Boris M. Klebanov
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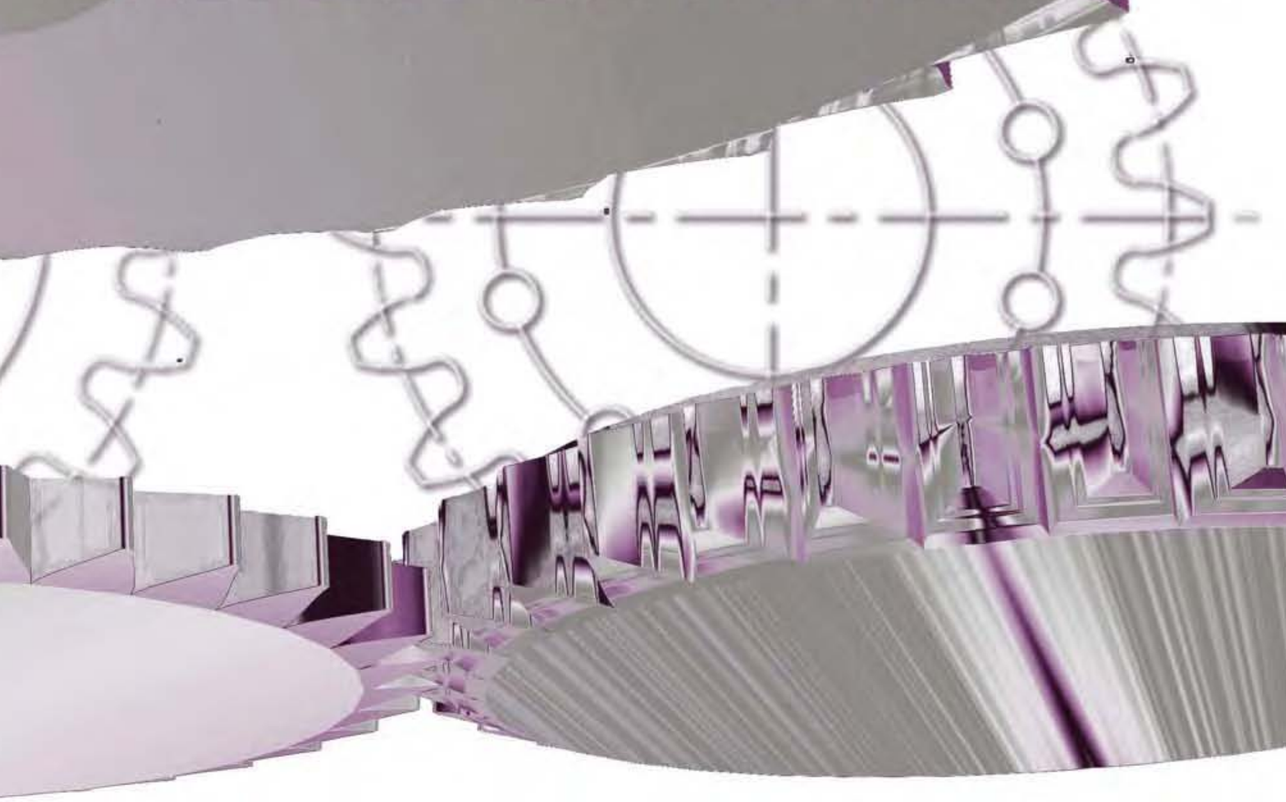

Machine Elements

Life and Design

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Preface

This book describes the behavior of some machine elements during action, based on our understanding accumulated over many decades of machine design. We have sought to describe the mechanisms of interaction between the motion participants in as much detail and depth as the scope of our knowledge and the volume of the book allow.

Our understanding is based in many respects on the work of others, and we have made reference to all authors and publications known to us. But the literature of mechanical engineering is vast, and we welcome notification by any author inadvertently omitted to enable us to amend this omission in the future.

Chapter 1 to Chapter 11 were written mainly by Boris M. Klebanov. Chapter 12 was written mainly by David M. Barlam, who also performed all the calculations using the finite element method (FEM) that appears in the book. Chapter 13 was written jointly by Boris M. Klebanov and David M. Barlam. Frederic E. Nystrom edited the entire work, including the text, tables, and illustrations.

This work is dedicated to our teachers.

Boris M. Klebanov
David M. Barlam
Frederic E. Nystrom

Authors

Dr. Boris Klebanov has spent all 48 years of his professional life in the design of diesel engines and drive units for marine and land applications, reduction gears, hydraulic devices, and mine clearing equipment. His Ph.D. thesis (1969) was on the strength calculation and design of gears. He is the author of many articles and coauthor of two books in the field of machinery.

Dr. Klebanov worked from 1959 to 1990 in St. Petersburg, Russia, as a designer and head of the gear department in a heavy engine industry, and then he worked until 2001 at Israel Aircraft Industry (IAI) as a principal mechanical engineer. Currently, he is a consultant engineer at Israel Aircraft Industry.

Dr. David Barlam is a leading stress engineer and a senior researcher at Israel Aircraft Industry (IAI), specializing in stress and vibration in machinery — the field in which he has accumulated 37 years of experience in the industry and seven years in academia. He is an adjunct professor at Ben-Gurion University. Dr. Barlam's current industrial experience, since 1991, includes dealing with diversified problems in aerospace and shipbuilding. Prior to that, he worked as a stress analyst and head of the strength department in heavy diesel engine industry in Leningrad (today's St. Petersburg). David Barlam received his doctoral degree (1983) in finite element analysis.

Dr. David Barlam is coauthor of the book *Nonlinear Problems in Machine Design* (CRC Press, 2000), and numerous papers on engineering science.

Frederic Nystrom has since 1997 held the position of senior project engineer at Twin Disc, Inc. (Racine, WI). He is responsible for management of both R&D projects and new concept development, focusing on marine propulsion machinery for both commercial and military applications. Prior to that, beginning in 1989, he worked as a senior engineer at Electric Boat Corp., Groton, CT (a division of General Dynamics).

While at Electric Boat he accumulated wide experience in the design of propulsion systems, product life cycle support, and manufacturing support for U.S. Navy surface ships and nuclear submarines. He currently holds U.S. Patent No. 6,390,866, "Hydraulic cylinder with anti-rotation mounting for piston rod," issued May 2002.

Introduction

We know nothing till intuition agrees.

Richard Bach, *Running from Safety*

Possibly, poetry is in the lack of distinct borders.

Joseph Brodsky, *Post Aetatem Nostram*

This book is mostly intended for beginners in mechanical engineering. Undoubtedly, experienced engineers may find a plentiful supply of useful material as well. However, we conceived of this work primarily with novices in mind. We remember all too well how we joined the engineering workforce upon graduating from college, not knowing where to begin. Admittedly, there is still much we don't know, as the processes in working machines are numerous and complex in nature. Nevertheless, we hope that thoughtful engineers will profit from our experience.

As one doctor singularly expressed, "What we know is an enormous mass of information, and what we don't know is ten times greater." We are skeptical about the tenfold estimate; presumably, it is much more. The problem, however, lies not only in the volume of knowledge but also in the fact that most of our knowledge is based on experience in the manipulation of experimental data, whereas many of the laws that govern physical processes are known only partly or not at all. Furthermore, natural, physical processes are statistical in nature, so that as a rule we can't be completely confident that our actions will bring the desired result. Despite this, what we do know allows us in most cases to solve fairly difficult technical problems.

If it is agreed upon that life is movement, then the being of machines can also be called life. To concentrate on the "physiology" of machines, we generally will not refer very much to the change in location of a mechanism's parts in relation to each other. Instead, we will mainly consider elastic and plastic deformations of parts under applied forces, changes in the structure of metals under the influence of stress (in the crystals and on their borders), temperature fluctuations, aggressive environments, and the effects of friction combined with aggressive surroundings, and so on. In all, the life of the machines proves to be very diverse and deserves attentive study.

Anyway, machines are in many respects similar to living creatures. Their birth is laborious. They get afflicted with childhood illnesses (the period of initial trials) and undergo a sort of adolescence (the break-in period); then they work for a long time, get old, and eventually pass away. Machines ache from rough handling; their bodies collect scratches and dents which deteriorate their health and weaken their capacity for work. They suffer from dirt, overheating, and thirst from a lack of lubrication. They also overexert themselves when given loads that are beyond their strength and will perish if nobody looks after their well being. They get tired in the same way from hard work and require check ups, preventative maintenance, and treatment just as people do. They also suffer and become unwell if they are not protected against moisture, heat or cold, soiling, and corrosion. It is no wonder that such terms from the world of the living as "aging," "fatigue," "inheritance," "survivability," and others have entered the technical lexicon. Just as some books focus on the physiology of animals' bodies and habits, this book is concerned with the life phenomena of machines and their parts.

We tried to avoid recommendations as "Do this, it's good" or "Don't do this, it's bad." As with biological life, it is not always possible to say definitely what is good and bad irrespectively of the machine. Sometimes the changes made to improve the design have contradictory results. In addition, many cheaper design solutions are good for less demanding conditions (for example, under relatively small loads, or if the expected service life is brief, or if a higher risk is allowed), but they prove to be unacceptable for the more serious applications. This is why our efforts are directed toward forming the beginning specialist's understanding of the subtleties of the life and work of the machine. Exposure to this material will help them to develop an instinctive impulse to think of those subtleties based upon their own experiences, i.e., to have "mechanical aptitude." This understanding makes the processes of design and calculation more effective, and the work of the designer more sensible, interesting, and creative.

"Ages ago," in 1948, a group of teenagers visited a small electric power station in a small town. This town, just as thousands of other towns and cities in Russia at that time, had been virtually destroyed during the war, leaving many families living in makeshift shelters. And so in the midst of this deprivation, the small power station, with a steam turbine and alternator of only 3000 kW, was a wonder of engineering for the poor children. Everything was fantastic in this shining machine room, but the elderly operator was even more wonderful. He told us:

> A machine is like a person: it likes cleanliness and good, fresh oil [in Russian "oil" and "butter" are expressed by the same word]; it likes when you look after it and take care of it, and is happiest when you don't overload it …

He spoke with inspiration, this unforgettable man, and his hand stroked the shining casing of the turbine …

Part I

Deformations and Displacements

Working mechanisms captivate the imagination. Nice-looking paint and bright chrome please the eye. Mechanical parts move back and forth along their paths, impressive with the accuracy of their purposeful, incessant movement. Everything works beautifully, looks well organized, and delights us all with the gift of engineering and the power of the human mind.

The mechanism works and works, all day, all month, all year … and then suddenly, it ceases working. Something went wrong with it, something broke or became jammed. Or it started to make a heavy noise and vibrations, forcing you to shut it off. Or it exploded and frightened you terribly, so that you started thinking of the stupidity of engineering and, generally, of the imperfection of the human mind. But it was working perfectly well! That means that something had happened to it while it was working! It means that, in fact, the life of the mechanism is much more complicated than is apparent. The captivating, purposeful movement of the parts has been accompanied by harmful processes (side effects), that didn't show any outward evidence until, with time, their accumulated result became apparent.

The physiology of machines is quite complicated. The parts of a mechanism are subjected to working loads and inertial loads. These loads cause the parts to deform elastically and sometimes plastically as well. This leads to changes in the structure of the metal and the accumulation of internal defects within it.

In the connections of parts, where there is sliding or even minute relative motion, the surface layers undergo structural changes and deterioration. Many micro-processes are involved in this macro-process, such as the shearing of microasperities, the plastic deformation of the surface layers, the impregnation of these layers with the components of the lubricant and the mating parts, the formation of particles of oxides and other chemical compounds, and the particles' movement from the contact zone. The friction also creates electricity that interacts with the contacting surfaces and lubricant.

At first, the processes described above may improve the work of the mechanism. In the areas of high stress concentration, the local plastic deformation leads to a more uniform load distribution and lowers the local stress peaks. In the friction zones the microasperities become smoothed out, and form the new structure of the surface layers that is more suited to the friction conditions than the initial one. But as the processes continue, the mechanism becomes less serviceable. It ages. The structure of metal deteriorates … the hinges wear out … the back is hurting … the knees …

It seems that we've moved to another realm! Alas, dear reader, we humans are mechanisms too, and as such, we feel the mechanical problems of aging all too well….

The mentioned above micro-processes in the parts and connections are of vital importance for the "health" of a mechanism and its ability to operate successfully during its service life, which is always limited. Let's focus our attention on these fine matters.

1 Deformations in Mechanisms and Load Distribution over the Mated Surfaces of Parts

A mechanism is a combination of rigid or resistant bodies so formed and connected that they move upon each other with definite relative motion.

Excellent definition! We would not be able to explain it better, so we took this definition from a well-known book.[1]

A mechanism usually begins with a mechanical diagram. The designer draws it on a computer screen or on a piece of paper, depending on where he was caught by a surge of inspiration — sometimes his ubiquitous boss provides him with an initial concept. One day, he draws up a diagram of a parallel link mechanism intended for lifting and lowering a weight (see Figure 1.1a). In this chart, everything looks perfect: two lines (1 and 2) symbolize the upper and lower links of the mechanism hinged to weight 3 and to frame 4. The frame looks respectable compared to lines 1 and 2; such a solid, massive rectangle! Electrical winch 5 turns the links and shifts the weight up and down. The designer was not a beginner. He noticed that in the upper position the links were near dead center, and he checked forces F_1 and F_2 in the links. These forces proved to be large, but no problems concerning the strength of the links and the adjoined elements were found. The designer even checked the stability of the links under compressive load; everything was OK!

Everything was really OK until this mechanism was designed in detail, manufactured, and tested. At the first lifting test, when the mechanism was close to its upper position, shown in Figure 1.1b, the weight suddenly fell down with a great crash and came to a standstill in the position shown in Figure 1.1c. Fortunately, the testers were experienced guys, and they were standing at some distance; therefore, they were not injured. They were only a bit scared and very surprised. The subsequent investigation revealed the following:

Because the weight of the mechanism was required to be as low as possible, frame 4 was welded from thin sheets of high-strength steel (see Figure 1.1b) and was quite pliable; however, its strength was checked and found satisfactory.

In the hinges, "good" clearances were made in order to make mounting of the axles of the hinges easier.

Lower link 2 was designed as two rods connected by cross-members (see view "A"), and upper link 1 was made of one rod and placed in the middle of the lower link, so that the rods of the links were in different planes.

Under load, forces F_1 and F_2 were applied to lugs 6 of frame 4, which bent as shown in Figure 1.1c. The distance between the lugs became increased, and this, combined with the increased clearances between the axles and the lug bores, enabled the mechanism to pop like a convex membrane or a pop-top cap. Thus, this product is not a mechanism in the strict sense, because its members don't "move upon each other with definite relative motion."

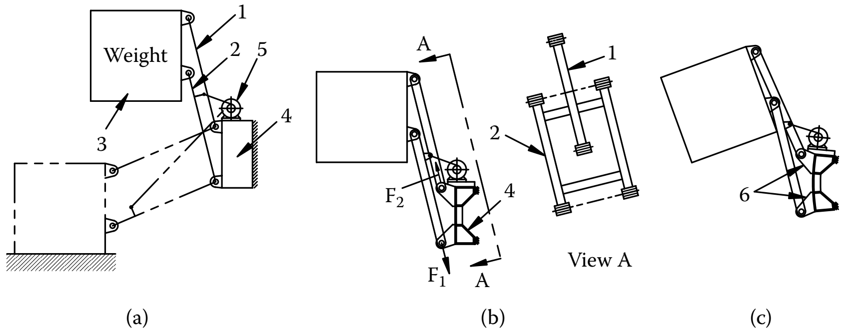

FIGURE 1.1 Lifting device.

From this accident, at least three conclusions should be drawn for the future.

1. The first is well known but worth repeating: Don't stand under the weight! In general, never mind how simple the mechanism, it is always better to stay a safe distance from it at first trials because you never know beforehand what its intentions and capabilities are.
2. The second conclusion: Link mechanisms, which have dead centers, need particularly cautious handling when used near these centers. Increased (for example, because of overload) elastic deformation of the links, enlarged (say, owing to wear) clearances in hinges, small deviations from the drawing dimensions, all become vitally important in these positions and may lead to unexpected and even perilous consequences either at the manufacturer's trials or later in service.
3. The third conclusion: The shape of the machine elements at work may differ significantly from those depicted in the drawing or built in the computer, even if they are manufactured in accordance with the drawing requirements. Therefore, it is expedient to perform a kinematic analysis taking into account the elastic deformations under load.

Let's consider one more occurrence: A designer drew a diagram of a block brake (Figure 1.2). In this brake, the rotation of drum 1 is retarded by blocks 2 with levers 3 and 4. The needed force is supplied by spring 5 through reverser 6. The brake is released by solenoid 7 connected to double-armed lever 8.

It is clear that the farther we want to get the blocks from the drum, the greater the stroke of solenoid 7 should be (or the less the ratio of lever 8 should be, but in this case the solenoid force must be greater). Increasing the stroke or the force of the solenoid leads to such a sizeable increase in its dimensions, weight, and cost that the designers usually make the distance between the blocks and drum (when released) very small, approximately 0.5–1 mm.

Our designer did just that. He had calculated levers 1 and 2 for bending strength only. When the brake was manufactured, it made a good impression on the workers; it was lightweight and smart. They tightened spring 5 by nut 9 to the needed length, and then pressed the release button. The solenoid clicked — the testers were certainly a little distance away, but not far — and they heard the click and saw lever 8 turn. But blocks 3 kept gripping the drum safely.

The post-test investigation revealed that the bending deformation of levers 3 and 4 was considerably greater than the designed displacement of the blocks. So when released, the deformation of the levers became less, but the blocks remained in touch with the drum, though the grip force decreased. As you see, in this case, making a kinematic analysis without taking into account deformations was erroneous.

When a designer uses a spring in a mechanism, he must take the length of the spring depending on its load. It is obvious. As far as other elements are concerned, there is some kind of inertia,

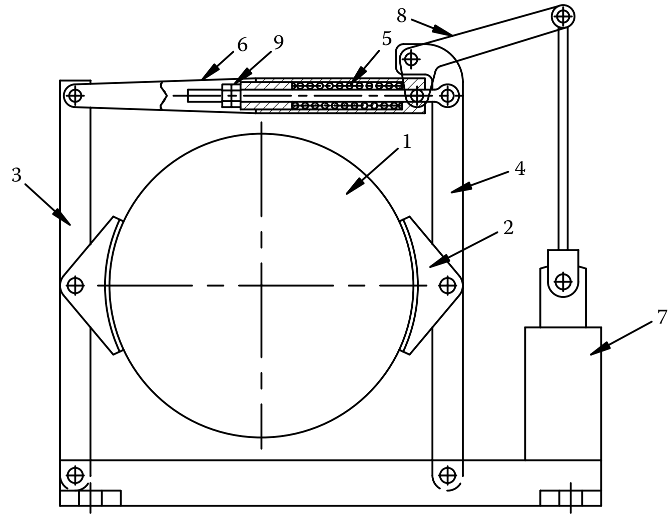

FIGURE 1.2 Block brake.

which possibly originates in calculations of beams for strength, where relatively small deformations are neglected. It should be noted that as the strength of materials increases, the deformations also increase, because the modulus of elasticity doesn't change, so the influence of the deformations grows.

In particular, frame 4, shown in Figure 1.1, was welded of thin sheets of steel of 900-MPa-yield strength, and under load, it was changing its form similar to a spring. It was actually visible!

In the kinematic analysis, even small deformations and clearances may cause important changes. Designers of mechanisms, which must have high kinematic accuracy, know that and take it into account. They design the machine's elements to be rigid, which often leads to increased weight. (For instance, levers 3 and 4 in Figure 1.2, after the trial had failed, were made much more massive, and this enabled the kinematics to be closer to the initial design.)

These two examples, which show how a lack of strain analysis leads to a mechanism's complete inability to work, relate rather to curious things, which are remembered by the participants with amusement. In practice, however, lots of examples may be found of how important it is to pay attention to relatively small deformations, even of microns. These deformations don't usually disable the mechanism, but they change the load distribution between mating parts as compared with the load distribution assumed in the strength calculations. This may result in the unsatisfactory functioning of the mechanism (increased noise, vibrations, overheating) or in premature failure. Such defects are often brought to light after a long period of time, when the mechanisms are being manufactured in quantity and their upgrade would require considerable expense.

Figure 1.3 depicts one end of a tie bar loaded with a variable axial force, F (the second end is similar). The tie bar consists of tube 1 and two lugs 2 welded to the ends of the tube. While in service, these tie bars have failed several times; the cracks were placed as shown: three cracks in 120° intervals. Investigation revealed that the cracks originated in plug welds 3. These welds are used for the preliminary attachment of the lugs to the tube before welding main seam 4. But the plug welds don't know that they are only needed to align the weld, and at work, they participate in load transmission between the lug and the tube. The plug welds' share of the load doesn't depend on their relative strength, but only on the ratio of compliances between the tube and shank 5 of the lug.

On the right of welds 3, the entire force F is transferred through tube 1. From the section where welds 3 are placed, part of the force is transferred through welds 3 directly to shank 5 of the lug,

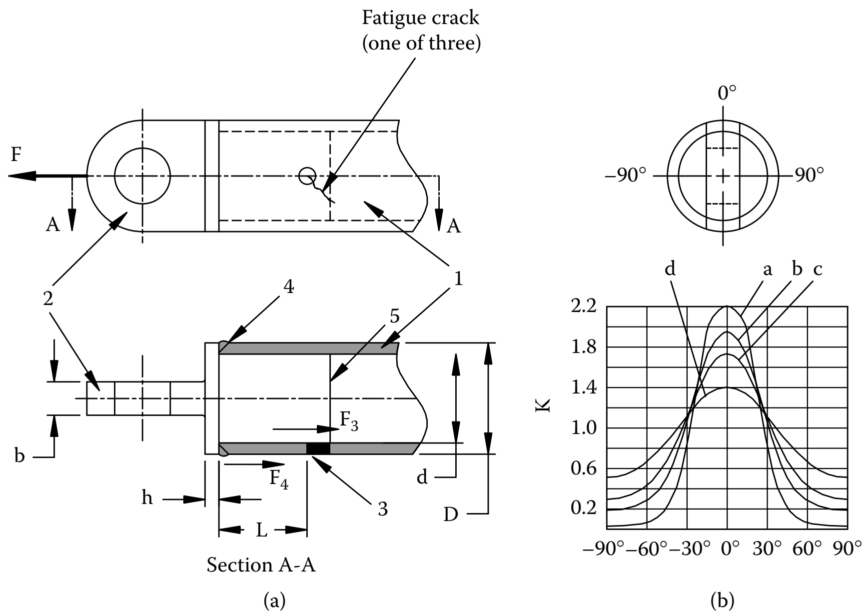

FIGURE 1.3 Tie bar (the left end is shown; the right end is identical).

and the rest of the force is transferred through tube 1 and weld 4 to the lug. The main thing we have to do to estimate the load distribution between the welds is to name the forces. Let's designate the forces transferred by welds 3 and 4 as F_3 and F_4 respectively. The rest is easy: just write the equations for deformations.

The elongation of shank 5 between welds 3 and 4 is

$$\delta_S = \frac{F_3 L}{E_S A_S}$$

The elongation of tube 1 in the same interval is

$$\delta_T = \frac{F_4 L}{E_T A_T}$$

In these equations, A_S and A_T are the areas of cross sections of the shank and the tube respectively, and E_S and E_T are the moduli of elasticity of materials.

Taking into consideration that $\delta_S = \delta_T$ and $E_T = E_S = E$ (because both the shank and the tube are made of steel), we find that

$$\frac{F_3}{A_S} = \frac{F_4}{A_T}$$

Because $F_3 + F_4 = F$,

$$F_3 = \frac{F}{1 + A_T / A_S} = \left(\frac{d}{D}\right)^2 F$$

In the case under consideration, the outer diameter of the tube D = 80 mm, and the shank diameter d = 64 mm, so

$$F_3 = \left(\frac{64}{80}\right)^2 F = 0.64F$$

This calculation is not exact because plug welds 3 don't connect the entire circumference of the shank but represent three local plug welds. Therefore, the actual compliance of the shank is greater and, consequently, the force F_3 should be smaller. The finite element method (FEM) gave F_3 = 0.48 F. This more precise definition is not of principle in this case, because the conclusion remains unchanged: the plug welds should be avoided.

Now, after we have canceled welds 3, the load of weld 4 doubled, and we have to check its strength. Lug 2 is placed near the weld; thickness h of the shoulder looks fairly small, and it is clear that the load distribution must be sufficiently uneven. If we don't use FEM, we can calculate the mean tension stress by dividing the force F by the weld area, which is approximately equal to A_T. But this stress is undoubtedly less than the peak magnitude of it. We also can calculate the stress assuming that the entire force is transferred in the area of the lug width b. In our case, b = 24 mm, and the calculated stress will be 4.7 times the mean value. This is more than the real peak magnitude, but if the weld doesn't stand this stress, we need to know more exactly the real peak magnitude. In Figure 1.3b is represented the stress distribution in weld 4 found using the FEM model. Curves a, b, c, and d correspond to h = 10, 15, 20, and 30 mm, respectively. On the ordinate is plotted value $K = \frac{\sigma_{\text{local}}}{\sigma_{mean}}$ In the following pages, we will often face the problem of load distribution between parts and their elements.

Sometimes the interaction of two parts is influenced by many other parts connected with them, so the deformation analysis becomes multifarious. Figure 1.4 shows a draft of a gear. The load distribution along the teeth depends mainly on the parallelism of the shafts or, to put it more precisely, on the parallelism of the shafts' segments, which are adjoined to the gears. Figure 1.5 depicts factors that have an effect on the possible lack of parallelism:

The nominal position and forces applied to the gear and the pinion (Figure 1.5a)
The bending deformations of shafts (Figure 1.5b)

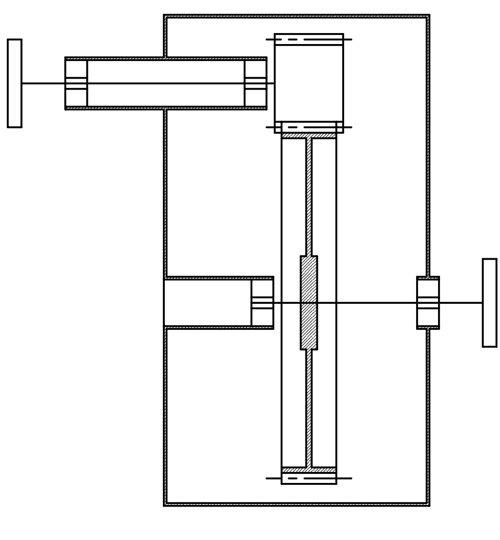

FIGURE 1.4 Gear sketch.

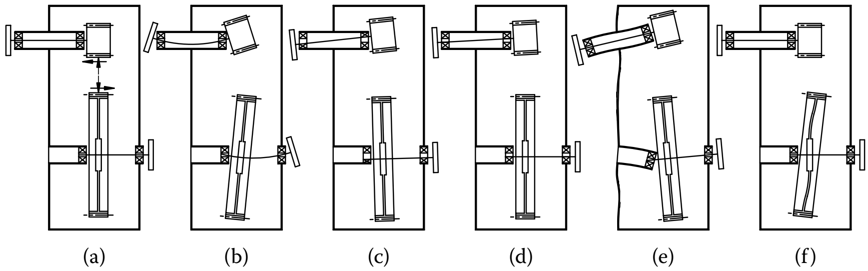

FIGURE 1.5 Displacements of shafts and gear rims caused by elastic deformations of parts and by bearings clearances.

The shafts' displacement caused by elastic deformation of bearings (Figure 1.5c)
The shafts' displacement caused by take-up of radial clearances in the bearings (Figure 1.5d)
The shafts' displacement caused by deformation of the housing (Figure 1.5e)
The gear wheel displacement caused by deformation of its body (Figure 1.5f)

Some factors which are hard to represent in the same manner should be added here, too:

Torsional deformation of the pinion, which may be considerable when the length of the pinion is greater than its diameter
Uneven radial deformations of the gear and the pinion caused by uneven heating and centrifugal forces (relevant mostly to high-speed gears)
Uneven rigidity of the teeth when the toothed rim is thin (see Chapter 8, Section 8.2)

Uneven load distribution along the teeth may also result from an unsuitable way of lubrication. The authors have observed deep pitting of the teeth profiles in the middle of the gear teeth, which took about 10% of the gear face width. It was placed exactly in the area where the lubricating oil was brought to the teeth by a narrow idler immersed into oil (called *rotaprint lubrication*; see Chapter 7, Section 7.10). To avoid such an effect, the width of the lubricating idler should be 70–80% of the gear to be lubricated.

Among the omitted factors are manufacturing errors and possible deformation of the housing while it is being attached to some foundation or substructure. These errors are very small, and the alignment of the teeth (bearing pattern) is finally checked by painting the teeth with dye and examining the pattern of dye transferred to mating teeth.

From Figure 1.5 we can see that the direction of displacements may change. This depends on the relative position of the bearings and gears, housing design, and direction of the applied forces. By means of proper design, the effects of some of the previously mentioned components of deformation might be mutually offset. This can be done by a reasonably chosen combination of gear element rigidities and direction of the axial force. But the main way is to decrease the deformations as far as possible.

From this point of view, the design shown in Figure 1.4 is extremely bad, because it is easily deformable. It is intentionally drawn to show the elements of deformations more clearly.

Analyses of deformation might be time consuming even if its most complicated components, such as the housing deformations, are made negligible by proper design. But the deformations of the majority of machine elements, such as shafts, bearings, gears, levers, etc., may be calculated by the so-called "engineering methods".

Sometimes the "engineering methods" are completely useless, and satisfactory results may be obtained by FEM only. Figure 1.6 shows a connection between piston 1 and connecting rod 2,

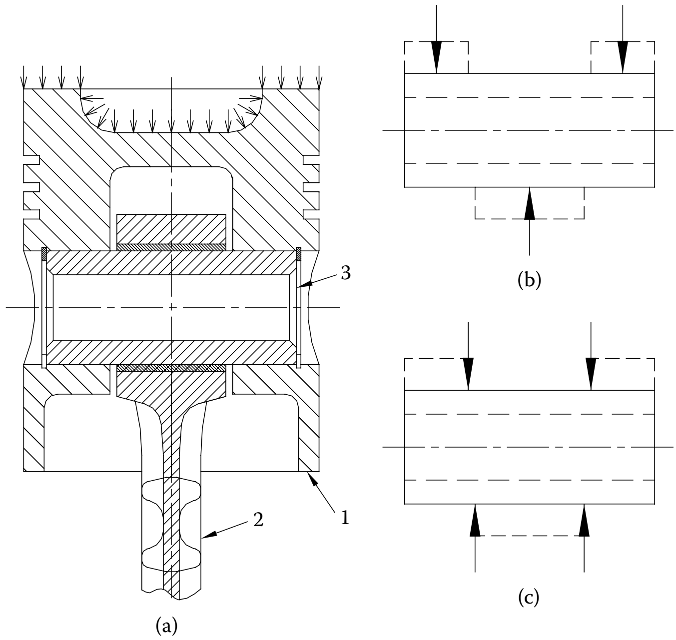

FIGURE 1.6 Loading of a piston pin.

provided by pin 3. The load distribution over the contacting surfaces of the pin depends on the elastic deformations of the parts and the hydrodynamic oil film parameters in the bearings. Trying to make strength calculation of the pin by engineering methods, we can consider two extremely simplified options of loading shown in Figure 1.6b and Figure 1.6c. The first option (Figure 1.6b) gives the stress almost twice as high as the second option (Figure 1.6c). But this pin is hollow, and, in addition to bending and shear stresses, it suffers from bending of its cross section (ovalization). These stresses can be easily calculated for a ring (two-dimensional problem), but the three-dimensional problem seems to be too hard for simplified analysis. What is important, the dependence between the stress and the load is nonlinear in this case. That means the stress increase in the pin is less than the increase of force *F*. The reliable determination of stresses can be achieved here only by FEM analysis and, finally, by measuring them on a working machine.

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