High Speed Gears - New Developments

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1. Introduction

Compressors and other turbomachines are constantly developed and improved to run at higher speeds with more power. The gear manufacturer therefore has to design gears which can transmit high toothing forces at very high rotational speeds. But the toothings and the bearings are bound to certain limits for thermal and mechanical load which must not be exceeded for safe operation. These limits have been elevated over the last years by continuous development of toothing and bearings; see Figure 1 for the pitch line velocity. Fig. 1

To solve the basic design problem of high speed and power, there are in general two possibilities: To design a gear with power split, i.e. with two or more power paths or to increase the limits for toothings and bearings by development of these components.

Gears with power split reduce the load on bearings and toothings and allow the designer to stay within well known limits. On the other side, such gears have a more complex lay out, they have more tooth meshes and more bearings and they need some mechanisms for a correct power split. These mechanisms can be quill shafts, self adjusting bearings or others. The costs to build such a complex gear must be almost twice as much as for a simple two shaft gearbox.

Therefore, the question is put: How can toothings and bearings be further developed in order to allow higher limits for load and speed? Consequently, more power and speed could be transmitted in simple, reliable and cost effective two shaft gear designs.

In 1985, MAAG started a research program to investigate on bearings and toothings. Gears with high power and speed were designed to be carefully tested on a special back to back test bed. During the design phase, one recognized that before the toothing, the pinion bearings reached their load limits. New radial and axial tilting pad bearings had to be developed for these gears to allow for safe operation with maximum white metal temperatures below 130 °C. The design of these new bearings as well as the test results under full load are presented in this paper.

2. Back to Back Test Bed

The mentioned back to back test bed is shown in Figure 2. It consists of two identical gearboxes which are mechanically coupled with a torque meter device on the low speed shafts and with a special toothed coupling on the high speed shafts. Fig. 2

With the single helical toothing, the gears can be loaded up to full load just by applying an axial force on the wheel. The axial shifting causes a rotary movement and the wanted closed torque circuit between the two gears is established. The axial force is produced by means of hydraulic pistons. With a variable hydraulic pressure on these pistons, any desireable load between zero and full load can be achieved. As driving power, only the total losses of the two gears have to be provided.

Some technical data of the back to back test bed:

- Nominal power: $P = 30'000$ kW
- Nominal speed: $n = 6380/15574$ rpm
- Overspeed 120 %: $n = 7656/18689$ rpm
- Nominal pitchline velocity: $v = 200$ m/s
- Overspeed plv.: $v = 240$ m/s
- Center distance: $a = 422$ mm

Two sets of gearwheels are tested: One with a helix angle of 13 °, the other with 19 °. The rotors are equipped with strain gauges at the tooth root and with thermocouples near the toothing and journals. With these instruments, thermal distortions due to unequal temperature distribution and actual load distribution across the face width of the toothing are measured. The bearings are equipped with thermocouples in the hottest zones, in order to determine maximum white metal temperatures and thermal deformation. Figure 3 shows the instrumentation of the test gear rotors. Fig. 3

All shafts equipped with instruments are hollow to take all the cables which are connected via special high speed slip rings to the static analysis instruments. Rotor vibrations are to be surveyed by pic-up's; all other instruments and sensors are as for a common industrial platform gear. The wheel of the slave gear is not equipped with any sensors, but the hydraulic axial force device and the input motor drive are connected to it. At the input shaft, the total losses of both gears can be measured by means of a torque meter coupling.

The toothed coupling between the two pinions is a specially designed coupling for highest speeds. Its weight and overhang has been minimized in order to satisfy lateral critical speed requirements of the pinions. Without this couplings it would not have been possible to find a satisfactory solution for save operation at these speeds. Every other type of coupling, as for example a disc coupling, has more weight and more overhang and is therefore only of limited use for extreme high speed gears.

Due to thermal expansions and friction in the toothed coupling, additional axial forces will act on the gears. We have experienced from earlier back to back tests that for the unloading of such a unit, it is not sufficient just to release the axial force. The gears had to be unloaded by applying an axial force in the reverse direction, in order to overcome with the friction in the toothed coupling! From these facts we have learned that the friction in toothed coupling is existing and produces axial reaction forces which certainly cannot be neglected.

This led to the knowledge that extreme high speed gears, which must be equipped with toothed coupling for lateral critical speed reasons, should have a single helical toothing which is not affected by additional external thrusts.

On a double helical toothing, an external thrust would act on one helix only, which has to be considered as a worst case situation for the toothing.

3. Radial Tilting Pad Bearings

3.1 Design

For high speed gears, white metal lined slide bearings are commonly used. The known limits for such types of bearings are as follows:

- specific load 3,2... 4 N/mm2
- maximum white metal temperature 130 °C
- with circumferential speeds above 90..100 m/s, tilting pad bearings should be used in order to avoid bearing instabilities due to oil whip.

During the design phase of the back to back gears, it became evident that the pinion bearings could not be realised with a "conventional" design. A new design had to be found in order to keep the white metal temperatures within mentioned limits. The solution is a specially developed tilting pad bearing with following design features:

- Ratio width/diameter for the main pad: 1,4.
- Three pads, one main pad and two auxilliary pads.
- The main pad has a circumferential groove in the center to evacuate the hot oil.

Materials and fabrication methods are the same as for conventional bearings.

Table I compares important design parameters of conventional and new bearing design for back to back gears.

	Conventional	New Design
D (mm)	170	150
B/D	1.0	1.4
p (N/mm^2)	3.0	2.8
v (m/s)	139 (167)	122 (146)
Tmax (°C)	132 (141)	121 (130)
( ) values in brackets refer to overspeed 120%.		

Table I: Design parameters comparison.

The general design of the bearing is shown in Figure 4. Important is the circumferential groove for reduction of thermal deformations due to temperature gradients over the face width of the bearing. Fig. 4

3.2 Test Results

The back to back gears have been run at full load and up to 120% speed for many hours. The bearings, as well as the toothing, have been operating very satisfactory during the whole testing period. Careful inspection after test runs did not show any sign of wear or damage. This is of course not surprising since bearings and toothing are designed for infinite life.

The radial tilting pad bearings were equipped with thermocouples in the hottest area across the full width of the main pad, see Figure 5. Fig. 5

Measured maximum white metal temperatures are shown in Figure 6. Fig. 6

An analysis of the measured white metal temperatures and other test results leads to following conclusions:

- The maximum values have been always lower than 125 °C, at full load and full speed.
- Measured values are very close to the calculated mean temperature of 121 °C.
- The circumferential groove stabilizes the temperature gradient at a lower level.
- At overspeed 120 %, i.e. with a circumferential speed of 146 m/s, no signs of oil whip occured.

The stability of the proposed tilting pad bearings is, as expected, very good.

- Pinion lateral vibrations have been always below 1.1 mils, even at no load and overspeed.

- The lateral vibrations behaviour of pinion and toothed coupling is good, the damping of the bearings is satisfactory.

3.3 Deformation Analysis

An important question for the development of a wide bearing is the heat distribution across its width and the resulting deformations. If these deformations are too large, the outer parties of the bearing will not take significant load and the heat will be concentrated in the center. This would result in overload for the bearing and consequently lead to damage. The optimal compromise for the B/D ratio must be found. After some theoretical and experimental investigations, we ended up with a 1,4 ratio. With a finite element analysis, the deformations of the bearing have been calculated. The stationary temperature distribution in the steel body of the bearing and the pressure distribution in the oil film are acting as loads on the main pad. Figure 7 summarizes in a compromised sketch the complete results of the FE-analysis. Fig. 7

Following conclusions can be drawn:

- 80 % of the combined deformation are caused by unequal temperature distribution and only 20 % by mechanical bearing load.

- It is for this reason of utmost importance to keep the temperature in the bearing low and uniform.

- The maximum deformation at the outer end of the bearing is approximately 0,08 mm. Supposing a minimum oil film thickness of 0,03...0,04 mm under load, it is evident that even the outer areas of the main pad will contribute to take load.

4. Axial Tilting Pad Bearings

4.1 Design

High speed gears are for mentioned reasons often designed with a single helical toothing, i.e. toothing thrust is to be compensated. In addition, external thrust from couplings can act on the high speed shaft. The helix angle of the toothing must be high enough to limit the heat generation in the gear mesh. Thus, the question is put how to absorb high axial forces at high speeds. The problem cannot be solved with a single axial tilting pad bearing or by shrunk-on thrust collars, due to load limits and centrifugal forces.

For the described back to back gears, two axial bearings have been arranged in series. It was possible to find a mechanism which allows for any power split between several axial bearings. These mechanisms are described in detail in the international patent description.

The basic principle is explained with Figure 8: On the upper half, the unloaded axial bearings are shown; below is the same arrangement under axial thrust. Each axial bearing (5) is supported by a ring (6) which rests on an arrangement for load adjustement (7, 8). These arrangements are supported by a rigid casing (9) which is connected to the main (gear) casing. The arrangement for load adjustement consist of movable pistons which are preloaded by compressed springs. The preload for these springs can be chosen according to the load limit of the used axial bearing or to the preferred stage load for certain operating conditions. The piston of the last arrangement (7) has a stop (7.3) which limits the total axial mobility. The axial bearing clearances of the various stages are different: They are increasing from the inner to the outer bearings. With increasing axial thrust, the inner bearing starts to take load first until its spring preload limit is reached. An axial movement takes place until the second bearing has load and so on. With full axial thrust, each bearing has load according to the predetermined load sharing. Fig. 8

There are many different variations of the described basic principle. A very interesting one is the thrust split between rigidly coupled machines, as for example in Figure 9 turbine and gearbox. Fig. 9

Here, the toothing thrust helps to unload the turbines' axial thrust bearing. In addition, a well defined part of the total thrust is absorbed with a bearing in the gearbox, without hindering the shafts to expand thermally. The bearing clearances have to be adjusted in a way that the gears' axial bearing is loaded even with maximum shaft expansion. The stationary bearing (7) can be positioned at the cold end of the turbine what helps to reduce the thermal expansions to a minimum.

The described arrangement for axial bearings can be used not only for gears, but principally for any rotating machines where high thrusts at high speeds must be absorbed. It is further of some interest that with a correct design, the power losses of two small bearings in series are lower than of one big bearing alone.

4.2 Test Results

For the pinions of described back to back gears, the axial bearings have been designed according to the load sharing principle, see Figure 10. Fig. 10

The toothing thrust is acting here outwards and the outer bearing will be loaded first with increasing power transmission.

Technical data are as follows:

pinion speed:	15'574 rpm
helix angle:	19 °
total axial thrust:	81'000 N
preload of springs:	50'000 N
specific bearing loads:	3,5/2,35 N/mm^2
circumferential bearings velocities:	137/144 m/s
bearing power losses:	34/31 kW

It would not be possible to take such a high total thrust with a single bearing at this speed. For the case of torque reversing, a "back" bearing is installed. Oil supply and instrumentation with thermocouples in the pads are realised exactly the same way as for a conventional single bearing design. The natural frequency of the preloaded piston springs was chosed to be different from potential exiting frequencies.

The results from back to back testing can be summarized as follows:

- No wear, no running tracks
- Maximum white metal temperature well within permissible limits, as calculated in advance
- No axial vibrations
- Free mobility of piston, no running tracks

From these results after many hours of operation one can conclude that the axial load sharing according to the described principle works perfect.

5. Summary and Outlook

The design of high speed gears with two shafts is limited by the bearings and the toothing. Often the bearing limits are reached before the toothing becomes critical.

New radial tilting pad bearings have been developed. With these bearings, the rotor speed can be further increased at the same bearing load. In order to absorb high axial thrust at high speed, a new principle has been developed which allows for several axial bearings arranged in series. The load sharing between the stages can be chosen to any desireable values.

Both bearing designs have been tested on a back to back test bed with pinion speeds of more than 17'000 rpm and power up to 30'000 kW. These tests have proven that both designs are ready for industrial application.

What is the possible increase in speed? Assuming a maximum white metal temperature of 130 °C and load, specific load and circumferential bearing velocity to be constant, one can find following relation:

$$n\ (\text{increased}) \approx \sqrt{1.4} \bullet n \approx 1{,}18 \bullet n;$$

where n = max. perm. speed with "conventional" bearings.

This means that with the same thermal load on the pinion bearings, the pinion speed can be increased by approximately 20 % when tilting pad bearings of the new type are used. Consequently, the range of high speed gears based on a cost effective two shaft design is considerably increased.

Figure No.	Description
1	Development of toothing pitch line velocity since 1920
2	Back to back test bed
3	Instrumentation of test gear rotors
4	MAAG radial tilting pad bearing
5	Instrumentation on bearings
6	Measured white metal temperatures
7	FE-Analysis Radial deformations under thermal and mechanical load
8	Multiple thrust bearing principle
9	Thrust splitting between gas turbine and gearbox
10	Double thrust bearing design of test gear

Development of Toothing Pitch Line Velocity Since 1920

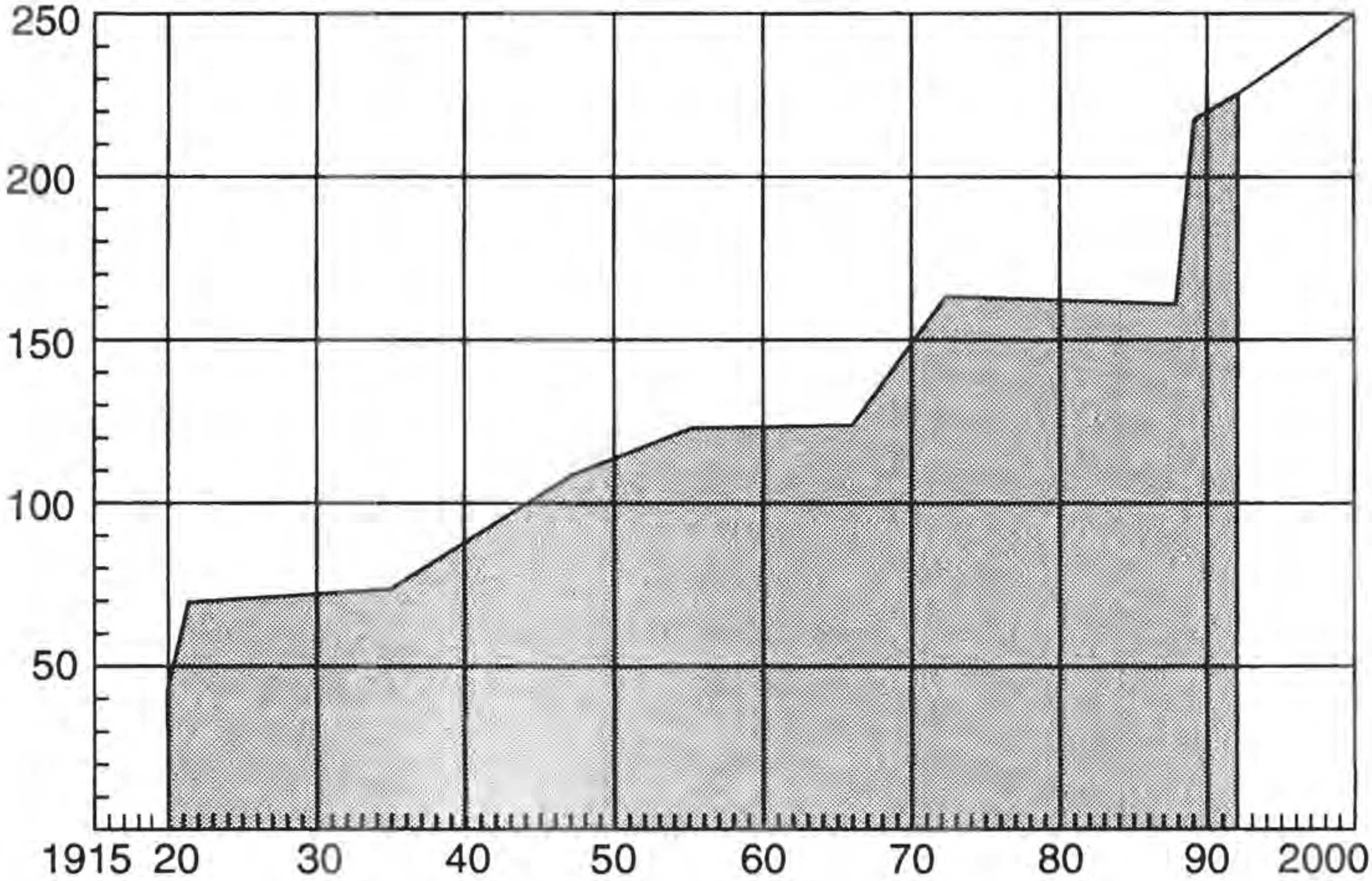


Fig. 1

Back to Back Test Bed

MAAG

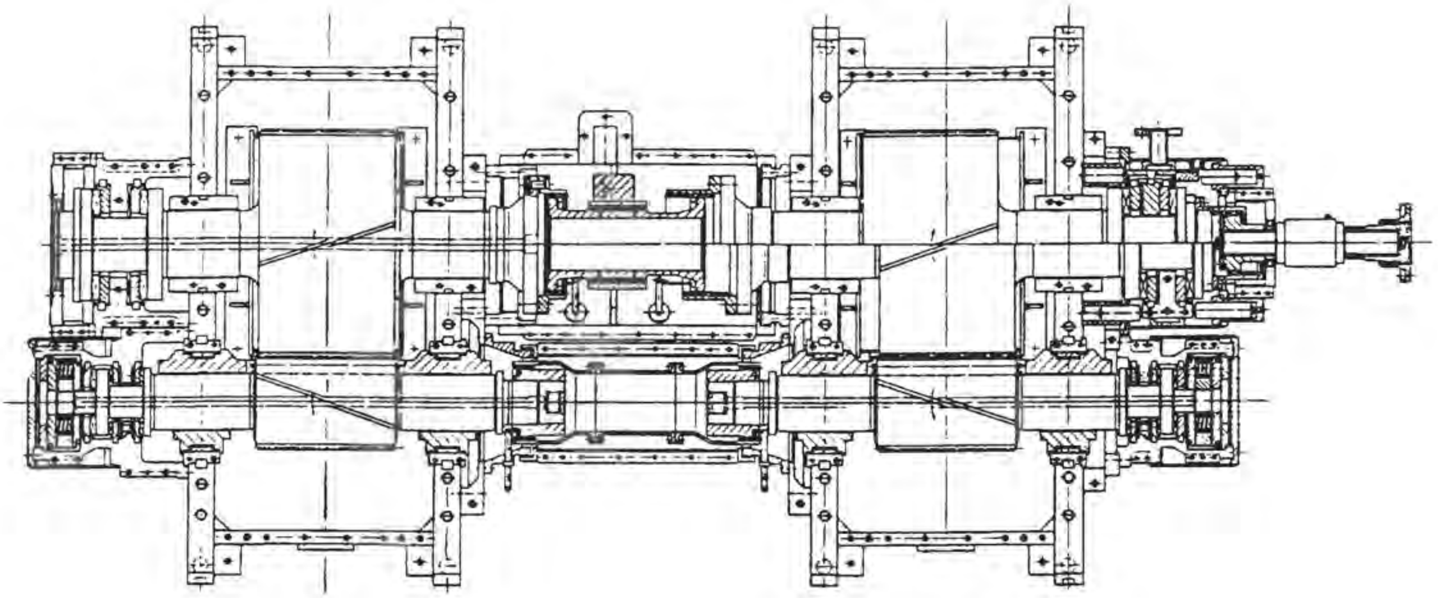


Fig. 2

	Conventional	New Design
D (mm)	170	150
B/D	1.0	1.4
p (N/mm^2)	3.0	2.8
v (m/s)	139 (167)	122 (146)
Tmax (°C)	132 (141)	121 (130)
( ) values in brackets refer to overspeed 120 %.		

Table 1

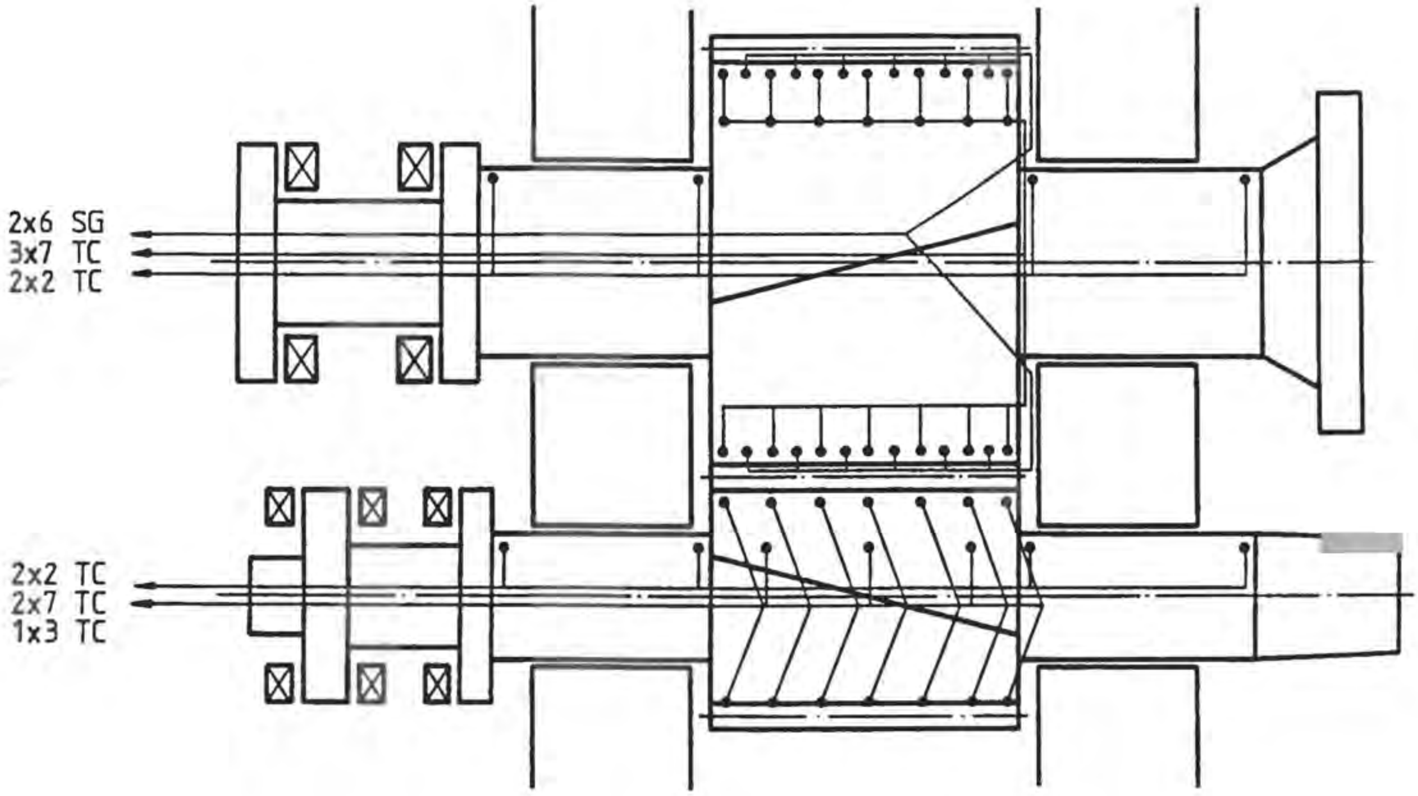


Fig. 3

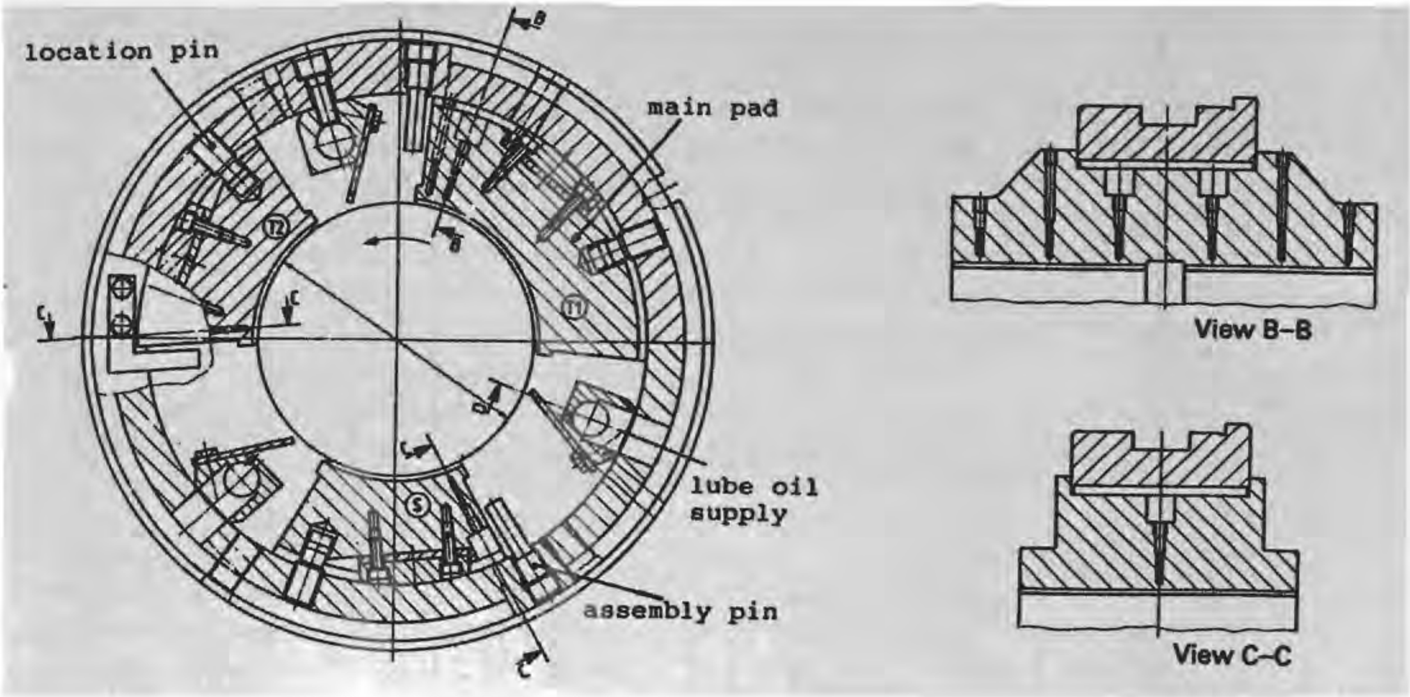
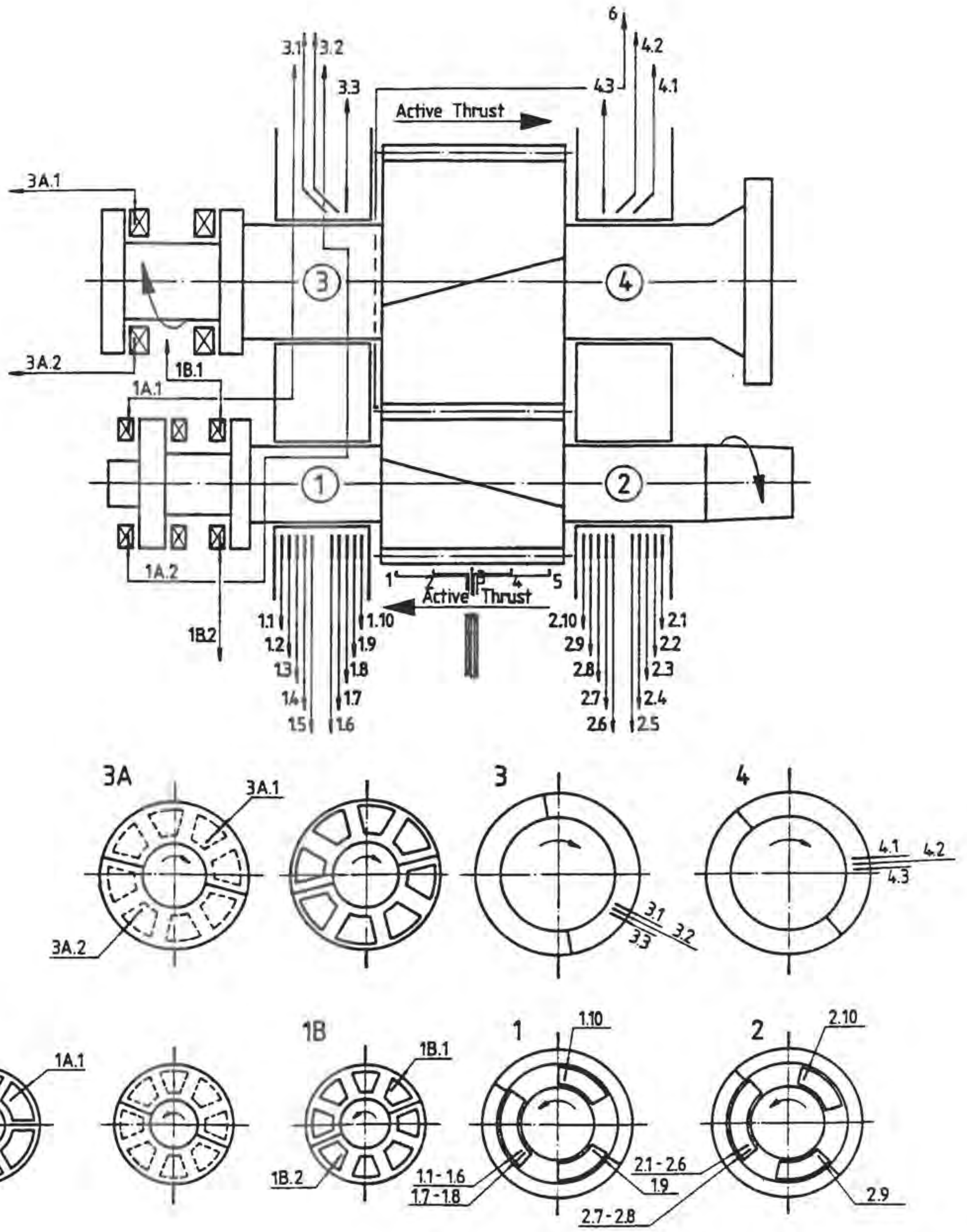

location pin
main pad
lube oil
supply
assembly pin
View B–B
View C–C

Fig. 4



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4.3
4.1
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Active Thrust
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3A.2
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Active Thrust
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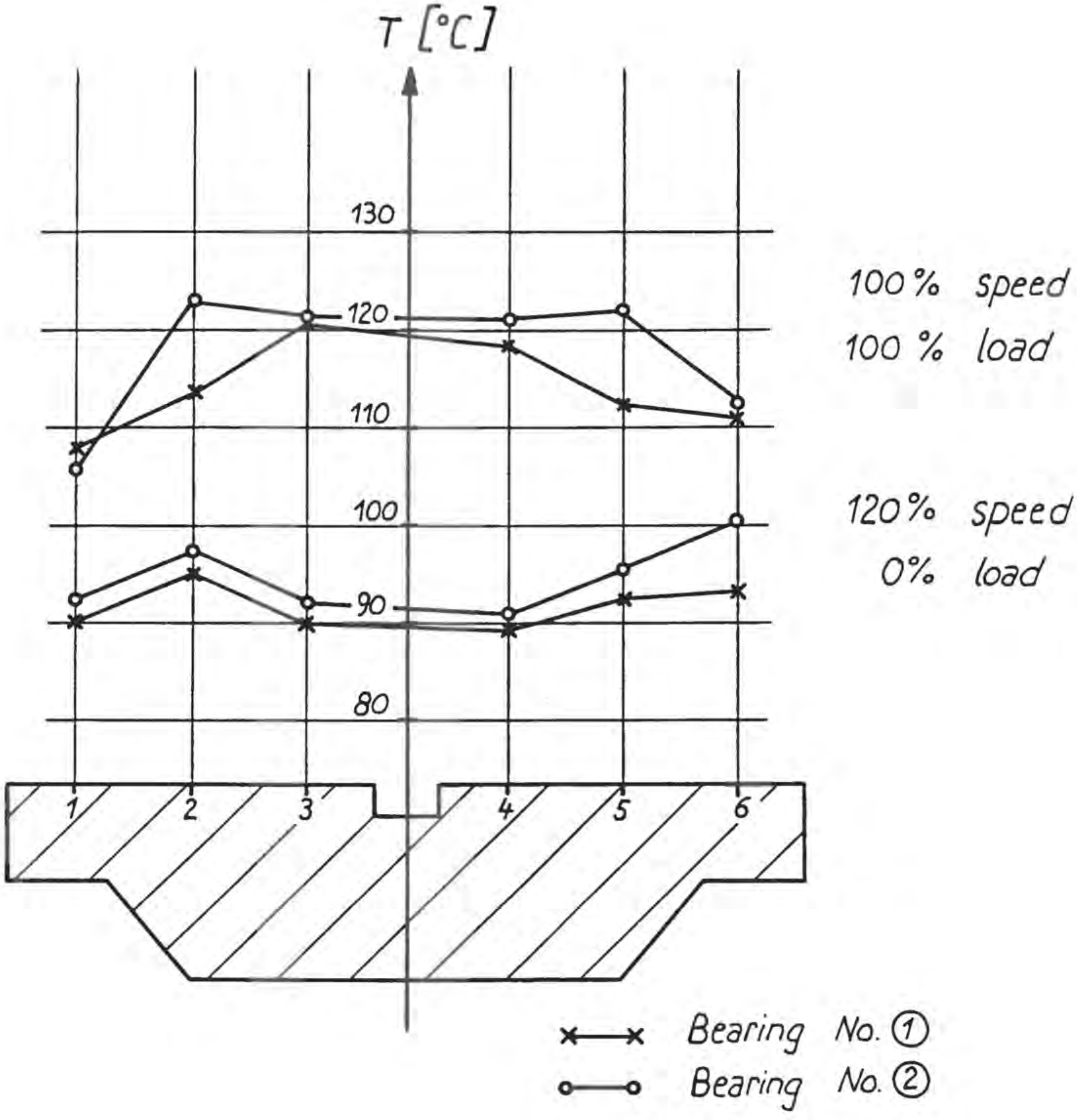
Fig. 5


T [°C]
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120
110
100
90
80
1
2
3
4
5
6
100 % speed
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Bearing No. 1
Bearing No. 2

Fig. 6

FE-Analysis - Radial Deformations under Thermal and Mechanical Load

MAAG

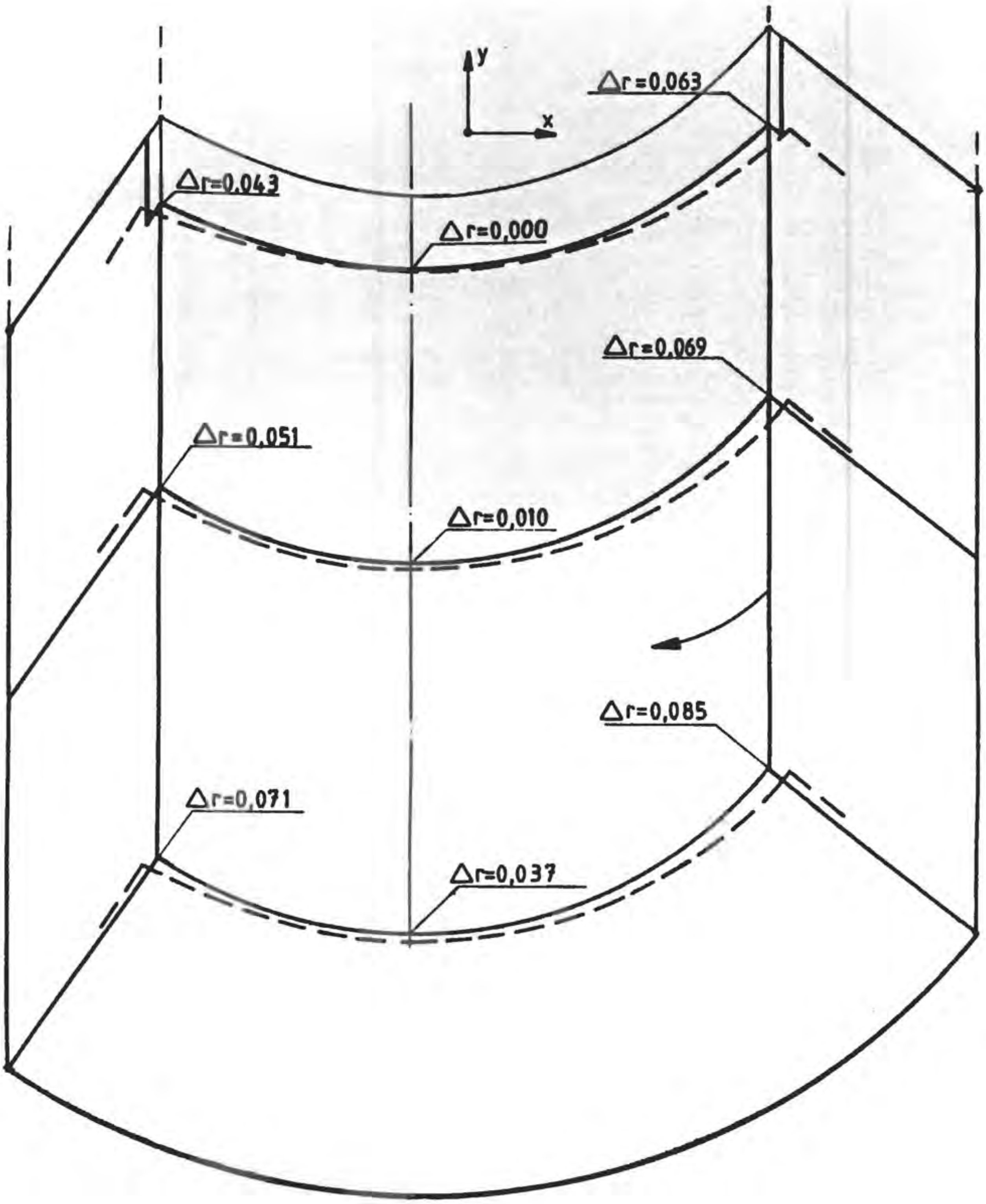

Fig. 7

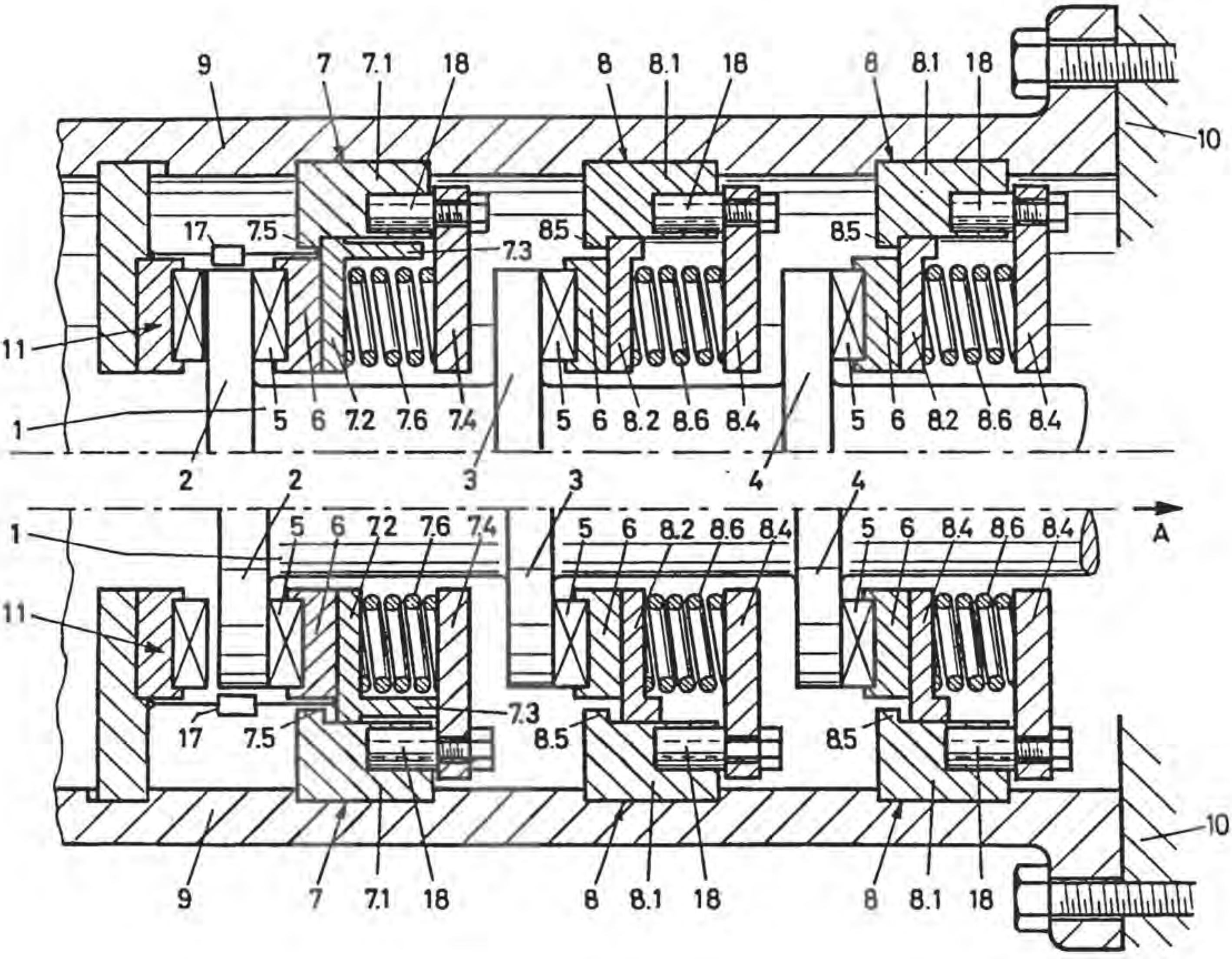
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Fig. 8

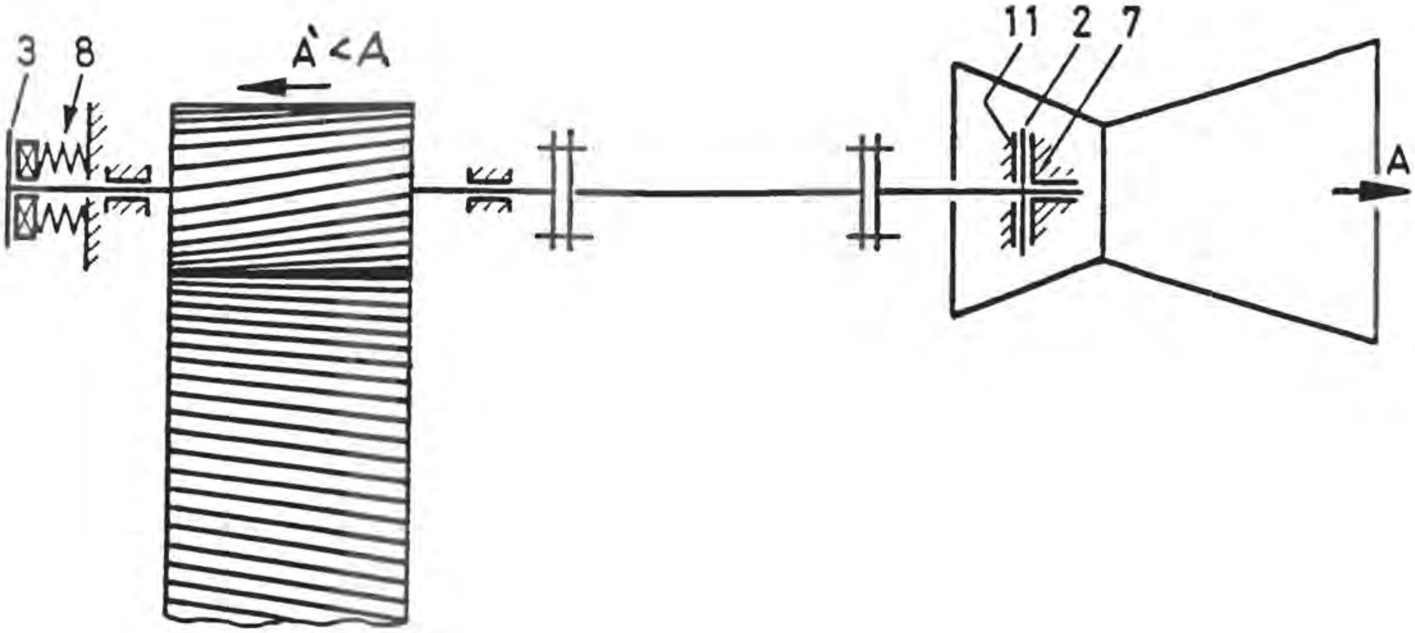

Fig. 9

Double Thrust Bearing Design of Test Gear

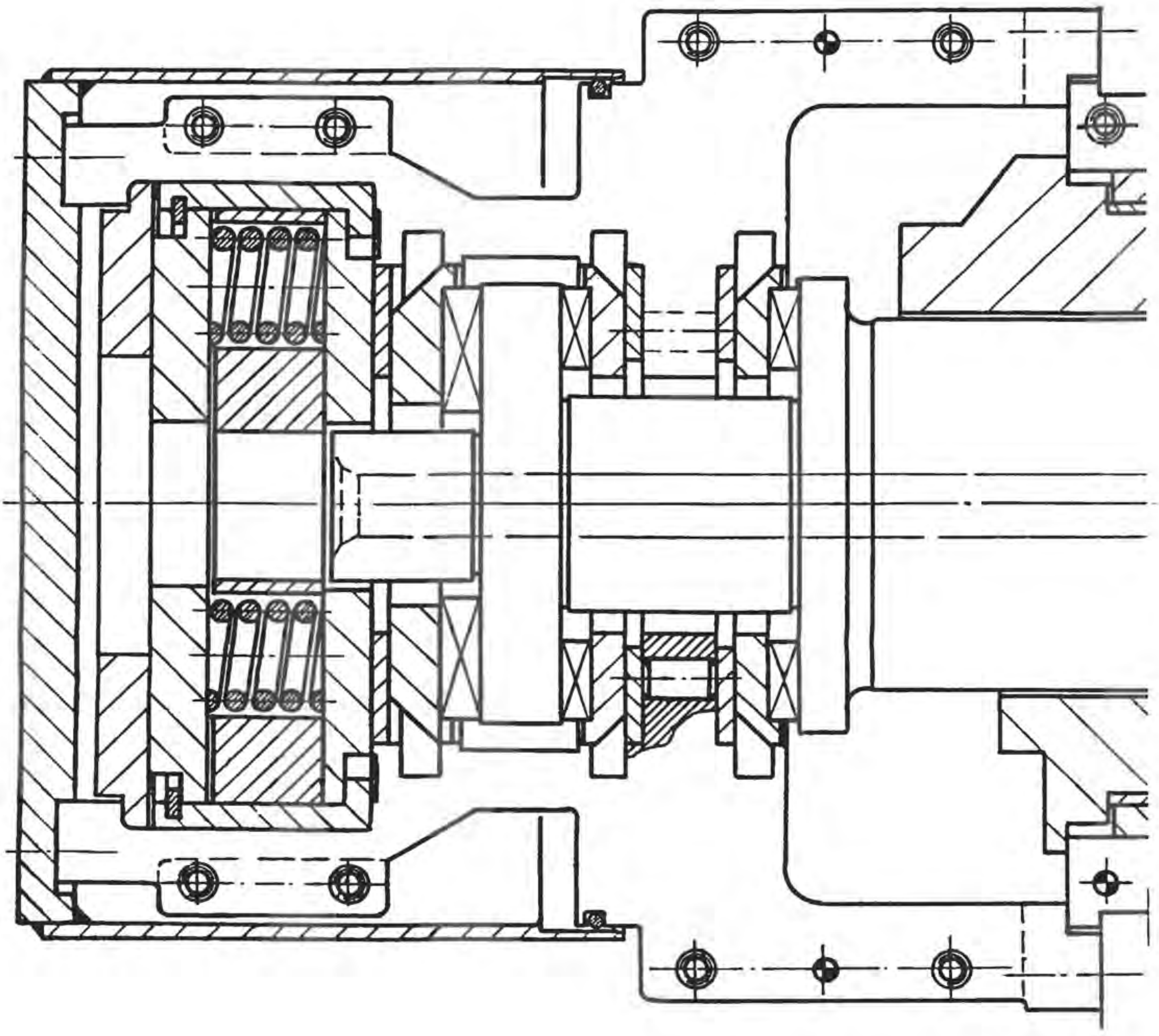

Fig. 10