THE OUTLOOK ON FUTURE DESIGN OF TURBO GEARING

by

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1 Introduction

The question is put:

For which power and speeds can speed increasing and reduction gears be built? Today and in the near future? This study gives an answer to these questions. The accompanying graphs show the max. speeds as a function of transmitted power and with the various influencing factors as parameters. These influencing factors are related partly to the toothing and partly to the high speed bearings. In a simple parallel shaft gear either one, toothing or bearings may be the limiting parameter.

Gears with devided power paths offer interesting possibilities. The speed limit for a given transmission power can be raised substantially and the bearing problem is eased considerably. With other words, the limit is then set by the toothing, its speed, and heat which is developed in the tooth mesh, mainly by the compression of the oil/air mixture.

The gear with power division (Type DTS) which MAAG proposes is described. It offers a power/speed capacity which shall provide the answer to the transmission problem of the compressor builder for many years to come.

Also the simple two shaft gear still has some margin for improvements. A research and development program is under way to improve the design of the tooth corrections and to find ways to increase the load capacity of the bearings.

2 Gears with Parallel Shafts

(Conventional Speed Increasers)

All limiting factors are related to the high speed shaft. Toothing and bearings must be studied separately but they are related to each other in such a way that the condition of one can only be improved at the cost of the other.

Toothing: For optimum reliability the various stresses in the teeth, the temperature variation in the gear rotors, their geometry and cooling must all be carefully balanced. Picking out one item and forcing a limit on it without consideration for the others is bad practise.

The graph Fig. 1 gives the speed limit (rpm) at the high speed shaft as a function of the power (MW). At the time that this graph was made (1984) experience with high power gearing was available up to tooth velocities between 170 and 180 m/s. The graph is plotted for the following parameters:

Tooth velocity	175 m/s
K-Factor	314 (API-SF = 1.4)
Elastic Pinion deflection	20 um
Speed ratio	3

For these conditions the heat developed in the tooth mesh is permissible. This curve represents approximately the state of the art today.

Bearings: The bearings must be able to carry the tooth load at a shaft-speed n1. Specific bearing pressure and bearing temperature must be balanced. Here again, one can only be improved at the cost of the other.

In the graph Fig. 1 a curve for the circumferencial bearing speed of 110 m/s is plotted. It is based on a specific pressure of 30 N/mm^2 and a bearing width/diameter ratio of 1.0. Today, these conditions are considered safe limits for tilting pad bearings.

It should be noted that for a tilting pad bearing with twin-pads (main pad with overall width/diameter ratio of about 1.4, divided into two halves by a grove) the load can be increased which lowers the bearing circumferential speed quite a bit, thereby increasing the limit, in certain regions, for which such gears can be built.

Fig. 2, 3, 4 and 5 show the influence of the major factors which determine the power/speed limits.

These factors are:

- Tooth velocity
- Elastic pinion deflection

- K-factor
- Speed ratio
- Bearing velocity
- Bearing width/diameter ratio
- Bearing pressure

3 Gears with Split Power Paths

A number of designs exist where the torque is transmitted via two or more paths. But for the high power and speeds and the small speed ratios (which is normally the case in gasturbine driven compressor plants) some of these designs must be disregarded at the outset. Costs and gear losses eliminate some other designs. Finally, accessibility of the gears and bearings is also a factor to be considered.

The major representatives of this class of gears are briefly described and their suitability discussed:

3.1 Epicyclic Gear

With fixed annulus.

With three or more satellites.

Limits: The minimum speed ratio is about 3.2. The centrifugal load on the satellite bearings is excessive at high power and speeds. For this reason this gear cannot reach the power/speed limits of the parallel shaft.

3.2 Star Gear

Same principal design as the epicyclic gear but with fixed satellite carrier and rotating annulus:

Limits: The minimum speed ratio is about 2.2. The maximum practical peripheral speed at the toothing is 110 (120) m/s in order to keep elastic growth at the annulus within acceptable limits.

As far as the power and speed limit is concerned, the simple parallel shaft gear is superior.

3.3 Coaxial Gear with three Layshafts

Power split in three paths.

Advantages:

No radial load at input and output shaft bearings. High power and speed capacity; superior to gear with two parallel shafts.

Disadvantages:

The two tooth loads at each layshaft act in opposite directions and thereby tilt the layshafts. Quillshafts between high speed gears and low speed pinions are recommended. This requires a total of 16 bearings which means high losses (double reduction) and high costs. Accessibility is very poor.

3.4 Coaxial Gear with two Layshafts

(Locked train arrangement)

Power split in two paths.
Load distribution by quillshafts.

Advantages:

Radial loads at input and output shaft bearings are low. All gear elements, and arrangement, are conventional. Power and speed capacity superior to parallel shaft gear. Excellent accessibility.

Disadvantages:

Relative large number of bearings: 12. High loss (double reduction), high costs.

3.5 Locked Train, Single Stage with Idlers

Type DTS

Power split in two paths.

With a self-adjusting pinion to assure an even load distribution (Patented design).

Advantages:

Small radial loads at high speed shaft bearings. Power and speed capacity superior to the gear with two parallel shafts.

All gears in one plane.

Total number of bearings: 8

Losses relatively low (single reduction with idlers). Lowest costs within the family of gears with split power paths.

Accessibility to high speed and idler shaft bearings is excellent.

Disadvantages:

Accessibility to low speed shaft bearings poor.

This DTS gear is an excellent solution to raise the revolutions at a certain power, above the speed which is considered safe with the one stage, parallel shaft speed increaser. In other words the merits of this design is that it can be utilised to build a transmission with more conventional specific loads and circumferential tooth speeds. Especially the high speed bearings are no longer a problem or, as in some cases, the limiting factor.

4 Design Features of the DTS Gear

Fig. 10 Principle of the DTS concept.

Fig. 11 Rotor plan, Load Vector Diagram

Fig. 12 Cross Section

Fig. 13 Pinion bearings (4 LOBE)

The speed ratio between gas turbine and compressor is small. There is no need to apply more than one stage in the speed increaser.

However, gears with multiple power path require either idler gears (epicyclic; DTS) or double reduction gears. The best solution is the DTS-gear (see chapter 3), the only concept which is not epicyclic and has only a single reduction.

The input and output shafts are preferably vertically offset. The low speed and the idler shafts are supported by multilobe bearings. The high speed shaft has tilting shoes.

Both idlers have thrust cones which can in addition be designed to transmit the compressor thrust to the low speed shaft. The gear set is axially located by a tilting pad thrust bearing at the low speed.

This type of gear requires a special feature to provide equal load sharing between the two power paths. The pinion is allowed to adjust itself radially but only in a defined direction. This mechanism is best explained by a simple example with spur teeth; like Fig. 10.

It is well known and generally accepted, that the sun pinion of an epicyclic gear has no bearings at all. Fig. 10 a shows the tooth load-vector-diagram of such a gear with three planets. Obviously, the three vectors must be equal as the planets are arranged at 120° to each other.

In Fig. 10 b the top planet is removed and the annulus replaced by a gear with external teeth. In this DTS gear the missing load vector of the top planet must be replaced to keep the equilibrium at the high speed pinion. To do that it is necessary to place two bearings at the high speed pinion. However these bearings must be able to carry a load in one direction only which is parallel to the missing load vector, i.e. at an angle to the horizontal plane which is equal to the pressure angle of the toothing.

As the directions of the tooth loads, at the two tooth meshes, are well defined and can never change, the shape of the vector diagram remains a triangle of equal sides, no matter whether the loads are small or large. In other words, we have a system with equal load sharing in the same manner as we find it in any epicyclic gear with three planets and a free floating sun pinion.

This basic principle of the floating pinion is also applicable for single helical toothing with or without thrust cones. The pinion bearings are very simple. The load is carried by one tilting shoe only (Fig. 11). The pivoting point is located in a plane through the pinion axis at an angle such that the two tooth load vectors W1 and W2 are equal in size.

In order to hold the pinion in place in case of a torque reversal a second tilting shoe is arranged in a similar fashion, i.e. the tooth loads are again equal in size. With the torque acting in a definite direction, only one tilting shoe is under load and the pinion is free to adjust itself, the fulcrum being the pivoting point of the loaded shoe.

Apart from the power being split into two paths there is one more advantage of this design which is equally important. The bearing load at the high speed pinion can be made small simply by designing this gear with a small angle between the horizontal plane and the plane through the pinion and idler axis ( ). Therefore, the bearing load at the high speed shaft is no longer a limiting factor. As a matter of fact, the limit is set now by the permissible torsional stress and the speed in the journal, but both can be kept at safe levels.

5 Power/Speed Capacity of the DTS Gear

Similar n1/N-curves as for the conventional speed increaser are plotted in Fig. 6, 7, 8 and 9. It should be noted that the speed in the high speed bearings is relatively low so that the limits to power and speed are set by the toothing only. The speed ratio between input and output shaft has no influence on the limits. What is important is the ratio between idler and pinion, and this is more or less constant with all gears of this type (i = ca. 1.6).

6 Summary

The power/speed limits of the conventional gear and the DTS-gear are put together in one table (Fig. 14) for easy comparison. These limits are based on today's field experience with high speed toothing and bearings as plotted in the Fig. 1 and 6.

These limits shall be pushed up further with the research and development work in progress. However, considerable efforts shall be required and the steps towards higher speeds, for a given power, shall become smaller and smaller.

GEAR with TWO PARALLEL SHAFTS

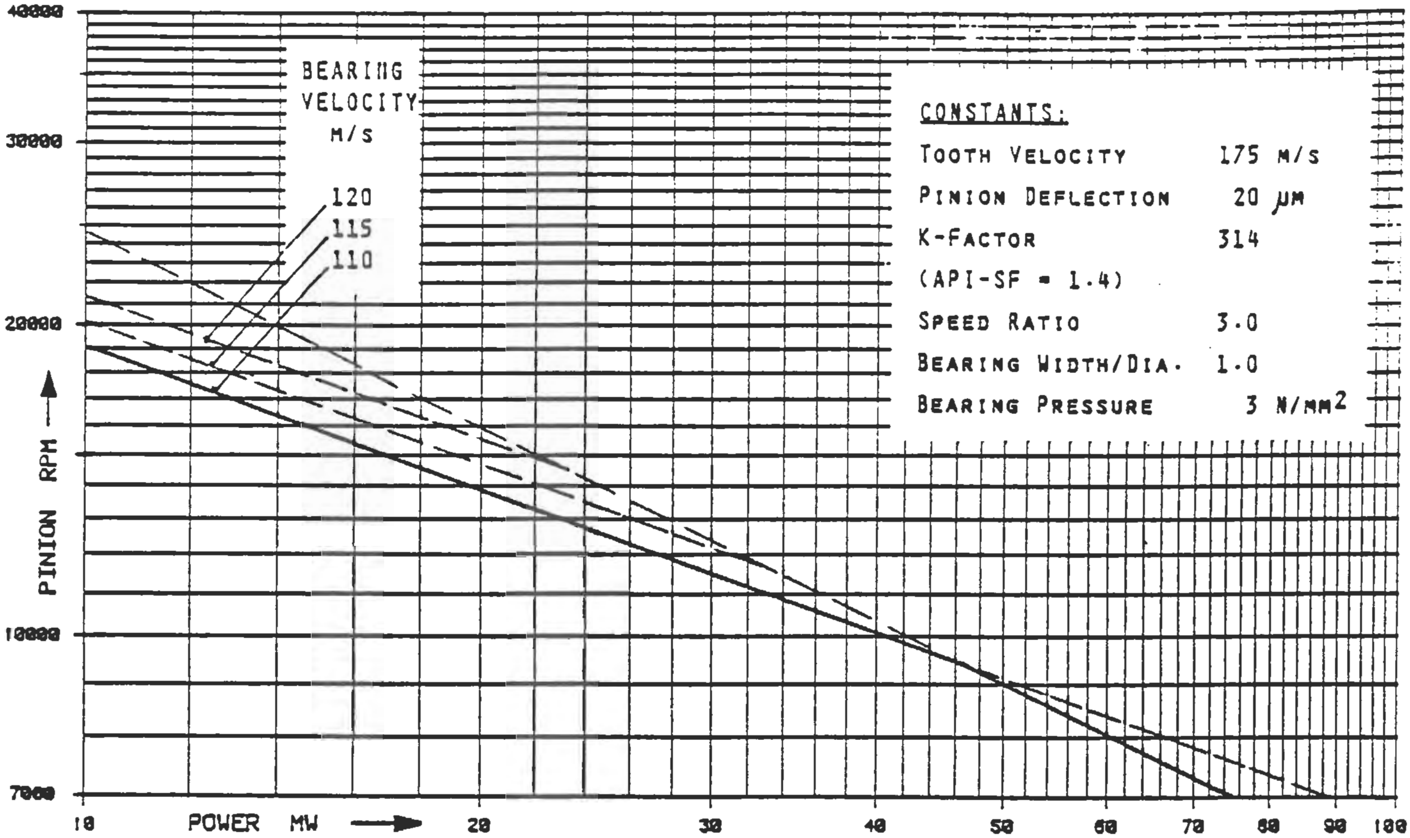
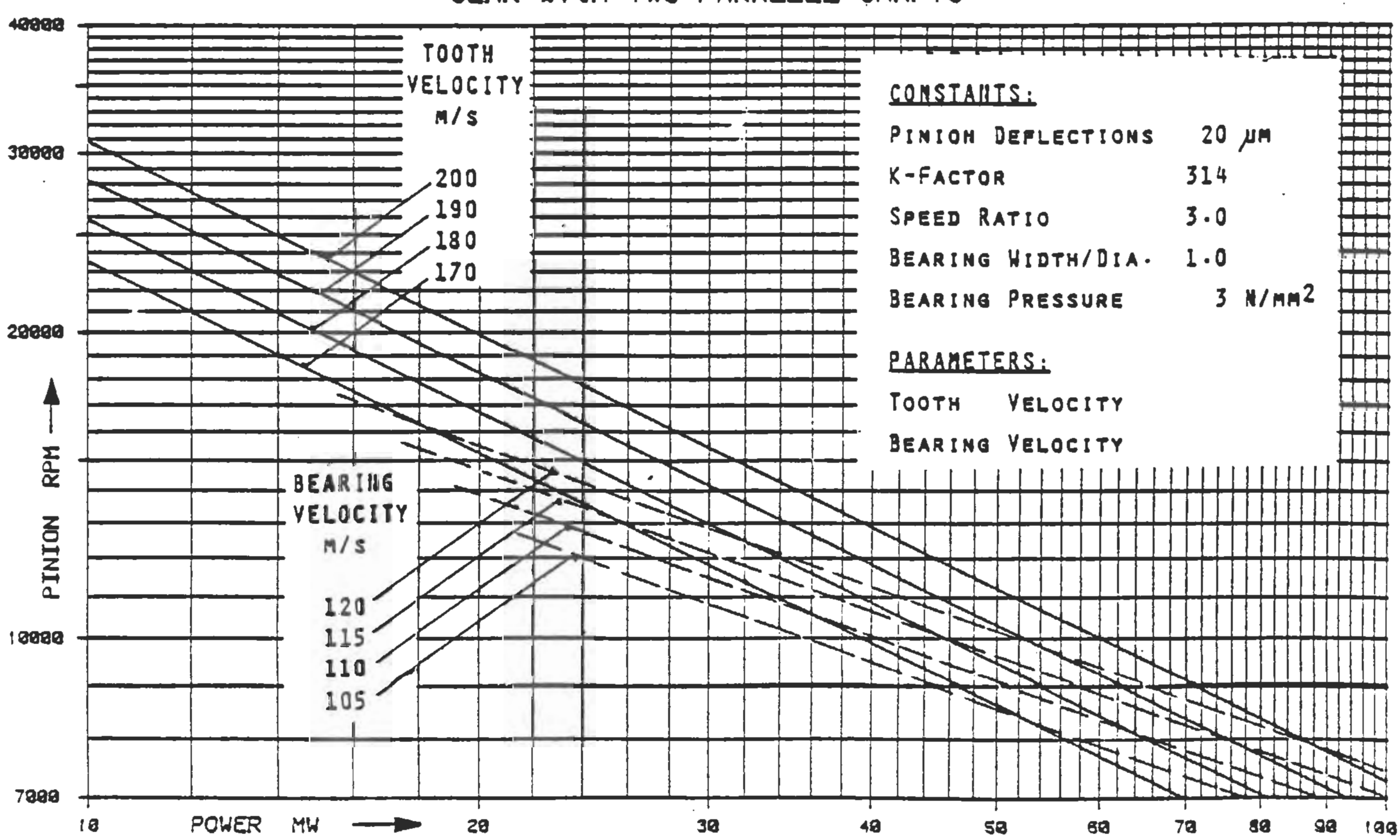


FIG. 1

GEAR with TWO PARALLEL SHAFTS



GEAR with TWO PARALLEL SHAFTS

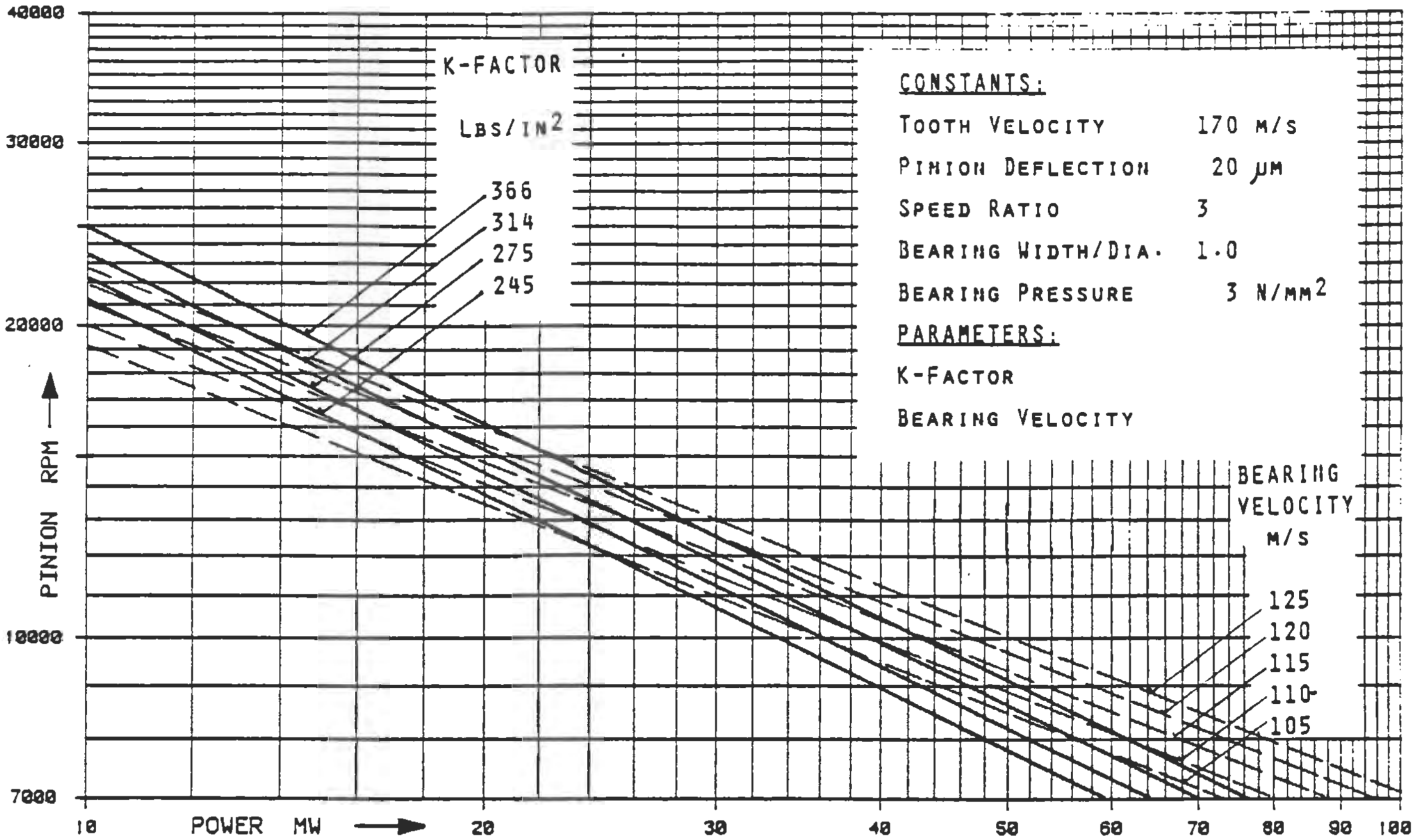
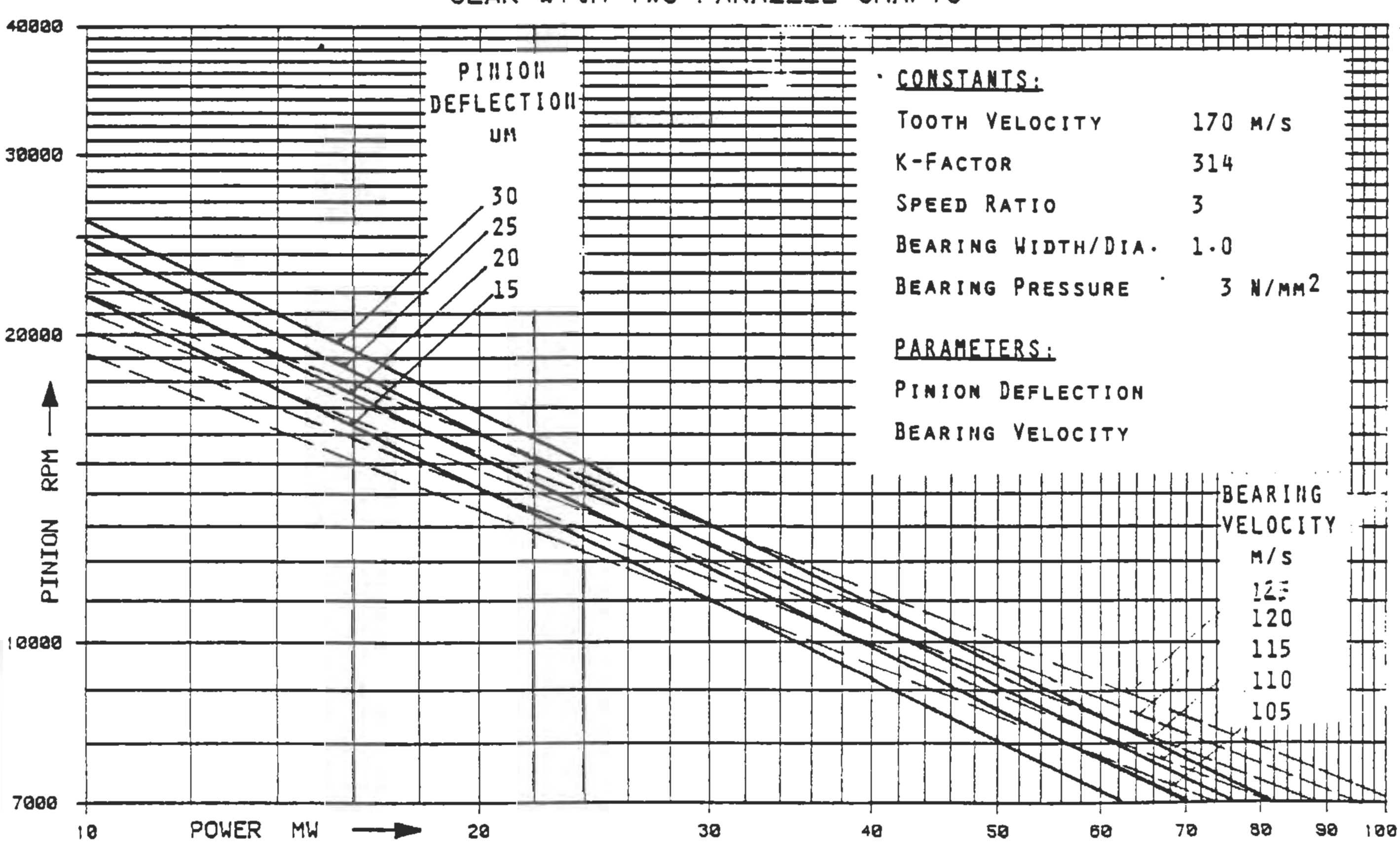


FIG. 3

GEAR with TWO PARALLEL SHAFTS



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GEAR with TWO PARALLEL SHAFTS

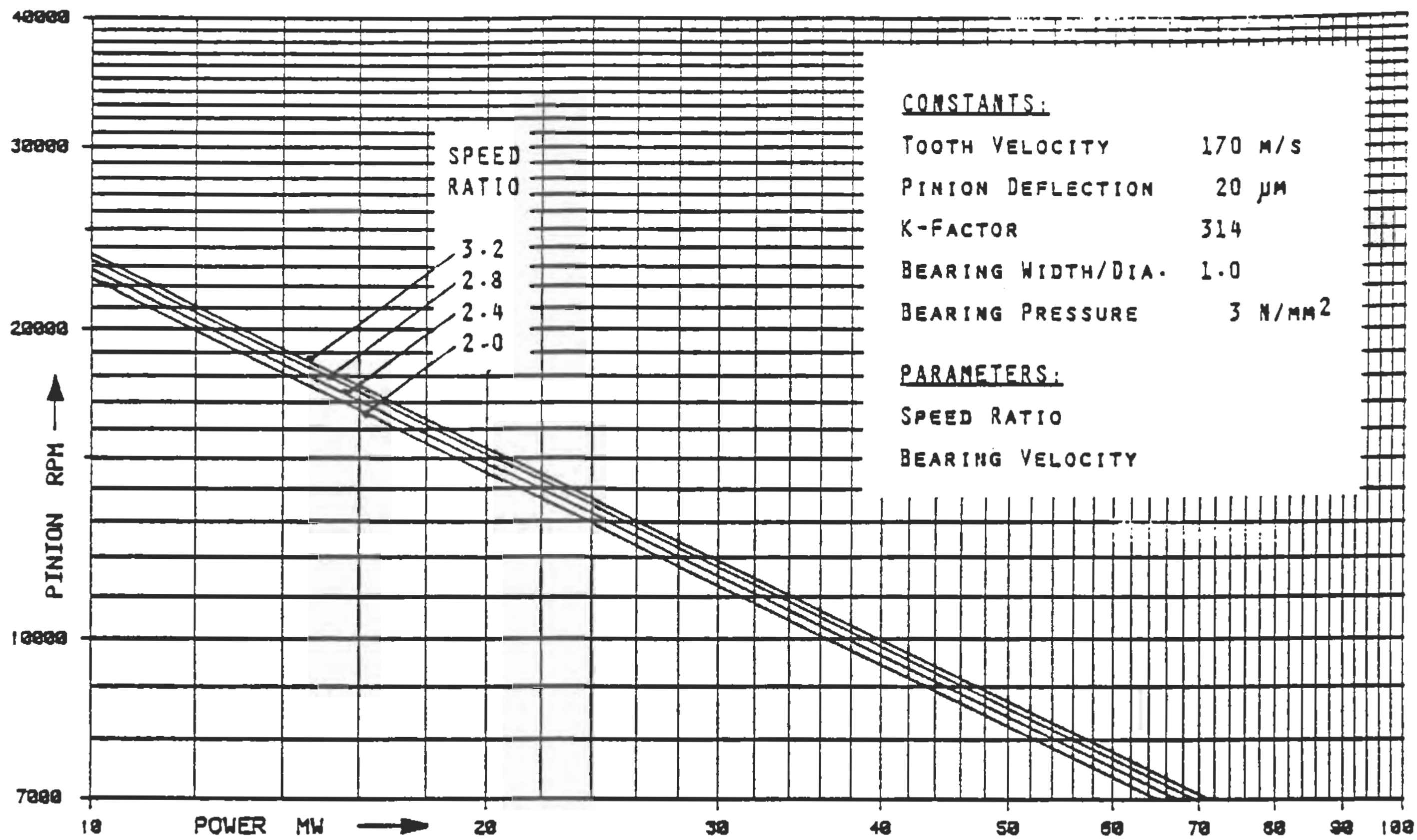
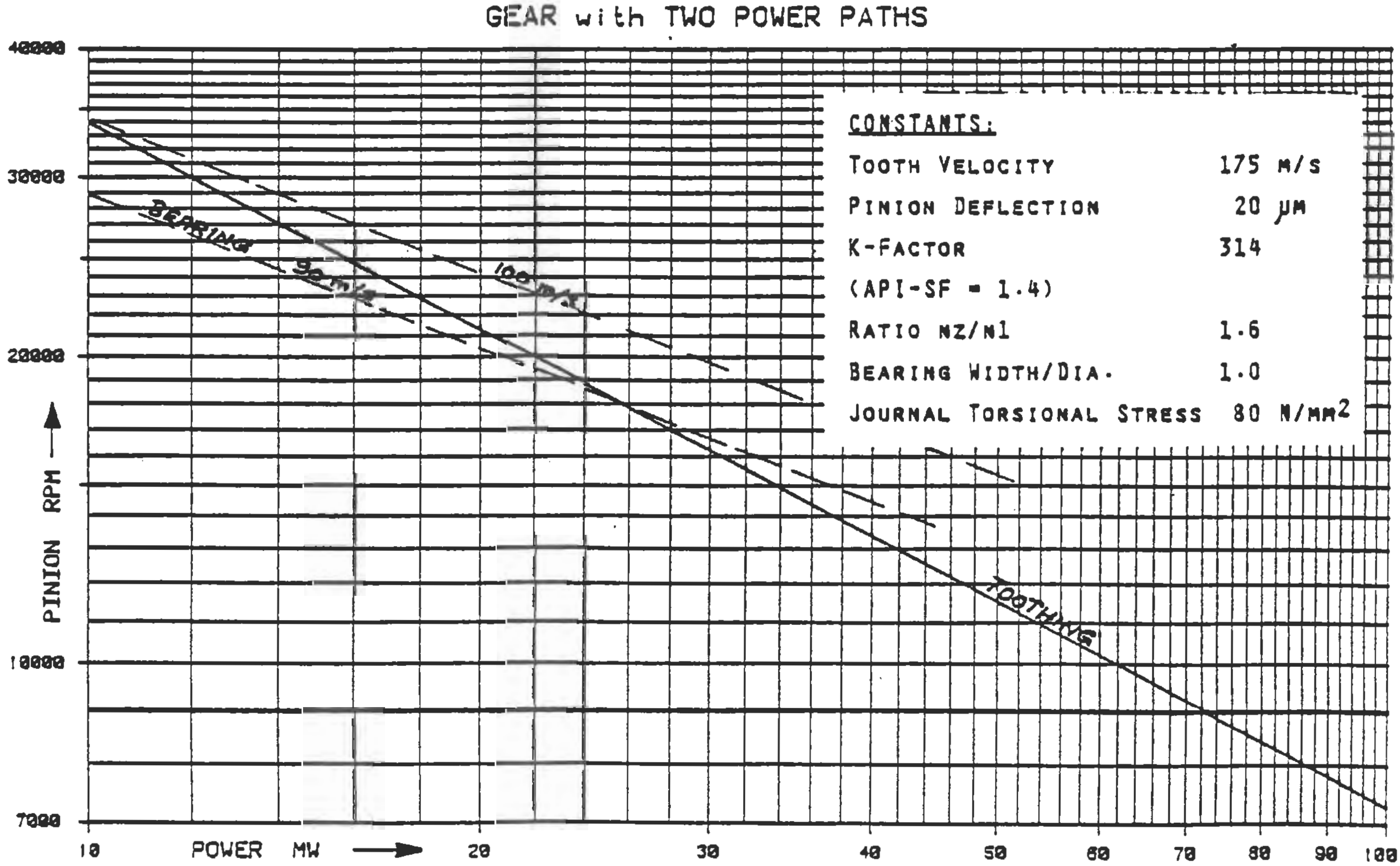


FIG.5

GEAR with TWO POWER PATHS



GEAR with TWO POWER PATHS

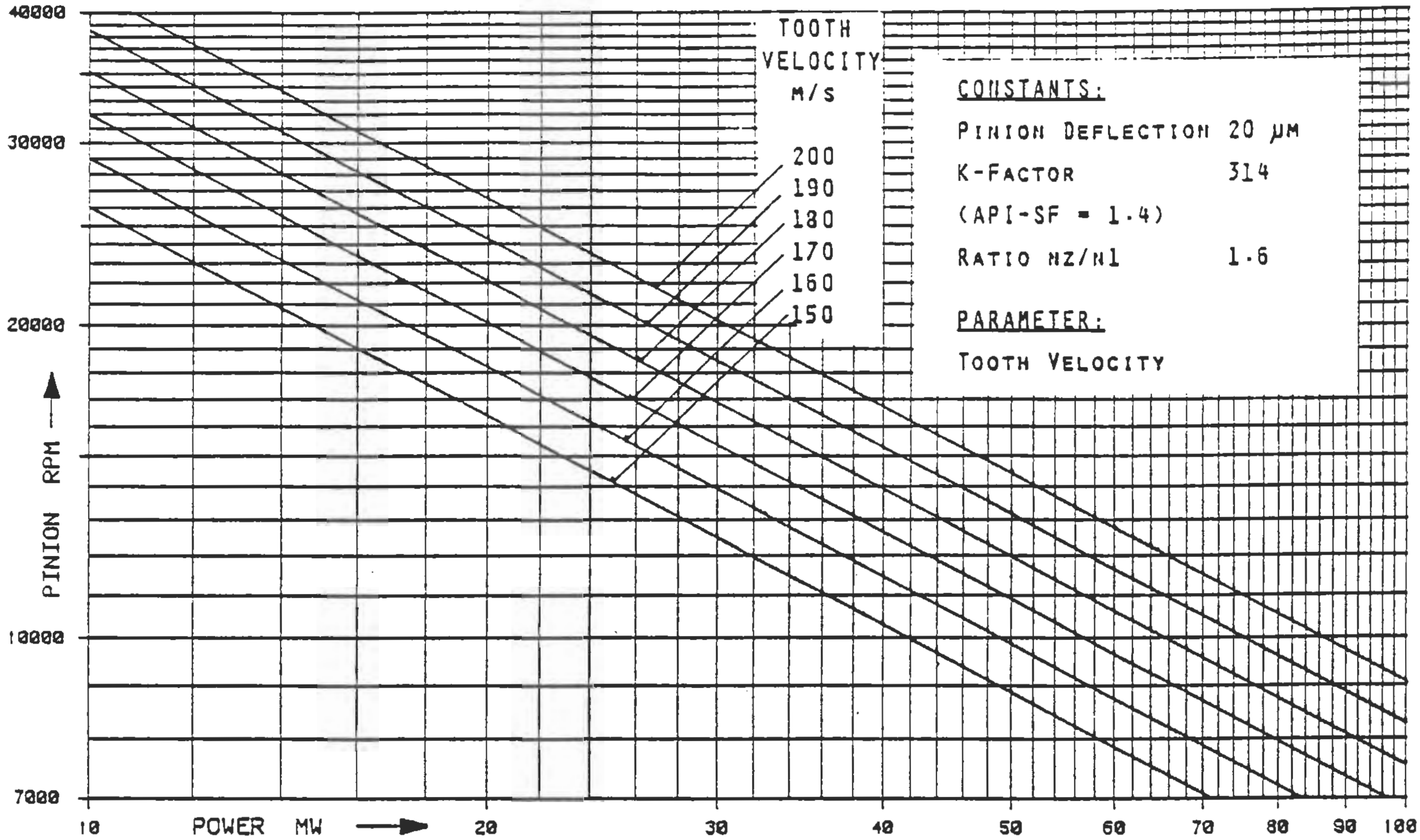
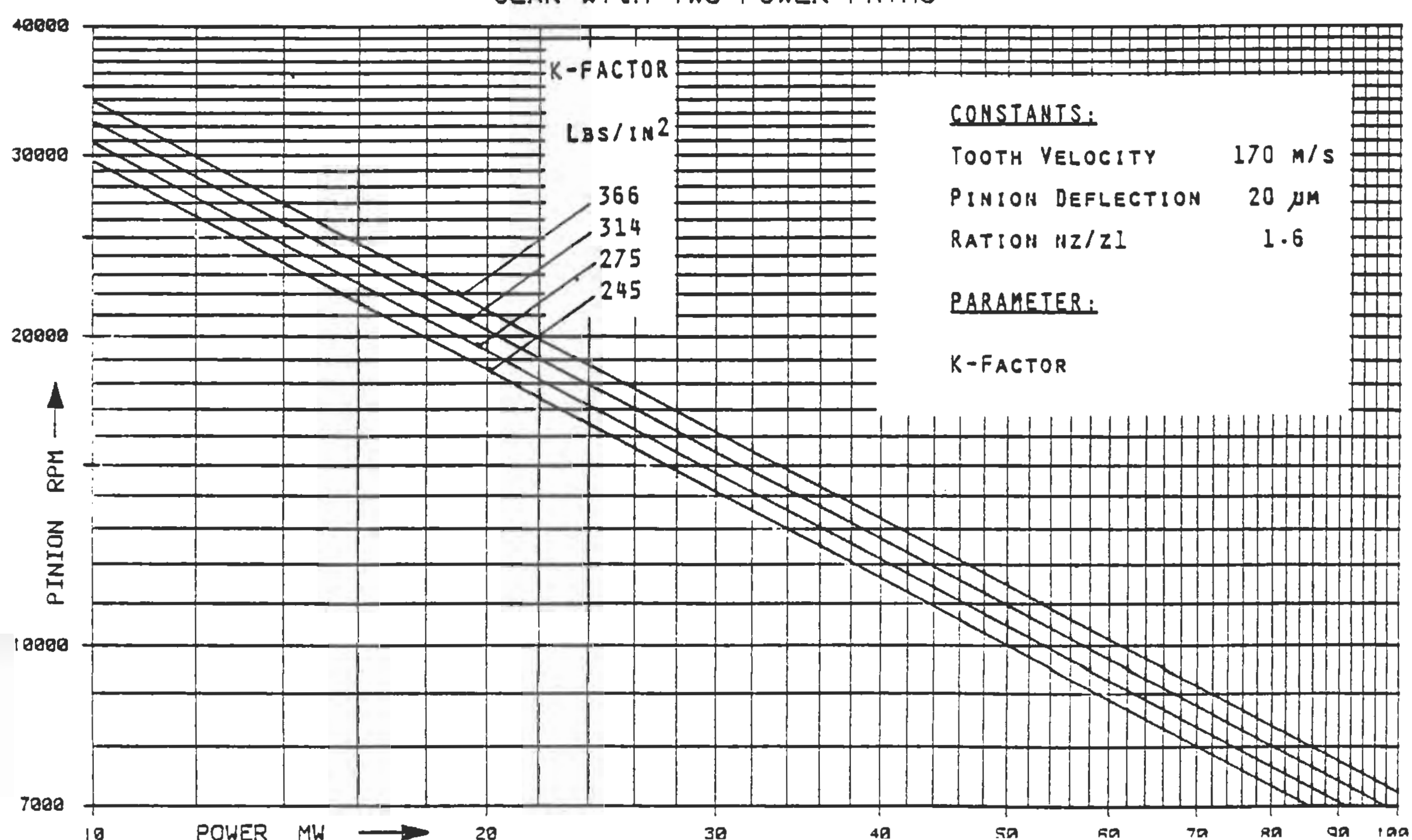


FIG.7

GEAR with TWO POWER PATHS



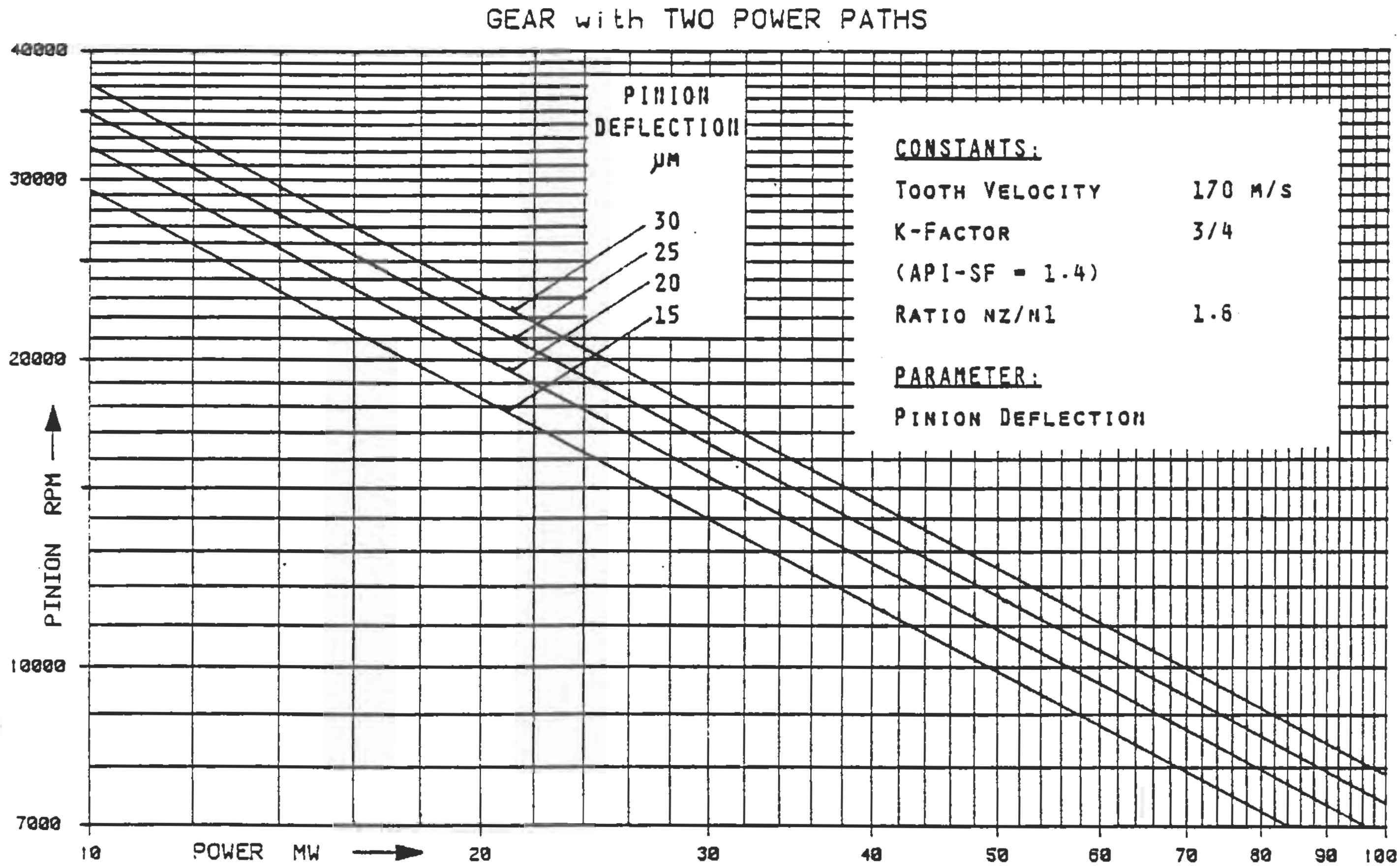
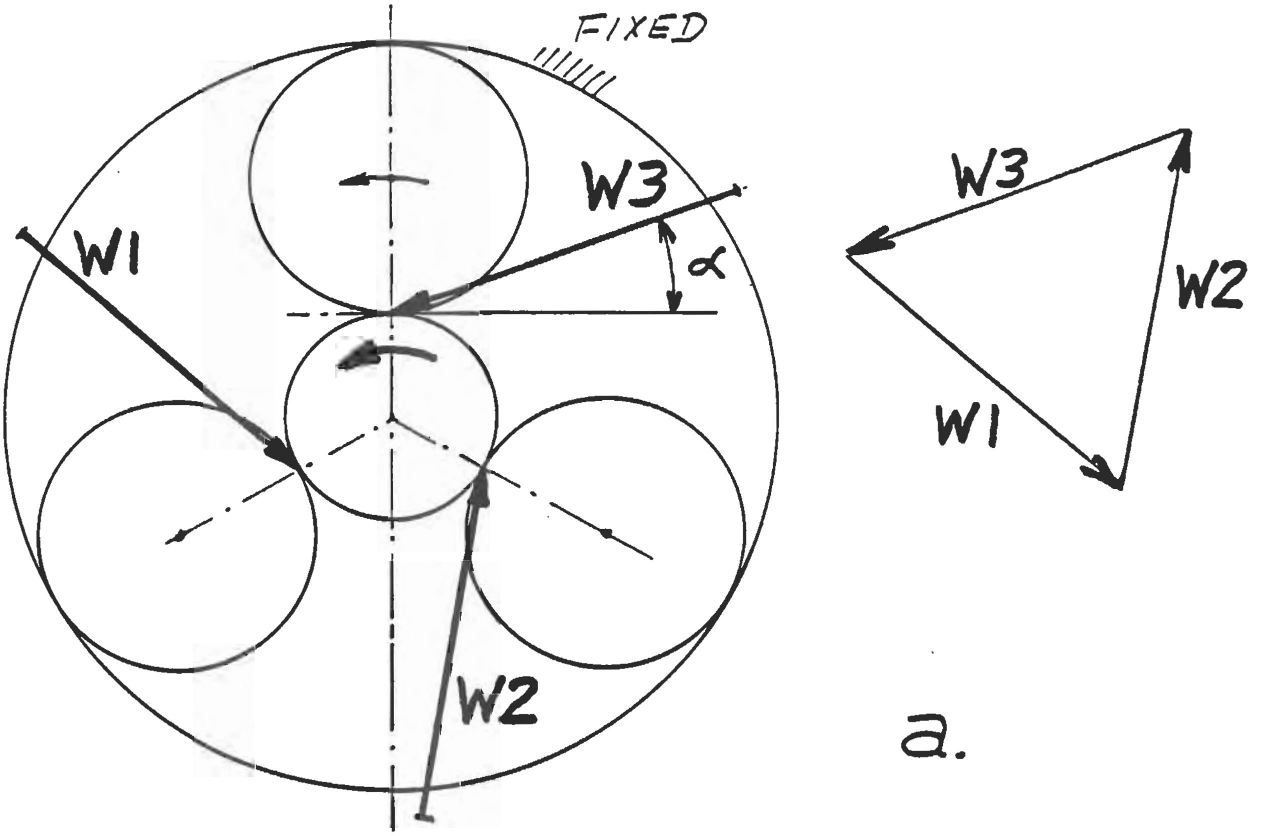

GEAR with TWO POWER PATHS
PINION DEFLECTION µm
30
25
20
15
CONSTANTS:
TOOTH VELOCITY 170 M/S
K-FACTOR 3/4
(API-SF = 1.4)
RATIO NZ/N1 1.6
PARAMETER:
PINION DEFLECTION
40000
30000
20000
10000
7000
PINION RPM
10
POWER MW
20
30
40
50
60
70
80
90
100

FIG. 9



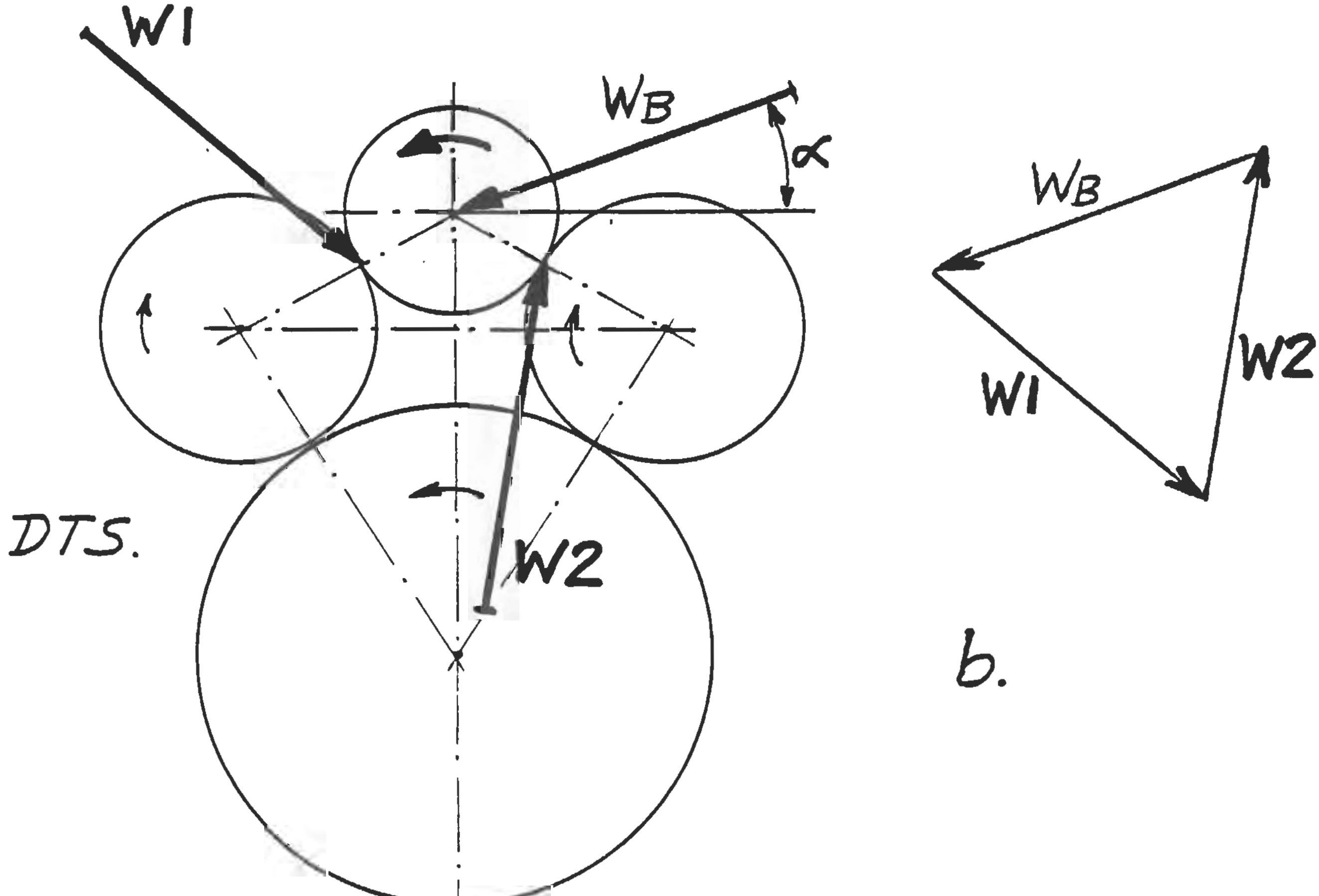


FIG.10

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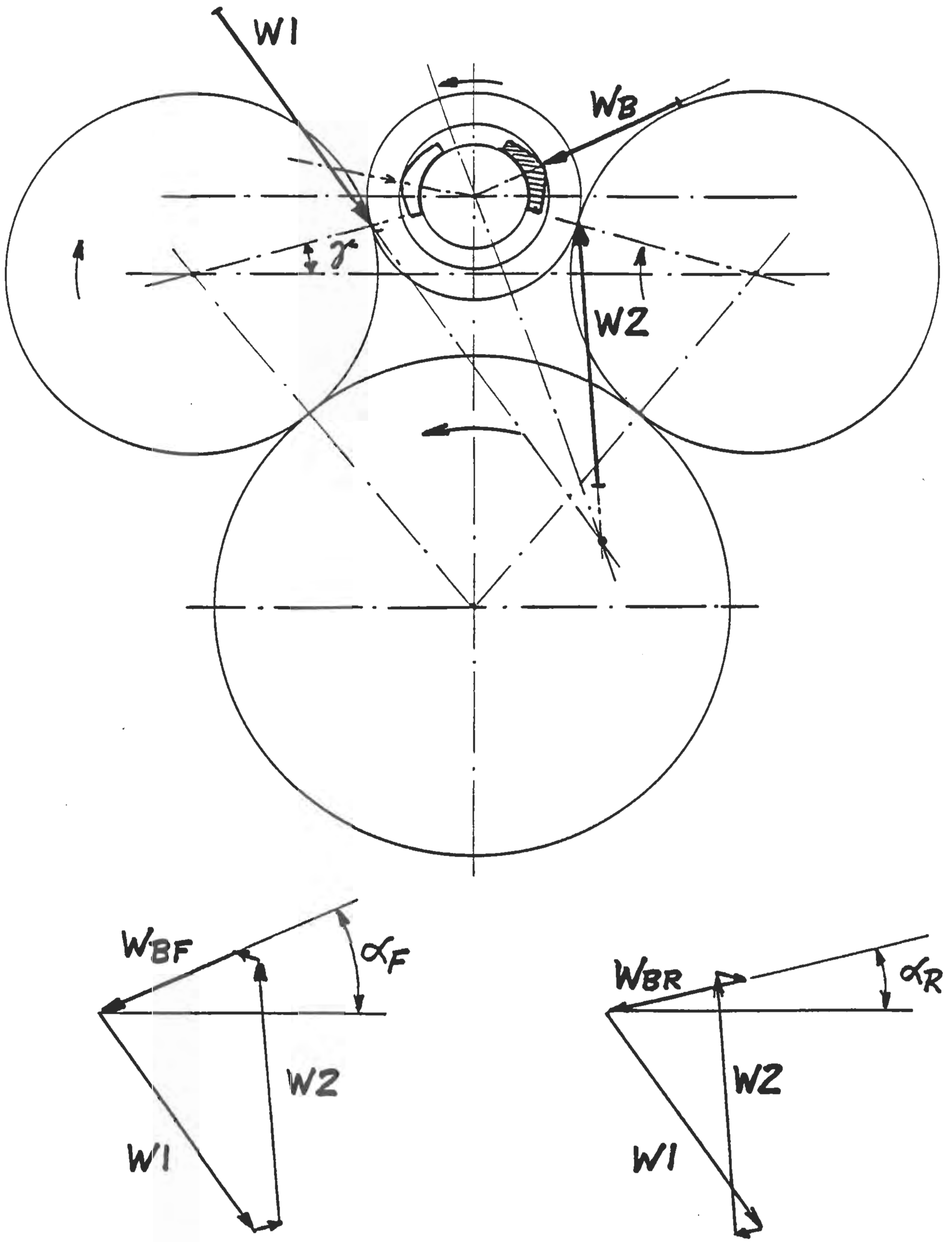

FRONT BEARING

REAR BEARING

FIG. 11

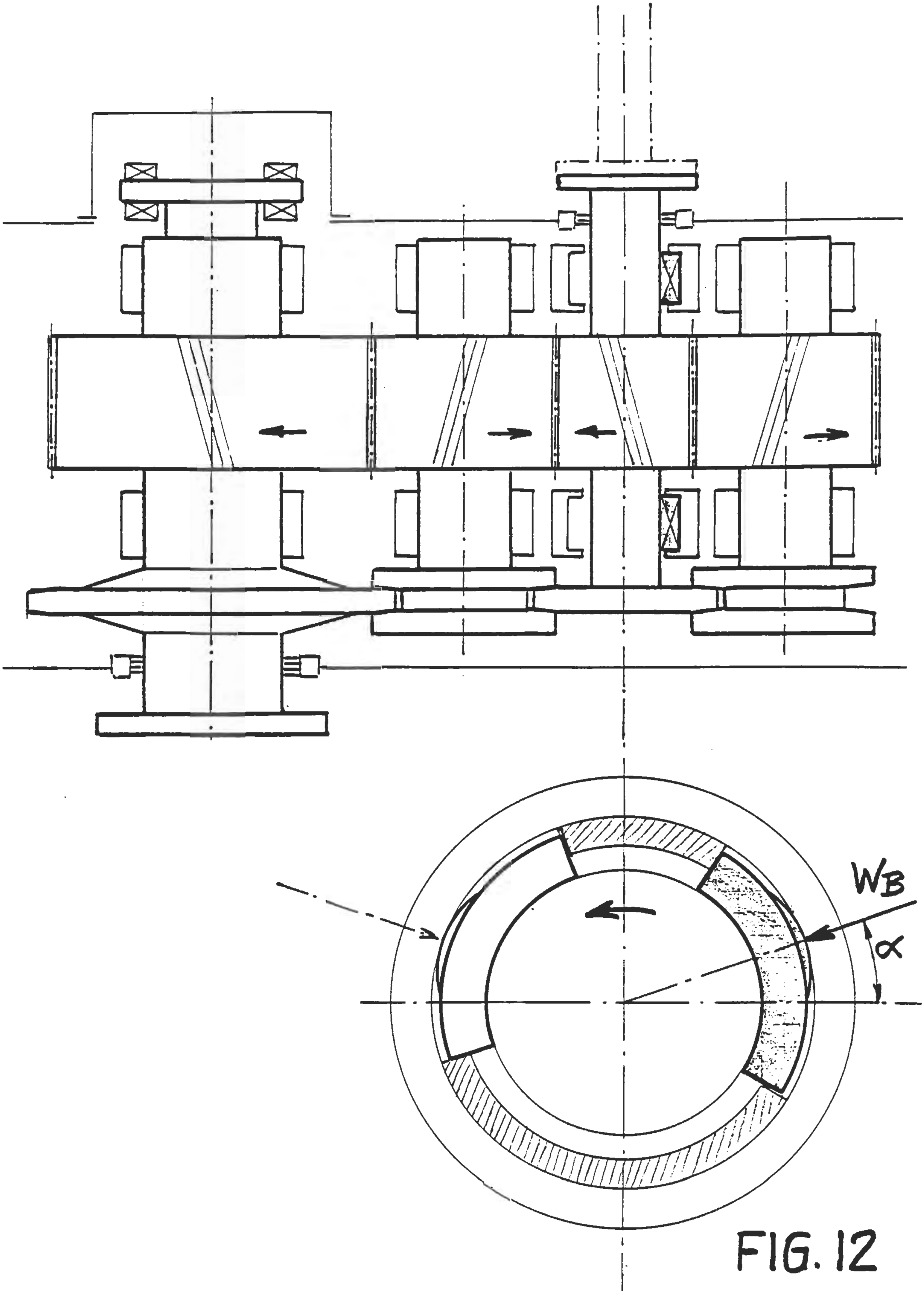

FIG. 12

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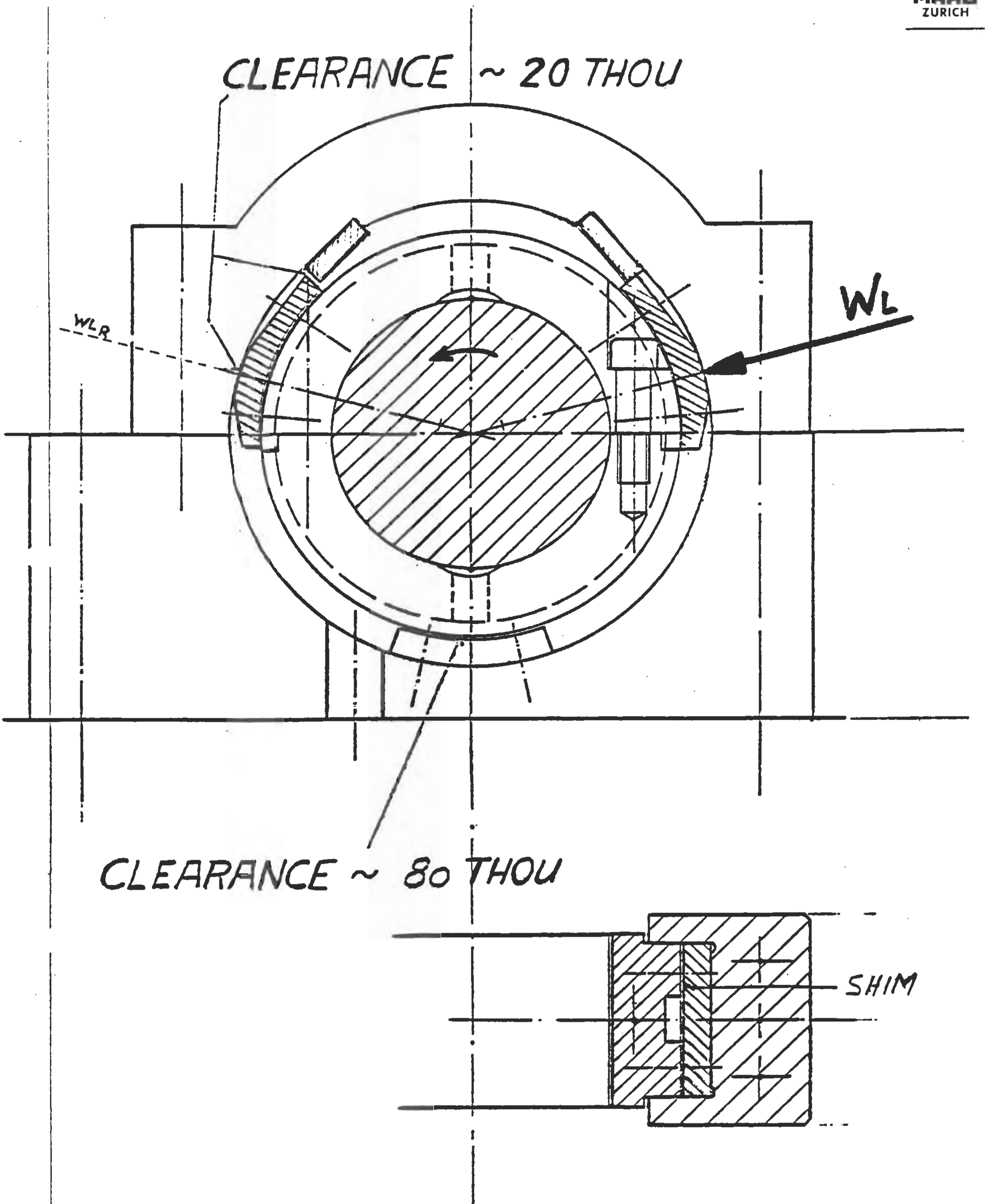

FIG. 13

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POWER	GEAR WITH TWO PARALLEL SHAFTS	GEAR WITH TWO POWER PATHS : DTS
MW	BEARING VELOCITY 110 m/s	BEARING VELOCITY < 90 m/s
	PINION REVS rpm	PINION REVS rpm
15	15800	23600
20	13800	20400
25	12500	18300
30	11500	16200
35	10700	14700
40	10100	13400
50	9000	11500
60	8000	10200

PARAMETERS:

TOOTH VELOCITY	175 m/s
ELASTIC PINION DEFLECTION	20 μm
K-FACTOR (LLOYD'S)	3/4
SPEED RATIO	3.0

FROM FIG. 1 and 6.

Fig. 14