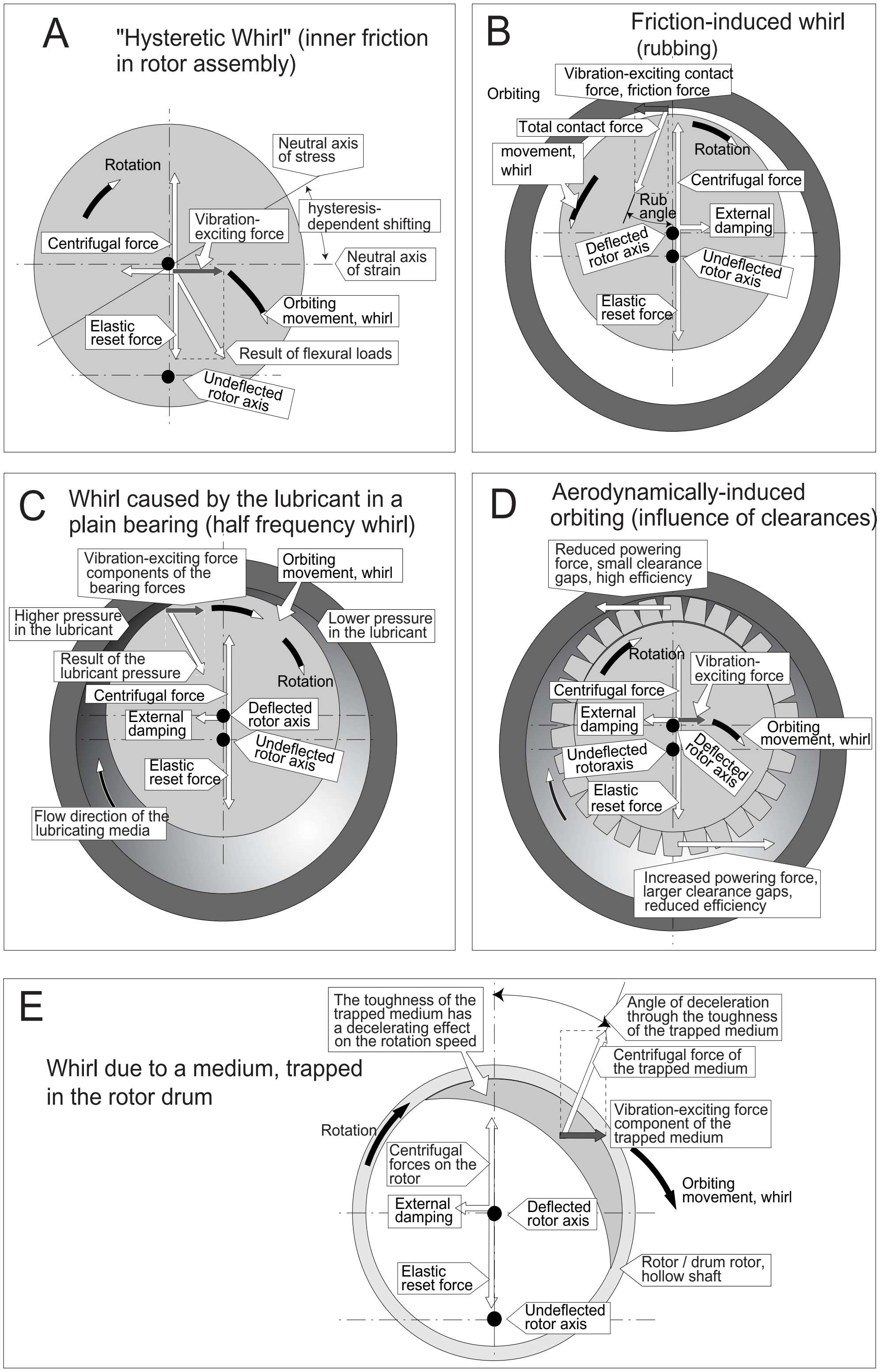
Effects/influences that lead to a self-exciting rotor vibration (shaft deflection, whirl):



Ill.5.4-6

continued from page 5.4-12
temperature leads to a drop in frequency; Ill. 5.3-1).

Orbiting by self-exciting vibrations: *These are causally related to mechanisms that orbit at a critical frequency without an external excitement.* ***In this form of rotor instability****, the characteristic, radially deflected orbiting (Ill. 12.6.3.1-13) of a shaft* ***creates a force that acts on the rotor in a direction that is tangential to the radial deflection.*** *This deflecting force grows corresponding to the deflection. This process is referred to as* ***whirling*** *or* ***whipping.*** *If the RPM level is reached, at which the externally-acting stabilizing damping is not sufficient for the forces, it causes a shaft deflection that increases more and more (****shaft flutter****). This is called a coupled self-increasing instability. The inciting RPM must not correspond with any special circumferential frequency (Ill. 12.6.3.3-11). Even if the damping shifts the frequency, it does not lead to a smaller amplitude, which would be the case in an external excitement.* ***A self-increasing vibration system is driven (reinforced) by internal friction in the rotor assembly, rubbing of the rotor against static parts, or aerodynamic effects*** *(Ill. 5.4-8). The orbiting movement can occur either in or against the direction of rotation.*

“A“ Orbiting by hysteretic whirl caused by friction in the rotor assembly: *This dynamic instability is caused by internal friction in the rotor (Lit. 5.4-8). Internal friction occurs primarily at the contact surfaces of the rotor. (Ill. 5.4-5.1) These include centering collars, flange surfaces, and fan blade clapper assemblies. Internal friction causes neutral strain and stress axes to shift and induce a tangential force that acts perpendicular to the deflection, i.e. whirl instability. The deflection thus increases the stresses, which in turn increase the deflecting force. The deflection can often be induced by a small initial impulse, such as the seating of detachable connections (e.g. centering seats). This phenomenon of a whirl instability only occurs at* ***RPM above the first critical RPM.*** *It can be prevented by avoiding plug-and-socket connections.*

“B“ Orbiting caused by rubbing: *Experience has shown that this* ***whip instability*** *occurs* ***during warm-up procedures*** *with considerable friction forces of the type that occur at blade tips and labyrinths. As soon as the rotor and rubbing surface come into contact, the friction force puts tangential loads on the rotor. The friction force is roughly proportional to the radial force (μ=0.5). This creates conditions for a dynamic instability that moves against the direction of rotation.*
Because the friction forces usually occur at specific times (periodic in/out type), the ***dynamic stiffness*** *of the system is also time-dependent (Lit. 5.4-19). The coupling effect with the* ***friction process*** *depends on the following influences:*

- Contact forces perpendicular to the friction surface, depending on the angle of infeed, rubbing speed, tribo-system, etc.
- ***Contact surface.***
*- Resiliency (****degrees of freedom****) of the rubbing elements (e.g. elastic, damped bearings)*
- ***Dynamic stiffness of the structure*** *under normal operating conditions, and the structure that is additionally coupled to it through the friction process.*
- ***Contact time*** *in relation to the contact-free time (gap development).*

“C“ Orbiting caused by the flow in in plain bearings *(lubricants)* ***and labyrinth seals*** *(leakage medium):The following concerns air bearings. For air bearings,* ***half frequency whirl****, in which the shaft is deflected with half of the RPM, is a considerable problem.*
The instability is created when the air between the shaft and bearing surface circulates with half the mean speed of the shaft surface. Due to the

toughness of the air in the very tight space, higher pressure builds up ahead of the gap than at the gap exit.This leads to a tangential force on the rotor. A whirl-movement begins when this tangential force exceeds the inner damping. Experience has shown that this effect occurs when the shaft runs at approximately twice the critical RPM. Therefore, remedies include reducing the RPM and/or suitable structuring of the bearing surface (segmented bearings, foil bearings).

"D" Aerodynamically-induced orbiting: *The mechanism of this phenomenon is evidently not yet completely understood. Aerodynamically effective rotor components such as bladed compressor and turbine stages can induce coupled shearing forces due to the motion of the disk. In principle, the acceleration or deceleration of the air flow leads to a tangential force on the blading. This is also the case if the clearance at the circumference changes.*

"E" Orbiting due to media caught in the rotor drum: *If liquid media get into the rotor (**leakage oil, cleaning fluids, condensation water**) and are not expelled by specifically provided **drainage holes**, it can result in a dynamic instability. **The fluid moves in a tangential direction at the circumference. This creates friction forces and corresponding tangential forces that act on the rotor.** This instability occurs between the first and second critical RPM (1st and 2nd harmonic).*

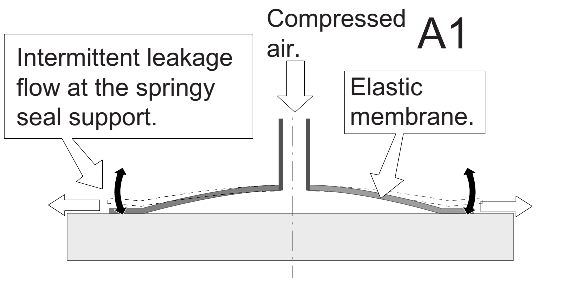
Ill. 5.4-7.1 *and* ***Ill. 5.4-7.2*** *(Lit 5.4-7, Lit. 5.4-11 and Lit. 5.4-12): The following shall deal with the mechanisms of selected **excitation effects** of high frequency vibrations by means of examples.*

*„**Hydrodynamic paradox**" („**A**"): In the region of the **labyrinth mount below the stator** of the low pressure turbine first stage (sketch above right) from a bigger shaft aeroengine, after a constructive change, several failures as dynamic fatigue fractures in the HCF range occurred. Concerned have been dynamic fatigue cracks in circumferential direction (sketch below right) at the front wall of the ring duct which is formed by the seal mount (detail above right).*
*The design change in connection with the failures was a wire sealing ring between stator and the labyrinth mount. The sealing effect against a leakage air flow from the space with the higher pressure level before the wall to the ring duct in the seal mount was **only guaranteed by the spring force** of the disk like frontwall from the labyrinth mount („**A2**"). After some hundred operation hours the seal ring and the contact surfaces of the sealing groove in the labyrinth mount showed heavy fretting wear (chapter 5.9.3).*
*As failure cause a vibration excitation of the springy seal wall at sufficient progressed fretting wear was suspected. This excitation principle (**„hydrodynamic paradox"**, Lit 5.4-11) is based on the springy effect from the cover and the pressure drop inside the leakage flow (Bernoulli) and is presumed as exemplary physical experiment (sketch **A1**).*

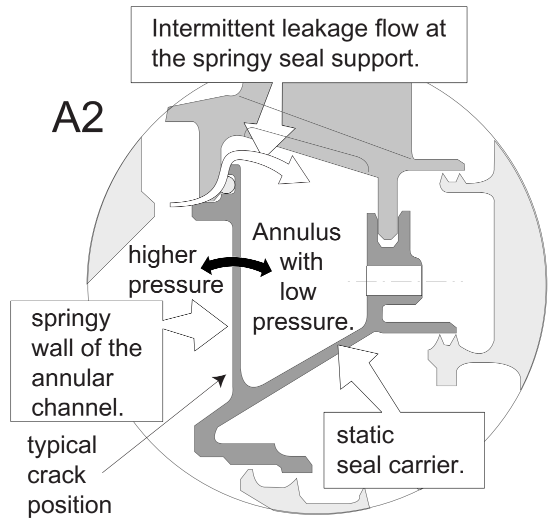
continued page 5.4-18

Possibilities of the excitation of high frequency vibrations.

A Excitation by the principle of the "hydrodynamic Paradox".



If the seal has contact, pressure builds up under the membrane till the seal lifts. Then at the sealing gap in the escaping air stream the pressure drops and the gap closes again and so on.



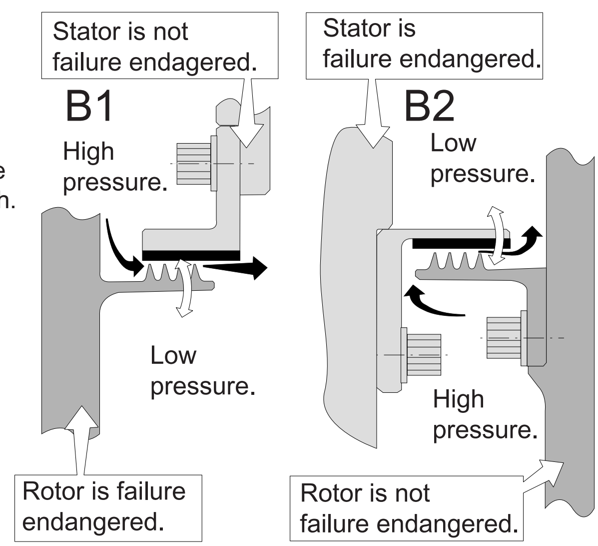
B Vibration excitation due to leakage flow in labyrinth seals.

The direction of the leakage flow determines the vibration endangered component of the labyrinth.

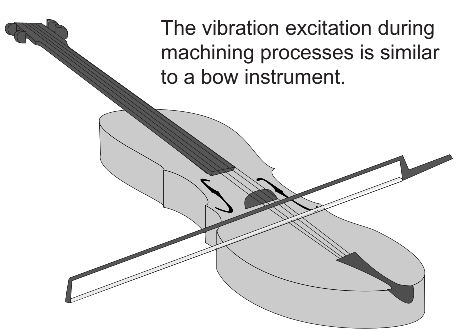
! The rotating part of the labyrinth is endangered when:

- The mount of the labyrinth is at the side of the higher pressure.

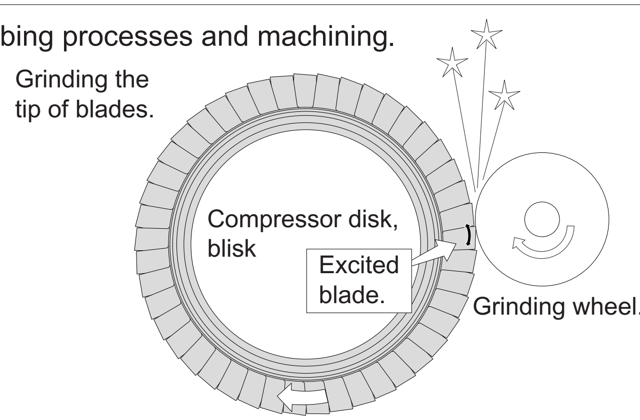
- The labyrinth is integral connected with the rotor (lacking damping).



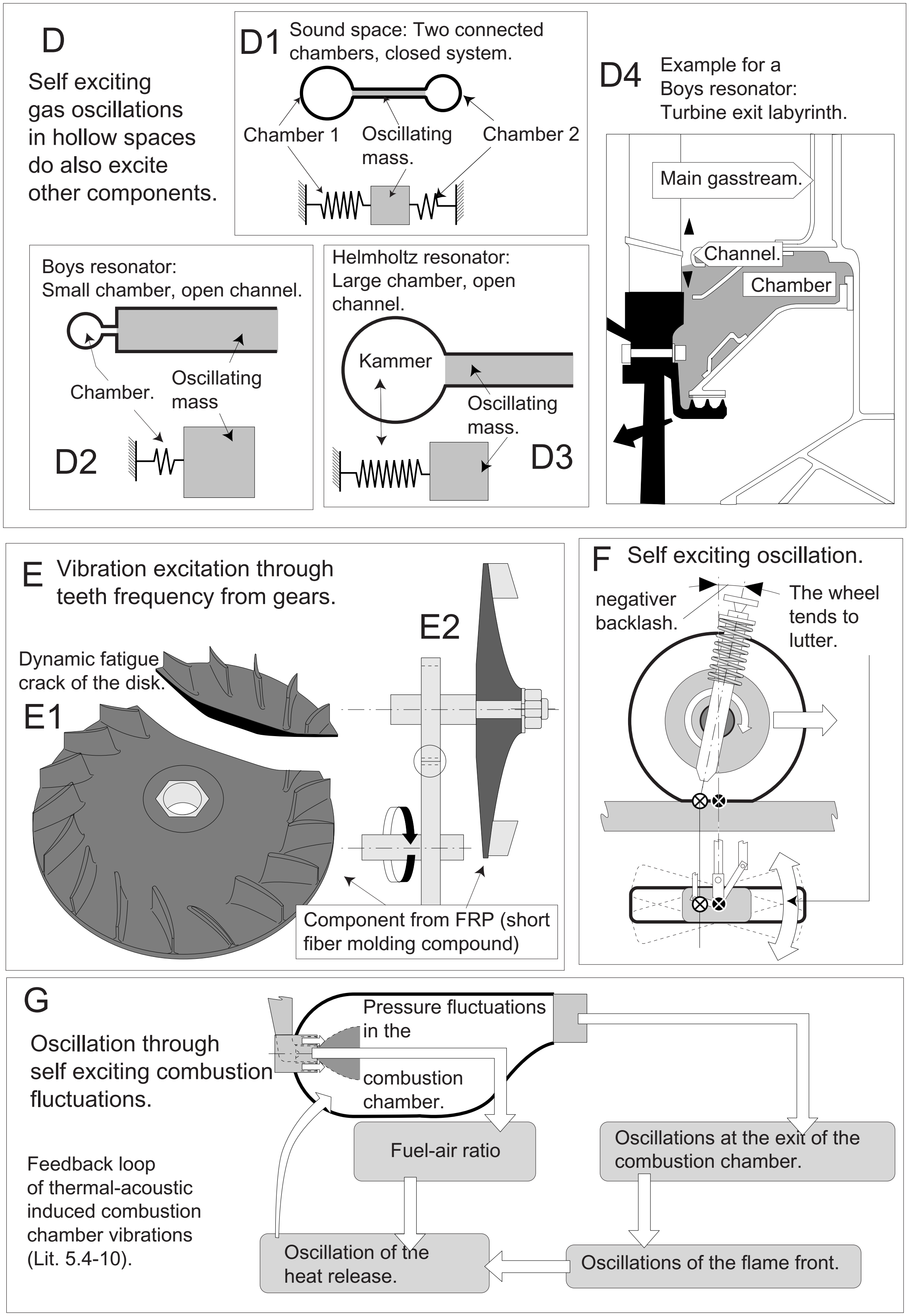
C Vibration excitation during rubbing processes and machining.



The vibration excitation during machining processes is similar to a bow instrument.



Ill. 5.4-7.1


D
Self exciting gas oscillations in hollow spaces do also excite other components.
D1
Sound space: Two connected chambers, closed system.
Chamber 1
Oscillating mass.
Chamber 2
Boys resonator: Small chamber, open channel.
Chamber.
Oscillating mass
D2
Helmholtz resonator: Large chamber, open channel.
Kammer
Oscillating mass.
D3
D4
Example for a Boys resonator: Turbine exit labyrinth.
Main gasstream.
Channel.
Chamber
E
Vibration excitation through teeth frequency from gears.
Dynamic fatigue crack of the disk.
E1
E2
Component from FRP (short fiber molding compound)
F
Self exciting oscillation.
negativer backlash.
The wheel tends to lutter.
G
Oscillation through self exciting combustion fluctuations.
Pressure fluctuations in the combustion chamber.
Fuel-air ratio
Oscillations at the exit of the combustion chamber.
Feedback loop of thermal-acoustic induced combustion chamber vibrations (Lit. 5.4-10).
Oscillation of the heat release.
Oscillations of the flame front.

Il. 5.4-7.2

continued from page 5.4-15

The confirmation of the failure hypothesis could be induced with the help of a simple demonstration test. A stator with the sealing wire ring was assembled and operation near pressure conditions simulated at the component. Thereby, exactly as forecasted, with sufficient wear and therefore decreasing spring force an intense vibration excitation of the cover plate with correspondent pulsating leakage flow could be observed.
After the failure mechanism was known and could be reproduced at original components the requirement for a ensured remedy was given.

***Vibration excitation through leakage flow** („**B**“, Lit 5.4-13): The **susceptibility of a labyrinth seal for an aeroelastic vibration excitation** through the leakege flow primarily depends from the change of the gap during the vibration processn and with this from the position of the circumferential nodal line. The phase position of the gap change to the phase of the pressure course in the labyrinth can excite or damp a vibration, depending from the position of the line of nodal line.*
*Together with the **natural frequency/resonance frequency of the seal part** applies („**B1**“, and „**B2**“):*
*A seal ring, which is mounted at the labyrinth side with the lower pressure level, will not be excited to **aeroelastic vibrations**, if his lowest natural frequency is below the acoustik frequency. If the sealing ring is mounted at the high pressure side, it can not be excited if its lowest natural frequency lays below the acoustic frequency.*

*Generally applies the **design rule**: Seals which are mounted at the pressure side must be dampened.*

*For this tey usually are provided with **mechanical dampers** (Ill. 5.4.3.3-8).*

***Vibration excitation through production, friction and rubbing processes** (Lit.5.4-7 and Lit. 5.4-12): Usually the excitation is due to the so called „**stick-slip-effekt**“ (Ill. 5.9.1-8). As example for such an excitation a string instrument can serve. Also in many operation conditions of machine elements such excitations can be observed. To these belongs the **squealing** of brakes. Such squealing sounds are characteristic for high frequency vibrations. They can deteriorate the component in a short period of time. This can already occur identifiable before crack formation or fracture (Ill. 5.4-2).*

***Selfexciting gas vibrations in hollow spaces** („**D**“): An aircolumn can act in different arrangements, comparable a spring in a mass-spring system, and excite this system to resonance vibrations. The larger the space volume and the diameter of the connection duct, the lower is the spring stiffness of the air column. Primarily the vibrating mass is limited at the connecting duct. This means the more voluminous the air column, the larger the vibrating mass. As well in a ring chamber between disk and labyrinth as also in the main gas stream different pressure waves can develop and intensify each other.*
*A **„tonal space“** („**D1**“) consists of two small closed chambers, which are connected by an air column. The air filled chambers act like springs. Such configurations can arise in aero engines in the region of the cooling air carrying ducts/lines in breather pipes and for the pressure balance.*
*The **„Helmholtz resonator“** („**D3**“) consists of a relatively large chamber which, with an air column as spring, is open connected with the mass of the opposite side. Such configurations arise in the main gas stream of aeroenines at the strut fairings of the casings.*
*Exists a small chamber, i.e. a high spring stiffness and a large air column volume, which are connected by a narrow duct, we speak about a **„Boys resonator“** („**D2**“, Lit 5.4-14). This configuration corresponds ring ducts at inter-*

*stage seals or end labyrinths of the compressors and turbines („**D4**“). The relatively small ring duct/chamber is connected by a narrow ring gap with the main air stream as vibrating air column.*

***Vibration excitation by teeth frequency from gears („E“)**: After frequently dynamic fatigue fractures occurred at blades of an **oil cooler wheel** from a high strength Al wrought alloy (about 12 cm diameter, about 20 000 U/min), wheels made from **fiber reinforced plastic** („**E1**“) because of the high inner damping have been introduced. These wheels from short fiber molding compound (glass fiber/epoxid) run long time without complaints. However in one case a wheel suffered a dynamic fatigue fracture (sketch above left) inspite of the high inner damping of the plastic. The wheel was replaced by a similar new part. Also this wheel fractured in similar manner within about 100 operation hours. With this a failure causing fault in the wheel was unlikely.*
*The failure analysis revealed, that a gear wheel in the **drive chain** („**E2**“) was damaged probably during the assembly. Concerned was a hardly noticeable **deformation of the edge** at the tooth flank. Obviously this 'little' tooth damge lead to the extremely vibration excitation. After the gear wheel had been exchanged, no fatigue cracks at the oil cooler wheel occurred, even after long operation times. This example demonstrates how dangerous vibratiin exitations by teeth frequencies can be.*

***Self exciting vibration, „flutter“** („**F**“): Flutter can be excited at different systems. In interplay with a partly or totally separated flow and whirl formation ('Kármán' vortex street, Ill. 3-9.3) **dangerous vibrations** can build up. To such objects flown around also belong **airplane wings**, slender profiles like **blades** of turbo machines, **bridge decks** (Ill. 3-9.2) as well as **chimneys** (Ill. 3-9.3). **Also mechanical flutter can be excited**. **Wheels** tend to flutter at negative backlash (Ill. 3-9.1). Then the **supporting force of the wheel in driving direction** is **before the penetration point of the swivel axis**. So an elastic deviation of the wheel leads to a moment which deflects the wheel farther from the driving direction. Is the elastic retraction sufficient increased, it forces the wheel to swing to the other side and the process repeats. If the **damping** of the steering and the hinges of vehicles is too weak and/or have too much play, this process can build up dangerously. In cases of flutter the excitating parameters (e.g., speed) **must be at once brought below a design specific value**.*

***Self excitating combustion oscillations** („**G**“, Lit 5.4-15): For combustion processes low frequency **pressure oszillations** in the range of 50-120 Hz („**rumble**“) are typical.*
The many influencing parameters at the thermal acoustic oscillation behaviour of a combustion chamber aggravate the forecast of unacceptable vibrations. These influences are of different nature, like the flow resistance respectively acoustic resistance of the turbine stator, the damping effect of the perforated combustion chamber wall or the sensivity of the fuel supply and admixture for the pressure fluctuations. The turbine reflects the sound waves similar to a solid wall and the walls of the combustion chamber act as damper. The flame „answers“ at the pressure oscillation of the combustion chamber with a heat release fluctuation. The intensity of this 'flame answer' is based at acoustic properties of the flame. The heat release fluctuations gets more intense for more compact combustion chambers, i.e. higher energy density. Because smaller combustion chamber walls have a lower damping, the proneness for instability increases. From this can be concluded, that just combustion chambers of aeroengines with its typical high power concentration let expect especially problems during a development to less emissions. The following constituitive criteria of a combustion chamber can reduce the proneness to combustion chamber oscillations:
- Cange of the „flame answer“.

- Increase of the damping of the combustion chamber walls.

- Continuous adjustment of the fuel rate during the combustion process by means of a control loop. For this are necessary suitable pressure sensors and a sufficient fast control of the fuel supply.

***Ill. 5.4-8** :From a **failure mode** with a little luck **conclusions at the dynamic operation conditions** of a shaft can be drawn. Thereby it's about features like position and formation of **dynamic fatigue cracks, tracks** and **deteriorations** at the bearing races and position and extent of **rubbing marks**.*

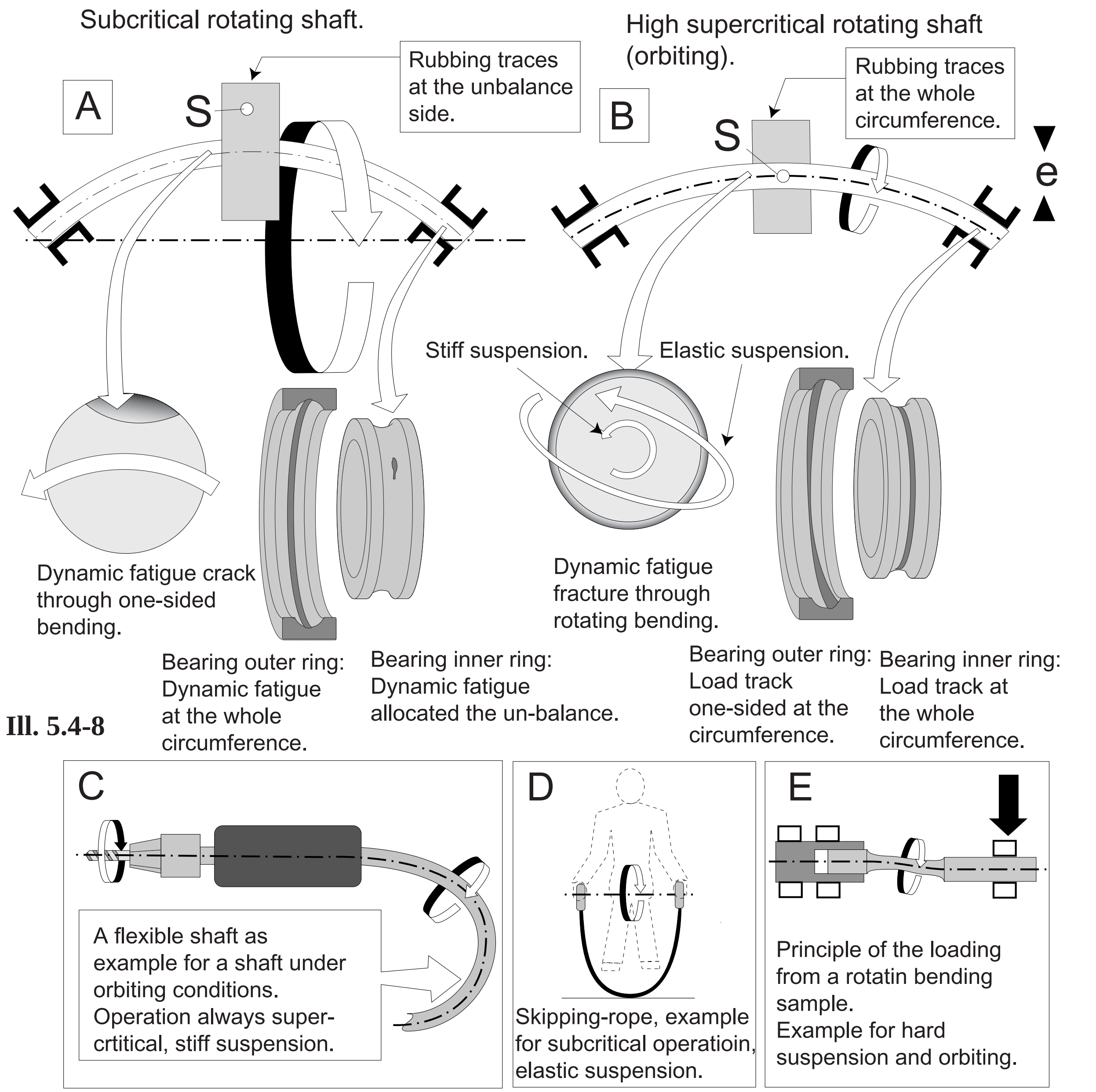
*The **elastic deformation** of the shaft/axis from a rotor determines its dynamic load. This is also true for the **bearing** with **supporting structure**. Thereby it is of importance, if the operation condition is subcritical or supercritical (Ill. 5.4-4).*

*At **subcritical operation** the bending rotates between the bearings („**A**“) through an unbalance around the rotational axis. The center of gravity lays at the outer side of the shaft (Ill. 5.4-4). This can be compared with a skipping-rope (“**D**“). Under these conditions the shaft experiences a static bending load with tensile stresses at the outer side. Will this condition be frequently enough approached, can develop here **one sided dynamic bending fatigue fracture**. (Ill. 4.4-6). Traces of **rubbing processes** through **bridging of the gap** can be found findet **only or intensified at the „unbalance side“**, but not even at he whole circumference. **Bearings** show more intense **tracks** up to dynamic overload failures at the race (**fatigue pittings**) of the rotating ring also limited at the unbalance region. However the race of the standing ring is loaded by the rotating unbalance at the whole circumference. Corresponding even is the race track or the overload effects at the circumference.*

*At the **supercritical operation** the center of gravity of the system lays between shaft and rotating axis. In the extreme case (rotation speed is far in the supercritical range) its position is at the rotation axis within the shaft („**B**“). This condition illustrates a flexible shaft („**C**“). The operation features are:*

*- Dynamic fatigue fracture/crack through **rotating bending** (Ill. 4.4-6) **proceeds widely concentric to the crircumference**.*

The running conditions of a rotating shaft can show in position and appearance of cracks and fractures of the shaft as well as load tracks/dynamic fatigue failures of the bearings.

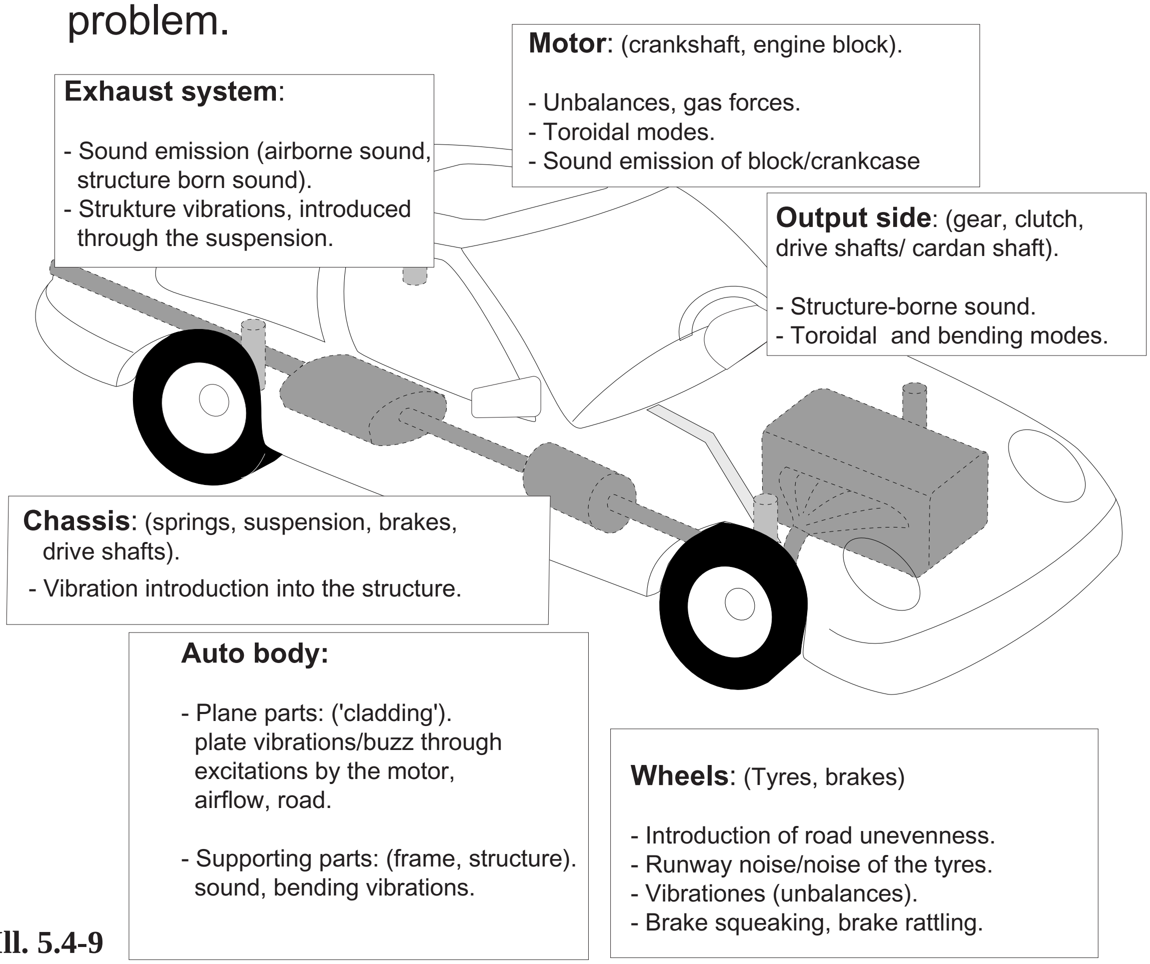


Ill. 5.4-8

- ***Race track*** *respectively fatigue breakouts at the race of the standing* ***outer ring*** *corresponding a* ***skew oriented shaft****. At the rotating inner ring widely* ***even at the circumference****.*
- ***Rubbing traces at the whole rotor circumference****.*

An important influence has the ***stiffness of the bearings****. This determines the elastic deformation and with this the dynamic load of the shaft (Ill. 5.4-4). The more stiff the bearings, the higher the load of the shaft. This effect is utilized with stiff bearings of testing machines for rotating bending („****E****“).*

The motor vehicle as exemplary vibration system and problem.



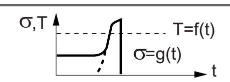
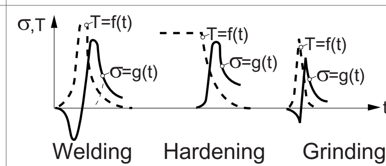
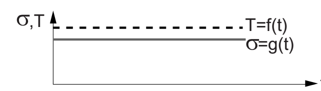
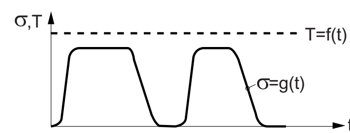
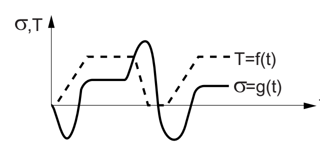
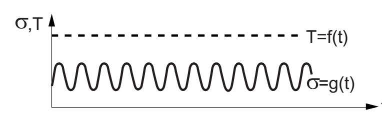
Ill. 5.4-9

***Ill. 5.4-9** : In a **motor vehicle** we find all types of dynamic loadings, low frequent and high frequent. So it is well suitable for the significance of dynamic loading in the technique.*
*The trend of a weight minimization during increase of the performance confront the designer just with regard at dynamic properties like noise, vibrations, driving dynamics and comfort with growing problems. Thereby the load of the components in view of costs, safetyand lifetime must be guaranteed. Frequently a **combination from highfrequency and lowfrequency loads** exists. Therefore in the following examples (no claim for completeness) appear components of both load types.*

*Components with **high frequency loads**:*
- *Car body (vibrations).*
- *Chassis (influence of the road).*
- *Drive side (unbalances, torques).*
- *Motor (e.g., valves, springs, crankshaft, connection rod).*
- *Exhaust system (vibrations).*
- *Anti friction bearings (wheels, drive system).*

*Components with **low frequency loads.** Here we deal primarily with thremal fatigue, i.e. cyclic thermal stresses (Ill. 5.4-10 and chapter 5.4.2.1):*
- *Exhaust system.*
- *Motor (valves, pistons).*
- *Clutch.*
- ***Brake disks.***

Typical examples from the field of engine construction for damages due to mechanical-thermal loads:

Load type.	Failure/failure sequence.	Examples	Temperature and stress progresson.
1. Short-time loads. 1.1 At constant temperature. 1.2 With sudden temperature change ("thermal shock").	- Plastic deformation - Crack initiation - Fracture	- "Runaway" turbines - Explosions, containment case - Foreign object damage.	
		- Crack initiation during welding. - Crack initiation during quenching. - Formation of grinding cracks. - Braking, rubbing process. - Bearing running hot. - Spark erosion.	
2. Creep loads.	- Creep deformation - Creep pore formation - Crack initiation - Fracture	Constant operation - Turbine rotor blades - Turbine stator blades - Rotor disk (turbine, HP compressor) - Housings (combustor, turbine)	
3. Loads in the LCF range ($< 10^4$LW). 3.1 Under the influence of mechanical forces.	- Damage (structure) - Creep deformation - Creep pore formation - Crack initiation with cyclical crack growth - Fracture	Cyclic operation - Turbine blades - Rotors of turbines - Pressure vessels (boilers), tubes. - Bolts and tie rods.	
3.2 Due to thermal stresses ("thermal fatigue").		- Turbine stator blades - Casings/housings. - Exhaust gas systems/ exhaust pipe.	
4. Loads in the HCF range ($>>10^4$ LW).	- Damage - Crack initiation with cyclical crack growth - Fracture	- Blade of a turbo engine. - Shafts and axis. - Casings/housings, boilers, pipes. - Dynamically stressed bolts and tie rods. - Springs.	

Ill. 5.4-10

Also components which through ***changes of rotation speed/centrifugal force*** *gain importance are markedly LCF loaded (chapter 5.4.1).*

- ***Flywheel****. This is especially true for systems with* ***intermittent engine cut-off*** *during stop. To these belong also:*

- ***Starter generators*** *at the crankshaft.*

- ***Kinetic energy storages*** *(Ill. 5.2.2-0).*

- High speed ***electric motors*** *for the traction drive.*

Ill. 5.4-10: *This table provides an* ***overview of the mechanical-thermal loads with the aid of typical examples*** *(Lit. 5.4-8). The examples are certain part-specific phases of operation. When one considers the entire load spectrum of an engine part, it is almost always the temporal consequence and/or combination of several load types.*

Short time loading:

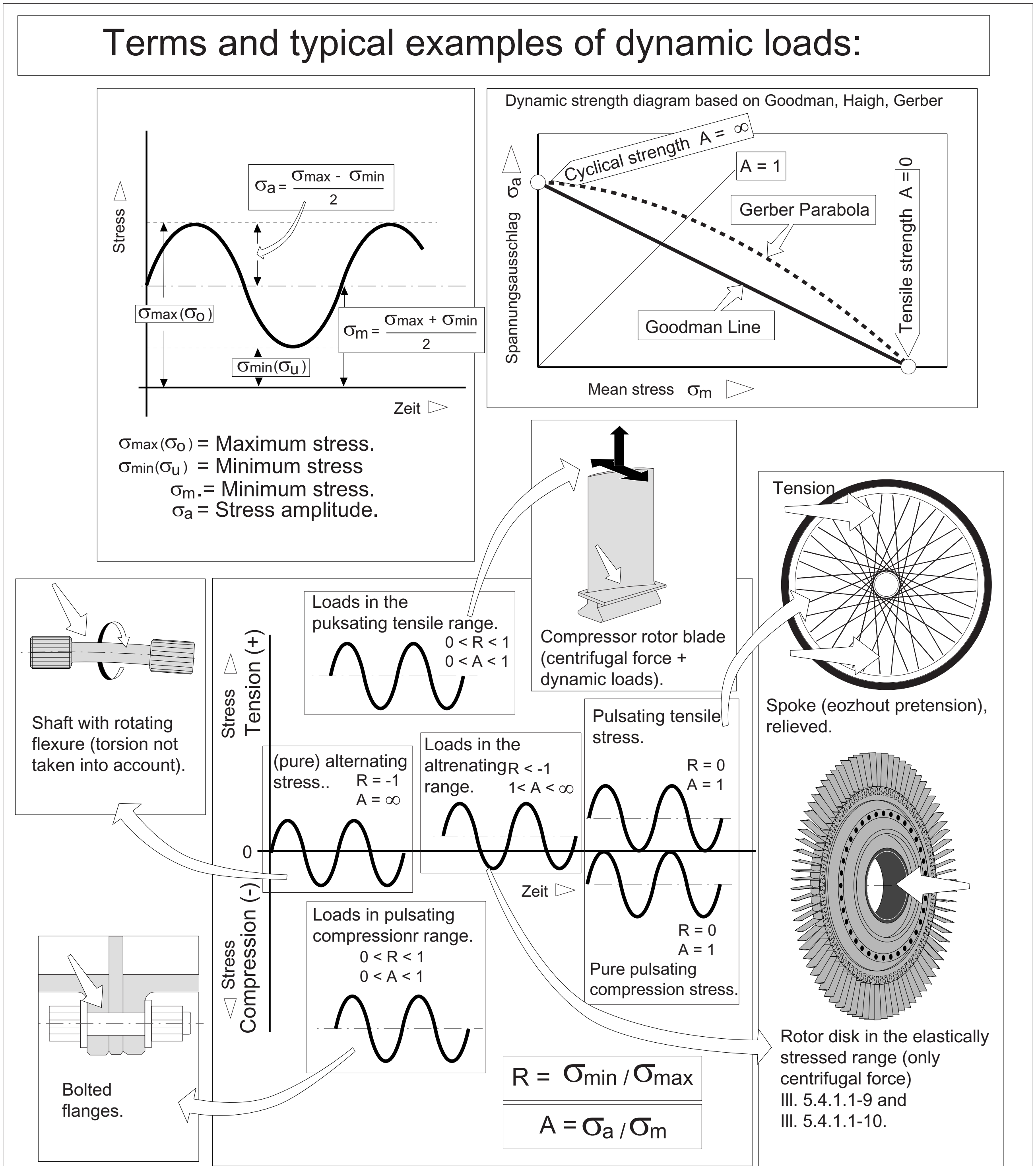
At ***constant temperature*** *means, that no causative thermal stresses act. Usually* ***forced***

fractures caused by a failure *are concerned. This can occur at* ***extreme overload*** *of an undamaged component or as* ***residual forced fracture*** *from a crack (see chapterl 4.4 and chapter 5.2.1).*
During a ***fast temperature change*** *(„thermal shock“, Ill. 5.4.2.1-3). Such loads are known from* ***overheating of brakes****,* ***crankshafts*** *and* ***clutchs****. Also in the* ***procuction*** *or during* ***overhaul*** *this failure type occurs. Examples are grinding cracks and welding cracks or crack formation through* ***electric discharge machining*** *(EDM).*

Creap load *(Ill. 5.3.2-3) is static. It occurs in the walls of pressure vessels, tubes/pipes and in bolts/ tie rods. Also rotors underlie this load during constant rotor speed.*

Load in the LCF range *(Ill. 5.4-1 and chapter 5.4.1) can be exspecially found in dimensioning conform, limited lifetime. Concerned are high dynamic loads in the plastic range which* ***have not quite the features of a vibration/oscillation*** *(periode).*
Mechanical force effect: *Usually in connection with power changes and load cycles which act at operation pressures, temperatures and rotation speeds.*
Thermal stresses *(thermal fatigue, chapter 5.4.2.1): It concerns thermal streesses caused by temperature gradients. These can be frequently traced back at temperature changes of a medium which flows along a part (gas/steam, liquid)n. Also at seemingly* ***high frequency temperature changes*** *like at* ***exhaust valves of piston motors*** *LCF occurs in form of thermal fatigue. However really concerned are relatively lowfrequency temperature changes. Here the cycles are determined by* ***start-stops*** *(main cycles) and significant* ***power changes*** *(minicycles). The temperature of the valve does not follow dimensioning relevant the gas temperature during the particular exhaust process.*

Load in the HCF range *(Ill. 5.4-1, chapter 5.4.3.1): Usually concerned are* ***forced, high frequency vibrations*** *with cycles to failure above 10^5. These are especially dangerous in the resonance (Ill. 5.4-4 and Ill. 5.4-5.1) and must be therefore avoided during the dimensioning. But also dangerous high dynamic loads can occur far from a resonance respectively natural mode (Ill. 5.4.3.1-4 and Ill. 5.4.3.1-6). For example cracks/fractures at motor valves caused by the forces during opening and shut.*



Ill. 5.4-11

Ill. 5.4-11: *In order to understand specifications for* ***material characteristic data****, especially for dynamic loads (e.g.* ***Haigh diagram****), it is necessary to know the definition of* ***temporal stress progression*** *(Ref. 12.6.1-16). The maximum stress ($\sigma_o = \sigma_{max}$) is the highest amplitude of a sinusoidal load (left diagram). In the same manner, the minimum stress ($\sigma_u = \sigma_{min}$) is the lowest load. Half of the difference between the maximum and minimum stress is the stress*

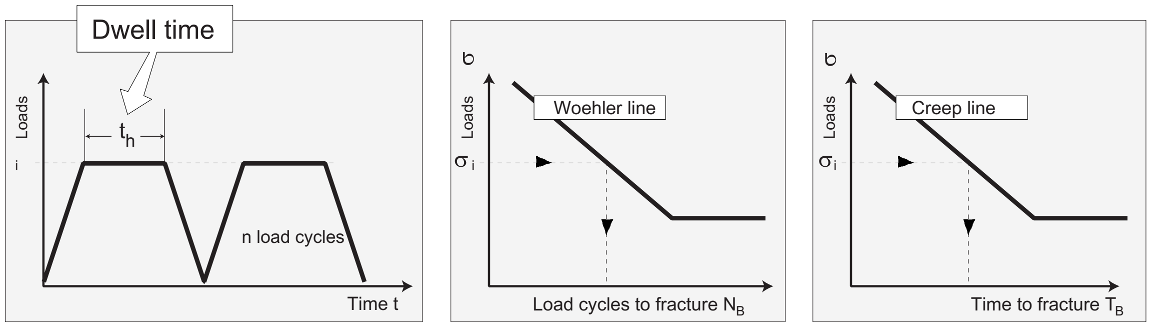
amplitude σ_a. The stress amplitude is around the mean stress (σ_m).
Depending on the height of the mean stress, certain load ranges are defined and can be identified by the stress relationship $R=\kappa = \sigma_{min} / \sigma_{max}$. In this formula, σ_{min} corresponds to the sum of the smallest value, and σ_{max} corresponds to the sum of the greatest value. The results are as follows:

R= -1 pure alternating stress,
R< -1 alternating range,
R<+1 pulsating stress range,
R=0 pure pulsating stress

*The top right diagram shows dynamic strength limits as they are often used in English-language materials (acc. Haigh and Gerber, Lit. 5.4-17). The dynamic strength is plotted as the tolerable stress amplitude σ_a relative to the mean stress σ_m. Dynamic loads below the **Gerber parabola** or **Goodman line** can be tolerated without damage. At the bottom, a typical example is provided for each of the different dynamic loads.*

***Ill. 5.4-12** (Lit.5.4-2): The easiest possibility for estimating the total damage from creep stress and dynamic loads is a **linear superposition of the damages.** The estimation is analogous to the **Miner rule** for a linear damage accumulation of multi-stage stresses (Ill. 5.4.3.2-12). It can technically be applied to all types of low-cycle problems.*
- Any order and size of load peaks.
- Any shape for the load/time progress.
- Free of contradiction from pure dynamic damage to pure creep loads.
*It is especially important to ensure that the **specimens** used are representative of the material conditions in the part (Ill. 5.3.2-11, Ill. 5.4.2.1-7and Ill. 5.4.2.1-8).*
*The various uses do not mean that the results always closely reflect reality. **Dynamic damages***

Even after longer dwell times the damage accumulation can be estimated, but the results should be viewed skeptically.



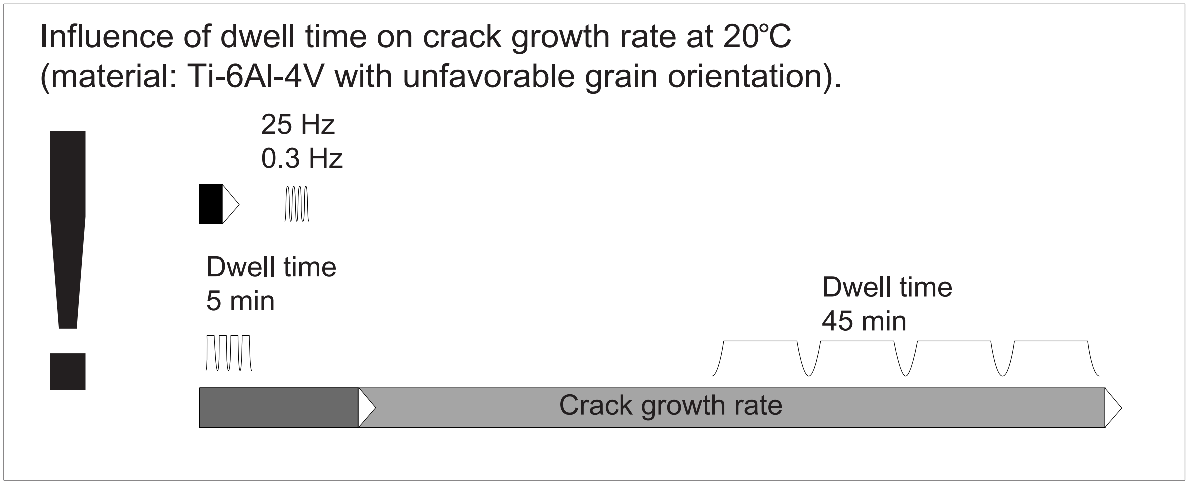
Ill. 5.4-12

$$D = \sum \left[\frac{n}{N_B} + \frac{n \cdot t_h}{T_B}\right]$$ Fracture occurs at D=1

and creep damages must occur *simultaneously (left diagram). Centrifugal force-induced loads in rotor parts during the startup/shutdown cycle fulfill this demand fairly well. Dwell times during relief must be considered if they contribute to recovery processes. If thermal stress occurs due to the temperature changes, then the conditions for such a simple estimation are generally not met. The thermal stresses are usually phase-shifted relative to the stresses induced by centrifugal force. Therefore, the temperatures change under loads. In the case of thermal fatigue,* ***relaxation processes*** *cause the maximum stress to change (chapter 4.5.1.1). If the maximum stresses are reached at very different temperatures, it can incite different creep mechanisms (Ill. 5.3.2-6 and Ill. 5.3.2-8). This would make the* ***linear superposition*** *completely unusable.*
Another method of damage calculation is ***strain range partitioning*** *(see also Ill. 5.4.3.2-12), which takes into account plastic flow and creep (and relaxation).*

Ill: 5.4-13 *(Lit.5.4-8, Lit.5.4.1-21 and Lit. 5.4.1-22): LCF tests of* ***specimens made from the titanium alloy*** *Ti-6Al-4V yielded important realizations that make spectacular damages in large rotor parts more understandable.There is a material-specific sensitivity to compressive stress (e.g. Ti-6Al-4V, Waspaloy, etc.) or* ***tensile stress*** *(e.g. CrNi18/8 steels, IN100) during the dwell time. A dwell time sensitivity, in which fatigue and creep processes occur, was discovered in many materials.*
The ***crack growth rate*** *in Ti-6Al-4V can be* ***several times greater with dwell times*** *(test data: 5 minutes and 45 minutes) at the stress maximum than under dynamic loads without dwell times (test frequencies: 0.3-25 Hz). Interestingly, Ti-6Al-4V is sensitive to compressive stress. In this case, the compressive stresses induced during LCF loading have an especially damaging effect during the "rest phase".*
This process is not fully understood.. Obviously it stands also in connection with the ***diffusion of***

Be careful with life span verifications using cyclical running tests. The dwell time at high loads can have a considerable effect on LCF behavior, and must therefore be maintained if in doubt.

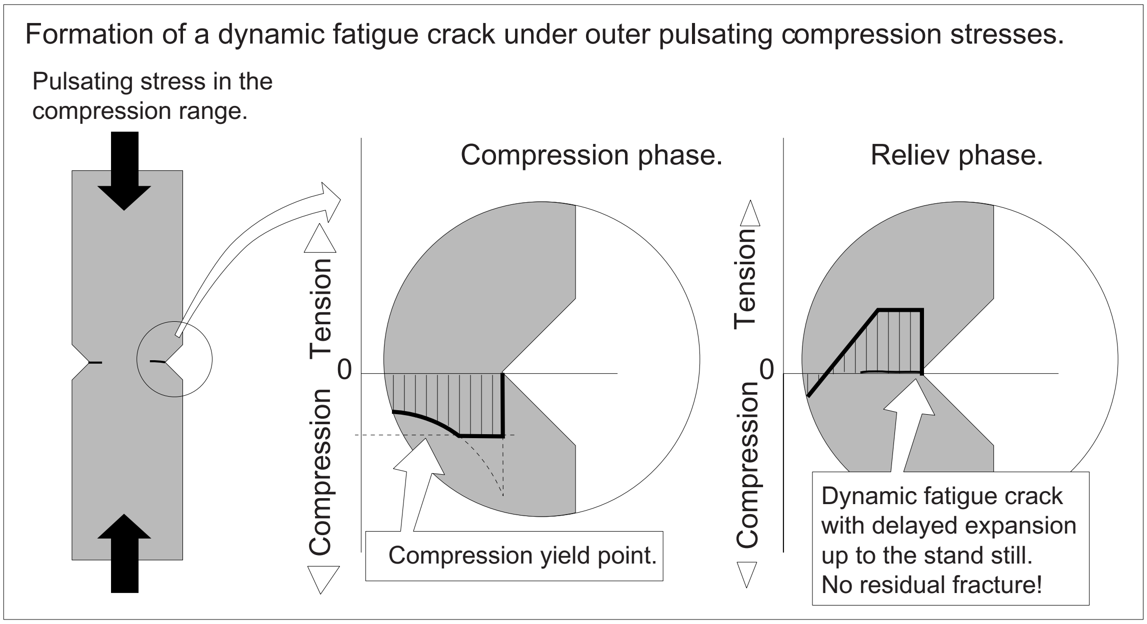


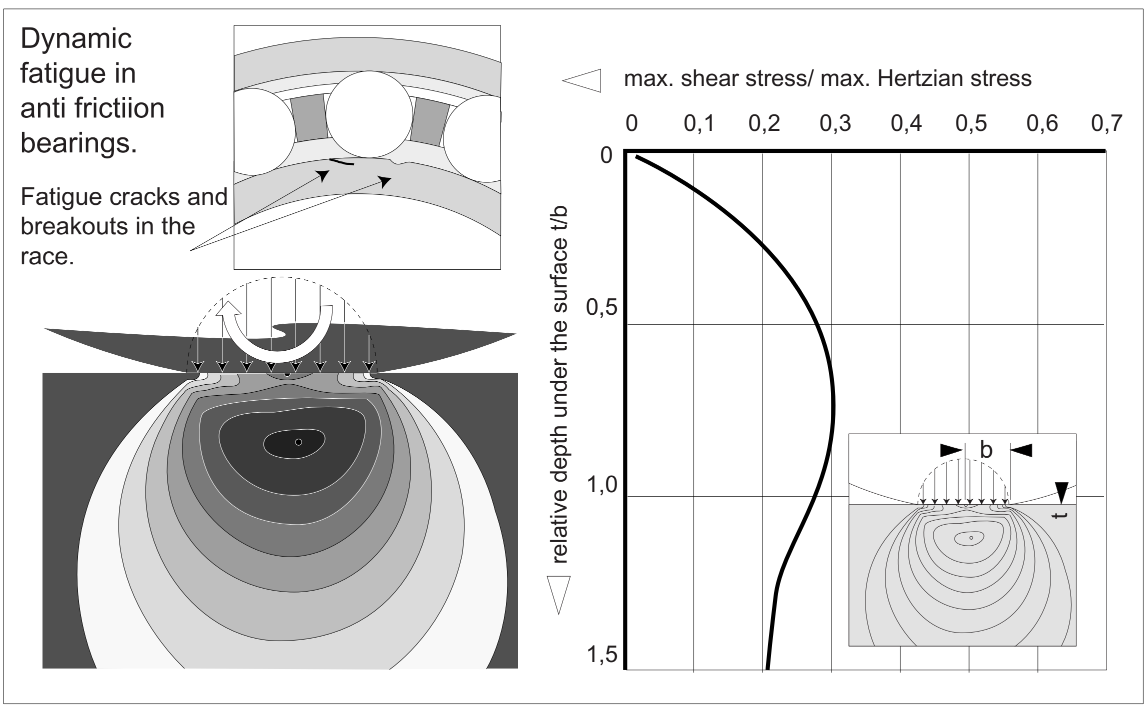
Ill. 5.4-13

already in the raw part existing hydrogen *(Ill. 5.7.1-3).*
- The effect of crack acceleration depends on the ***micro-structure*** *and therefore also the heat treatment.*
- The dynamic loads must act in an ***unfavorable direction*** *(perpendicular)* ***relative to the orientation of the sensitive structure.***
- At ***lower operating temperatures*** *(such as 20°C), the effect is considerably stronger than at high ones (the effect is negligible at 75°C).*
- ***Marine atmospheres, i.e. watery NaCl solutions, accelerate crack growth.***
- At temperatures below 75°C, ***strain-induced hydrogen embrittlement*** *(also see Ill. 5.7.1-4) ahead of the crack tip seems to be the source of the dwell effect.*
It must be noted that the effects are subject to various influences. There are evidently ***titanium alloys with varying degrees of sensitivity*** *(e.g. IMI 685). The temperature up to which the* ***dwell effect*** *occurs also seems to be higher in especially sensitive structures. Supporting evidence for this can be found in high-pressure compressor damage. Temperatures here can be assumed to be around 200°C. This sensitivity is most likely related to the* ***blank part production process*** *(see also Ill. 5.7.1-3), whereby* ***part sizes*** *play an important role. The larger the blank part, the more likely that unfavorable structures will result.*

The **dwell time sensivity** must be determined for dimensioning relevant data and must be, if needed, considered during the assignment of of the LCF lifetime. Thereby must be payed attention at the **compression stress sensivity** and/or **tension stress sensivity**

That dynamic fatigue cracks can develop, at least a tension or shear phase must exist. This is also true, if only outer compression stresses act.





Ill. 5.4-14

Ill. 5.4-14: ***Dynamic fatigue*** *can only occur, if there are sufficient high tension or shear stresses in the load cycle. However there are cases, which seem to contradict this principle. Thereby only pressure forces act from the outside at the system. However at more closely consideration it can be* *identified, that in the incipient crack phase indeed tensile stresses act in a load phase.*

Example 1 *(frame above): In notched components can develop under outer pulsating compression forces dynamic fatigue cracks in the notch root. In the* ***compression phase*** *the* ***notch root*** *will be* ***plastically compressed****. However the surrounding cross section does only deform elastic. In the* ***relieve phase*** *it rebounds. Thereby the plastically compressed region is elongated whereat tensile stresses occur.*

Example 2 *(frame below, Lit 5.4-5): During the roll-over of the race from a bearing the rolling elements execute pressure forces. The „Hertzian stress" produces high stress gradients below the surface. The consequence are* ***pulsating shear stresses****. These trigger incipient dynamic fatigue cracks below the surface. These grow to the surface and trigger fatigue breakouts (pittings). This is the typical. lifetime determining* ***failure mechanism of anti friction bearings****.*

Naturally there are also other similar phenomenons in the technique. To these belong the following failure machanisms:

- ***Cavitation*** *(Ill. 5.5.1.3-1),*
- ***Droplet impact****/erosion (Ill. 5.5.1.2-1).*
- ***Particle erosion*** *(Ill. 5.5.1.1-1).*

In these cases many small patrticles or droplets impact the surface, which fatigues as consequence of microcracks and/or breakouts.

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