TOOTHED COUPLINGS

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Notation:

Symbol	Unit	Term
A	m	Distance of the tooth to the slewing axis
b	mm	Effective face width of tooth
C	-	Constants
d	m	Pitch circle diameter
D_a	cm	External diameter of spacer
D_i	cm	Internal diameter of spacer
E	m	Eccentricity of floating member
e	um	Eccentricity of mass
F_R	N	Radial force
F_S	N	Friction force, coupling thrust
G	kg	Weight
G_H	kg	Weight of floating member
H	m	Axial distance between the teeth of the floating member
h'	mm	Common tooth depth
L	mm	Axial clearance
Mb	Nm	Bending moment
M_R	Nm	Friction moment
m_H	kg	Mass of floating member
n	min^{-1}	speed
P	kW	Power
P_a	N/mm^2	Allowable surface pressure
T	Nm	Transmitted torque
U	gmm	Residual unbalance
v	m/s	Mean sliding velocity
x	m	Distance between load and critical section
z	-	Number of teeth
α	degree	Angular misalignment
ε	-	Conti-Barbaran value
μ	-	Coefficient of friction
δ	N/mm^2	Tangential stresses
ω	s^{-1}	Angular velocity

1 Introduction

Shaft couplings play a major role as torque transmitting elements in the field of mechanical engineering. Of the numerous types of coupling encountered, in this wide domain, the toothed coupling is well established, particulary in power machine engineering. Fig. 1 gives a summary of the different coupling designs together with their characteristics (10). In this table the application possibilities for toothed couplings and their offshoots are indicated.

2 General Remarks Basic Considerations

2.1 Purpose

The purpose of the toothed coupling is the connection of two independently supported shaft systems and the transmission of a torque, where radial and axial shaft displacement may occur during operation.

2.2 Requirements

The requirements which toothed couplings have to meet are:

a) During transmission of the torque no wear shall occur on any coupling component.

b) The toothed coupling shall not create external forces

c) During operation the toothed coupling shall not excite vibration.

2.3 Behaviour

To establish how far the toothed coupling can cope with its requirements several theoretical considerations have to be reviewed. The following statements are of general validity. They are independent of type and make of the toothed coupling.

2.3.1 Design

The toothed coupling (hereafter called simply-coupling), consists fundamentally of 3 components i.e. 2 hubs (rigid members), which are mounted on the shafts to be coupled and one floating member. The hubs may have external or internal teeth, the floating member is then designed accordingly, see Fig. 2.

The various designs will be dealt with in a following chapter. For the teeth themselves the involute tooth form with a pressure angle of 20°-25° has proved to be advantageous. The teeth are barrelled in longitudinal direction to avoid corner loading when the coupling is misaligned, see Fig. 3 (the amount of barrelling is exaggerated for clarity). The toothed components may be designed such, that one member is centered in the mating member. To achieve the centering, it is general practice to raise the tip of the external teeth, over part of the tooth width, and to manufacture the root of the internal teeth circular to the sleeve axis. A small clearance will be made between tip - and root diameter. This design is known as "tip-centering".

When determining the clearance for the tooth mesh and the tip centering, the following is to be considered:

- Assembly
- Freedom of movement for coupling components while misaligned
- Expansion due to shrink fits
- Different expansion of the coupling members during operation.

The floating member of sufficiently loaded couplings will always be centered by the tooth flanks due to the torque, whether the coupling has an additional centering device (i.e. tip centering) or not. An additional centering may be of advantage for:

- Lightly loaded couplings.
- Couplings which are used in installations with which a string test at full speed and partial load is carried out.
- Couplings in installations which will be frequently run-up to speed, without load.

- Couplings which have to operate safely during a certain time with partially worn off tooth flanks (this requirement applies mostly for marine installations).

Tip centering for couplings is required by certain specifications. However, it must be born in mind that, especially for high speed couplings, the centering diameters have different expansion due to the different tangential stresses in the coupling components, due to rotation which will result in increased tip clearance (1,7).

2.3.2 Kinematics of the Toothed Coupling

The teeth have to transmit the rotation and the torque. The kinematics primarily concern the transmission of the rotation in a misaligned coupling. This subject is very academic and is of wide scope (6). This chapter considers only simplified the relative movement which occurs between the mating teeth of a misaligned rotating coupling. Fig. 4 shows that, during one revolution, a relative movement occurs between the flanks of a pair of teeth. In two positions, diametrically opposed, the tooth of the sleeve performs a pure slewing movement. In two other positions, rectangular to the slewing position, the tooth of the floating member tilts on the tooth of the hub. In all intermediate positions the slewing and tilting movements are superimposed. For the slewing movement a mean sliding velocity can be defined as:

$$V = \frac{2 \cdot d \cdot tg\alpha \cdot n}{60}$$

Tests (3) have shown that if the mean sliding velocity continuously exceeds approx. 0.12 m/s surface failure may occur.

2.3.3 Lubrication

A coupling in service encounters always a certain misalignement, this means that there is permanently relative movement between the teeth of the hub and of the floating member. Therefore the coupling has to be lubricated. One distinguishes two methods of lubrication:

a) Permanent grease or oil filling

Couplings with grease or oil filling are used for low speed only. The limit for grease packed coupling can be generally fixed to approx. 40 m/s peripheral speed of the pitch circle diameter (1). For oil filled couplings the limit is given by the tightness of the seals during operation and by the centrifuging of the oil (additives and impurities).

The operational reliability of the seal depends on its design, a general limit can therefore not be given. The beginning of centrifuging depends largely on the acceleration a_c due to rotation, acting on the particals in the oil. Centrifuging starts when a_c exceeds the value of approx. 4.5×10^4 m/s^2.

$$a_c = \frac{d}{2} \cdot \omega^2$$

With increasing value of a_c the oil of the coupling has to be changed more frequently (9).

b) Continuous forced lubrication

For high speed couplings the lube oil must be supplied continuously. Essentially, 3 different arrangements are used, see Fig. 5-7.

The arrangement according to Fig. 5 has the advantage that the teeth are constantly submerged in an oil ring which is generated due to the rotation. A small oil flow is sufficient for lubrication. The disadvantage is that impurities may be centrifuged; in this respect the coupling behaves like an oil filled coupling.

The arrangement according to Fig. 6 is used in high speed applications. The advantage is that impurities are collected partly in the annulus. The hole and a certain area around it remain clean because the impurities are carried away by the oil due to its high velocitiy in the hole. The area around the teeth remains clean for a relatively long period. A further advantage is that if the oil contains water (steam turbine drives), this will be drained continuously. The disadvantage of this design is the high

3.2.1 Type Designation

The type designation consists of a group of 3-4 letters, which represent the type, and a number, which defines the size of the coupling.
The first letter Z stands for toothed coupling.
The letter or group of letters in the middle of the designation defines the lay-out and function, whereby several combinations are possible.

A Spacer type coupling, external floating member made up of 3 components, sleeve with integral end covers.

B External floating member made up of 2 sleeves.

C Couplings for high torque and low speed.

E Spacer type coupling, internal floating member (Marine type)

L Spacer type coupling, external floating member made up of 3 components, with removable sleeve.

N Standard type with single piece, external floating member having standard length.

Q Shear pin coupling with single piece, external floating member.

R Spacer type coupling for vertical arrangement.

T The floating member is a solid shaft with 2 standard hubs. For low speed applications only.

U Spacer type coupling, external floating member made up of 3 components, teeth located at far end of the hub (low moment).

V Coupling with single piece, internally toothed hub.

X Coupling with limited axial float.

The last letter of the designation indicates the lubricating system.

D Individual lubrication of each tooth, so called "anti sludge" lubrication, see Fig. 6.

F Oil filled or grease packed coupling.

S Continuous lubrication (oil bath), see Fig. 5.

The notation ZF or ZS is used for N-type couplings but with floating member which is longer than the standard length.

3.2.2 Summary of Different Coupling Types

Sketches and data sheets of the different types of couplings are shown in a separate catalogue on couplings.

3.2.3 Design Details

1) Oil filled or grease packed couplings.
All the bolted joints are sealed with an O-Ring (static seal).
The seal between the fixed member (hub) and the floating member (sleeve) is also an O-Ring but with a larger section, giving greater flexibility, so as to absorb the small relative movements between hub and sleeve (dynamic seal). The oil bath which is generated by the rotation inside the sleeve does not usually extend to the dynamic seal. The sleeve has two holes for draining and filling.

2) Centering of bolted flange connections.
Two methods are known for centering the components of flanged connection.

   a) Spigot centering. The bolts are fitted in one component and have clearance in the other one. The advantage of this design is easy assembly. The disadvantage is that the spigot diameters have to be adjusted to each other to avoid either too much clearance or too much interference of the spigot. This entails problems in production and stocking. This design may be applied for single piece production.

b) The MAAG method is to center the components by closefit bolts. The bolt holes and the bolts are manufactured independently according to drawing. This method guarantees that the connection is free of clearance and that the components are fully interchangeable.

3) Limited axial float.
Couplings with limited axial- (or end-) float are used for connecting two shafts one of which has no thrust bearing. This shaft is axially located by the thrust bearing of the other shaft, by means of the coupling with limited axial float. The main application of this coupling is with electrical machinery where the axial force is small compared with the tangential force.

To limit the axial clearance of the floating member different designs are applied, depending on the coupling type. The limiting clearances are indicated in the following figures by L/2.

a) Types ZNXS, ZNXF
This design is used for the N-type coupling with standard sleeve length only. The contact area of the domed covers is surface hardened. The covers are bolted to the shaft end.

b) Types ZLXS, ZLXF, ZXS, ZXF, ZQS
The design conforms to Fig. 14 and consists of a ring nut which is secured by a dog point screw, a single piece distance ring, and an end cover.

c) Types ZAXS, ZAXF
A limited axial float for these couplings is achieved by means of a ring which is placed into the sleeve see Fig. 15.
This design may also be applied for ZLXS and ZLXF couplings.

d) Type ZEXD, ZUXD
The design is shown in Fig. 16. The buffer points are the spool piece or sleeve and the snap rings.

In all cases the hubs are to be fixed to the shaft ends. For cylindrical shaft ends, fixture is by means of a cover which is bolted to the shaft end. For tapered shaft ends the hub is held by a nut.

For all the above mentioned designs the total axial float is

$$L_{tot} = 2 \cdot L$$

To avoid jamming of the coupling, the clearance L_{tot} must be equal or larger than the values given in Fig. 17.

4) Shrink fits.

For shrink fits where the toothed hub is mounted by applying heat the following temperature limits must be considered when determining the amount of interference:

210°C for hubs having induction hardened teeth

280°C for hubs having case hardened teeth

400°C for hubs having nitrided teeth

For hydraulically fitted shrink fits, any of the hardening methods may be used. Further information is to be found in the publications of SKF (9).

5) Electrical insulation.

In installations with electrical machinery it is sometimes necessary to insulate the coupled shafts. By using a coupling, with an internal floating member (E-Type), the insulation material is placed between the flanges for hub and sleeve, and around the bolt, see Fig. 18. Preferably, the insulation is arranged on the hub of the shaft of the electrical machine.

3.2.4 Selection of Type

For the selection of the appropriate coupling type the Fig. 18 may be helpful. It shows the logical sequence of the considerations to be made.

3.3 Disengageable Toothed Couplings

These couplings are applied for connecting shafts which may be uncoupled during operation.

3.3.1 Non-automatically Engaging Couplings

These couplings are installed in shafts systems to disconnect one or more machines from the shaft system which remains in operation. To disengage the coupling the torque must be reduced to approx. 10% of its nominal value. Special designs are available to disengage the coupling at higher percentage of nominal torque. For re-engagement of the coupling the shafts to be coupled must be at standstill.

The disengaging couplings are available in two different designs:

a) Type ZSP, with external floating member and continuous lubrication; applied for low speed.

b) Type ZEP, with internal floating member and individual tooth lubrication; applied for high speed (see Fig. 20., 21).

The teeth of these couplings are mostly provided with tooth tip centering. Otherwise, the same design principles are applied as for the non-disengaging couplings. In the engaged position they behave like an ordinary coupling.

For engagement and disengagement one set of teeth is brought into, or out, of mesh by axially moving the coupling sleeve. In the disengaged position the sleeve is supported, and centered, on one side by the teeth, which are still being in mesh, and on the other side by a centering ring. Both center rings are arranged on the same shaft, i.e. the one which is shut down after disengagement.

To ease engagement, the face of the engaging teeth are champfered. One of the two shaft systems must be equipped with a turning device to rotate one shaft system relative to the other, in case the teeth to be engaged are aligned exactly tooth on tooth.

For fully automatically controlled installations a special design allows the relative rotation of the shafts to be initiated by the coupling shifting mechanism itself. The shifting mechanism has to be dimensioned to overcome the brake-away torque. However, for engaging the coupling the shafts must be at stand still.

The shifting mechanism can be designed for manual or power operation. For power operation, usually hydraulically operated cylinders are used. Power fluid is tapped off the lube oil of the installation. It is of course advantageous to connect to a source with a high pressure, for example to the control oil circuit.

The shifting movement is transmitted from the hand lever, or cylinder, to the coupling sleeve by a lever and guide ring. For manual operation the hand lever is locked to a point on the casing which is axially spring-loaded. In case of power operation, the coupling sleeve is held in the engaged position in one direction by the power fluid, and in the other direction by the cylinder which is connected via springs to the casing. These arrangements avoid overloading the guide ring when the shafts expand axially.

For power operated couplings, an additional locking device is provided which locks the coupling sleeve in its position, in the case of an excessive pressure drop in the power system. In addition, this device prevents an engaging or disengaging action until the specified pressure, in the power system, is reached, see Figure 21.

Basically disengaging couplings can be designed to the same ratings as non disengaging coupling may be. Additional restrictions are the maximum allowable circumferential speed of the guide ring and the overhanging weight of the disengaged coupling sleeve.

3.3.2 Automatically Engaging and Disengaging Couplings

These couplings are installed in shaft systems to disconnect, and reconnect, one, or more machines, from a shaft system which remains in operation. The coupling, called synchronous clutch coupling, has been designed to engage automatically as soon as the disconnected machine or shaft system ("driving" system) overruns the speed of the operating machine or shaft system ("driven" system). Basically the clutch coupling is a disengageable coupling equipped with a mechanism (the synchronizing mechanism), which detects synchronism of both shafts and intitiates the engaging movement.

There are two types of clutch coupling:

a) Type HS, this type engages automatically but disengages upon command.

b) Type MS, this type engages and disengages automatically.

The synchronizing mechanism, the same basic design is applied for both types, is an assembly consisting of a number of pawls and a multiple notched ratchet wheel which act as a free wheel drive. The pawl cages together with the pawls are connected to the "driving" system and the ratchet wheel to the "driven" system.

Fig. 22 shows part of the assembly with the driving system at standstill and the driven system at its service speed, the clutch being disengaged. The spring presses the pawl lightly on to the ratchet wheel. Due to the speed difference between pawls and ratchet wheel and the oil which is fed through holes in the ratchet wheel, an oil film is generated which physically separates the two components.

To achieve an engagement the "driving" machine has to be started and accelerated. Fig. 23 shows the condition when the "driving" machine accelerates, the speed difference decreases. The spring has become inactive and the pawl is pressed onto the ratchet wheel by the centrifugal force acting on the pawl.

The "driving" machine accelerates further until it overruns the "driven" machine, which brings a pawl into engagement with a notch, shown in Fig. 24.

The accelerating torque of the driving machine is now transmitted by the pawl to the ratchet wheel and by this to the driven machine.

The ratchet wheel is connected to the driven machine via the synchronizer splines (helical teeth). Due to the transmitted torque an axial force is generated, in the synchronizer splines, which initiates an axial movement of the ratchet wheel. The movement is a screw motion because the ratchet wheel is guided by the synchronizer splines. The axial force is transmitted to the other member of the synchronizing-mechanism by a spring loaded buffer. This buffer limits the dynamic load on the pawl during synchronizing at high angular accelerations. The synchronizing-mechanism follows the screw motion of the ratchet wheel which brings the clutch teeth into engagement (1st Phase, see Fig. 25, 27)

For the continuation of this initial movement, which is fully automatic, two different systems have been designed, the HS and the MS-clutch.

The HS-clutch (see Fig. 25 and 26) is equipped with spur clutch teeth. This allows the integration of the synchronizing-mechanism in the floating coupling sleeve, since the clutch teeth can operate as coupling teeth. An external force is required to fully engage the coupling sleeve. This external force is excerted automatically by the servo mechanism after the coupling has travelled a certain axial distance in the 1st phase. Once fully engaged, the HS-clutch can transmit torque continously in both directions, which is of great advantage for certain applications. To disengage the clutch, the torque has to be reduced, as in the case of an ordinary disengageable coupling, and a command is given to the servo mechanism which reacts immediately and disengages the clutch.

The MS-Clutch (see Fig. 27 and 28) is equipped with helical clutch teeth. The axial movement is performed by the synchronizing-mechanism only, which carries the helical clutch teeth and one set of the coupling teeth. The first phase of engagement, which is an action of the synchronizing mechanism, is concluded when the clutch teeth come into mesh, see Fig. 27. The torque now generates an axial force in these helical clutch teeth, which consequently fully engage. The clutch remains engaged as long as the torque is positive, i.e. a torque is transmitted from the driving to the driven machine. In this condition the synchronizing-mechanism is rigidly connected to the driven shaft system. The axial force generated in the helical clutch teeth is closed in a loop, no forces, except the coupling thrust, are acting on the shaft system. If the torque reverses, the MS-clutch disengages automatically.

Both the HS- and MS-clutch behave in the engaged position like an ordinary toothed coupling, and the same theoretical considerations are valid. Both types of clutches are designed such that the contact faces of the pawl and the ratchet wheel are physically out of contact, which assures that the synchronizing-mechanism takes no part in transmitting the torque, with the clutch coupling in fully engaged position.

A device can be added to both types which allows the driving machine to be rotated while isolated from the driven shaft system, this is known as the "pawl free" position.

The clutch couplings have a wide range of application, some of which are mentioned below:

- Peaking power plants (i.e. turbine/alternator units with power and synchronuous condensing operating mode)
- Air storage power plants
- Combined ship propulsion plants such as CODOG, CODAG, CODAD, COGOG, COGAG.
- Connecting additional power from expander turbines in petro chemical plants
- Combined cycle machinery
- Starting device for gas turbines
- Turning gear drives

A further application is the combination of a HS-clutch coupling with a hydraulic coupling for a gas turbine-alternator drive. This combination may be installed between a one-shaft gas turbine and an alternator which is equipped with a frequency converter. With the alternator operating as synchronuous condenser the clutch coupling is disengaged, the hydraulic coupling is empty, and the gas turbine is at standstill. If power is required the gas turbine is accelerated by filling the hydraulic coupling. After the turbine has reached the self sustaining speed, the hydraulic coupling is emptied. The turbine accelerates further until it overruns the alternator. The clutch coupling engages automatically. Power can now be transmitted from the turbine to the alternator. With this arrangement an expensive seperate starting device for the gas turbine is unnecessary.

Should a longer starting time be acceptable then even the hydraulic coupling can be eliminated. The gas turbine is started by the turning gear and kept at constant speed. At the same time the alternator is decelerated by using the frequency converter. As soon as the speed of the alternator falls below the gas turbine speed (given by turning gear) the HS-clutch engages fully. The gas turbine can now be accelerated by the alternator working as a motor. When synchronuous speed has been reached the alternator is synchronized and power can be transmitted. For this application the HS-type clutch is ideal, since having spur clutch teeth the clutch can transmit the torque equally well from gas turbine to generator and vice versa for the start up.

For certain propulsion systems having two different operating modes, such as marine reversing gears for ahead and astern rotation of a fixed pitch propeller driven by a uni-directional prime mover, or combined propulsion systems (i.e. CODAG) in which the diesel engine (prime mover) drives through a 2-speed gearbox, a fully automatic and smooth changeover from one to the other operating mode is required.

In such propulsion systems the prime mover is connected to the propeller via two shaft-trains, each having a clutch. According to the selected propulsion mode, the torque is being transmitted by one shaft train with its clutch (input and output shaft of the clutch have the same speed and direction of rotation), while the other shaft train is totally disconnected by its clutch and therefore cannot transmit any torque (the output shaft of the clutch may rotate faster, slower or even in opposite direction in respect to the input shaft), and vice versa.

An ordinary MS or HS type clutch does not fulfill the requirement of complete disconnection, since should the output shaft rotate slower or in opposite direction the clutch would immediately engage.

If however a MS-type clutch is combined with a inversed HS-type clutch a double clutch is formed (called DS-type) which can cope with the aformentioned requirements, according to the selected propulsion mode, that is:

- torque transmission with equal speed of input- and output shaft of the clutch

or:
- no torque transmission, totally disconnected clutch, allowing any speed and direction of rotation of output shaft relative to input shaft of the clutch

An integrated hydromechanical control system allows, at any time, the fully automatic sequence of the clutch functions, depending on the one hand on the selected operating mode, and on the other hand on the momentarily prevailing speed conditions of the input and output shafts. Clutch movements take place only at no torque conditions. The clutch and its control system are self-regulating, that means that neither the speeds of the input and output shafts nor the torque have to be changed.

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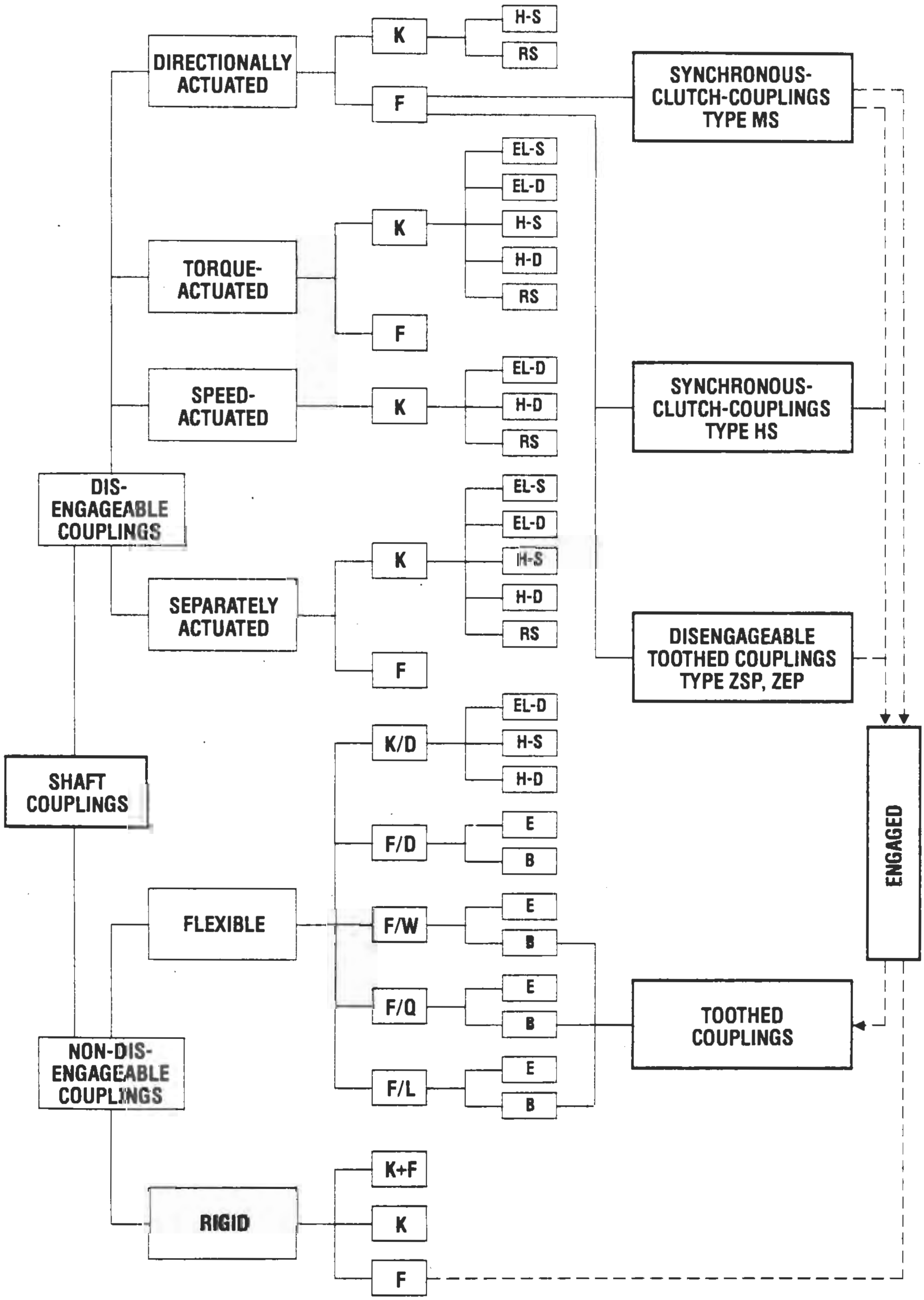


FIG. 1

B = FLEXIBLE
D = TORSIONALLY FLEXIBLE
L = AXIALLY FLEXIBLE
Q = RADIALLY FLEXIBLE
W = ANGULARLY FLEXIBLE
E = RESTORING

K = FORCE CLOSED
F = FORM CLOSED
RS = FRICTION CLOSED
EL-D = ELECTRO-DYNAMIC
EL-S = ELECTRO-STATIC
H-D = HYDRO-DYNAMIC
H-S = HYDRO-STATIC

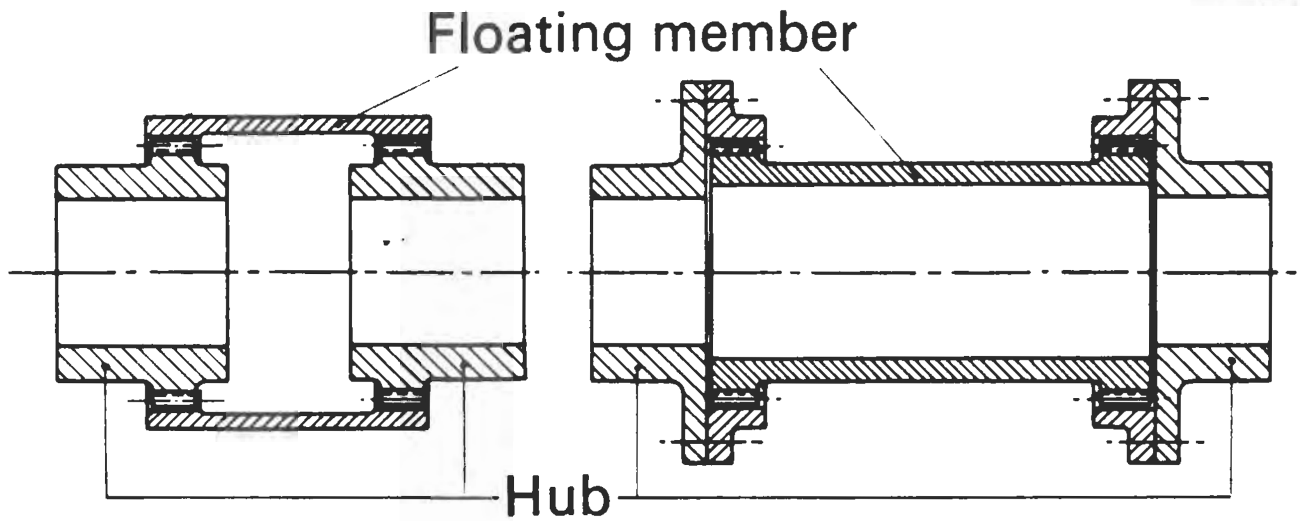


Fig. 2

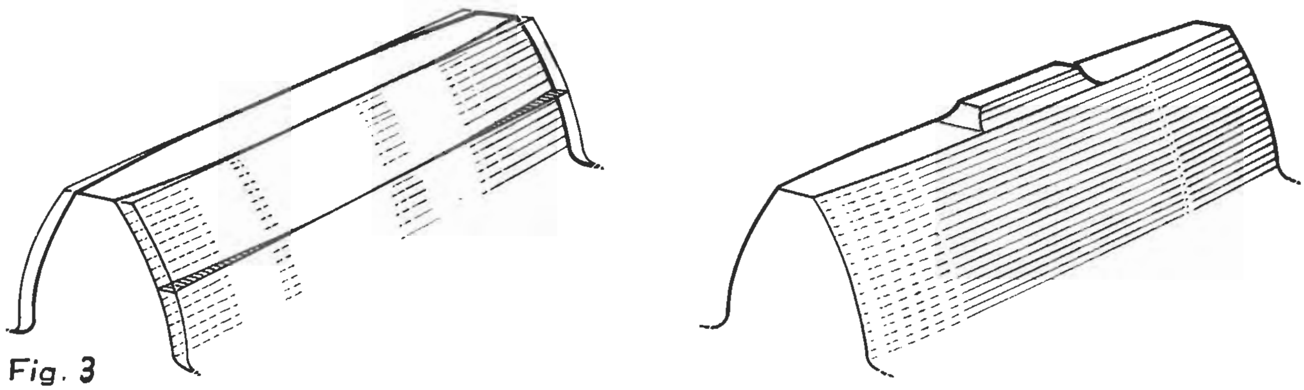


Fig. 3

Tooth without tip centering

Tooth with tip centering

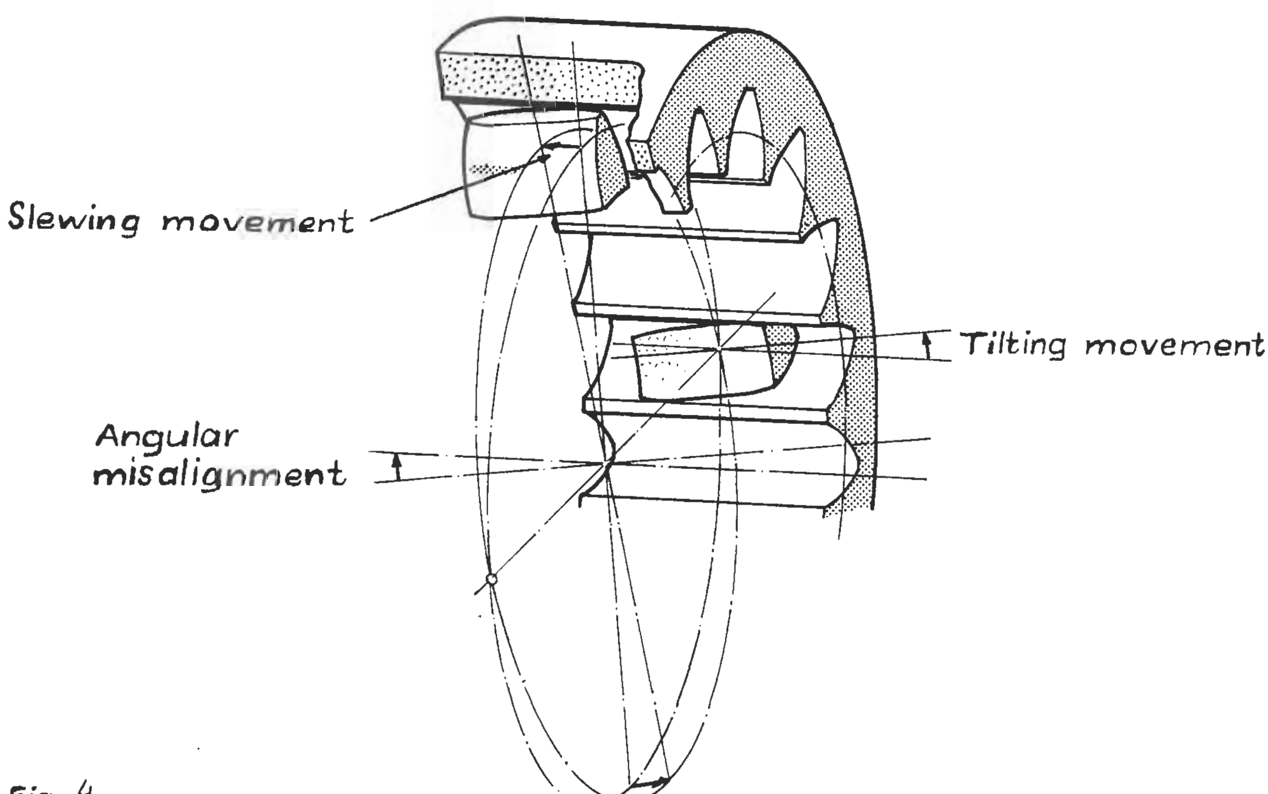


Fig. 4

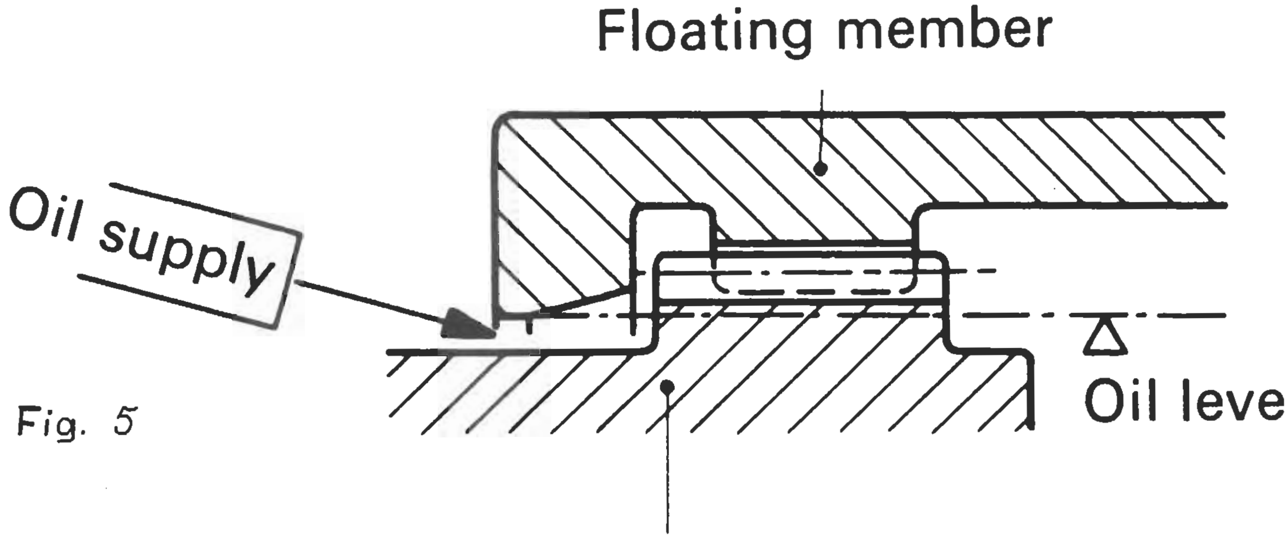


Fig. 5

Hub

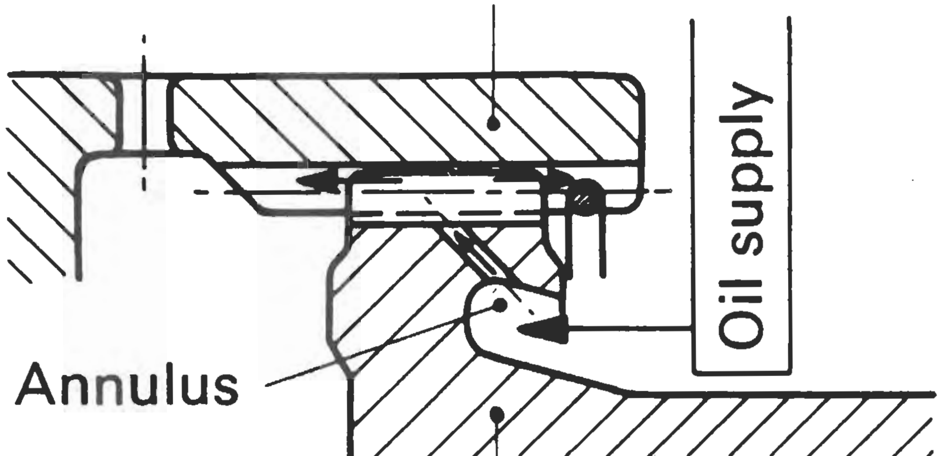


Fig. 6

Floating member

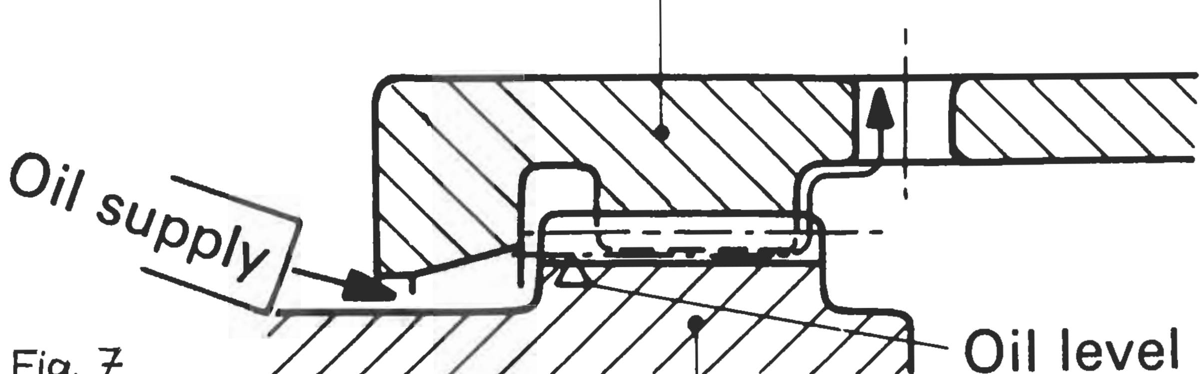


Fig. 7

Hub

Forces acting on the floating member

F_T = Circumferential force due to torque

F_C = Centrifugal force

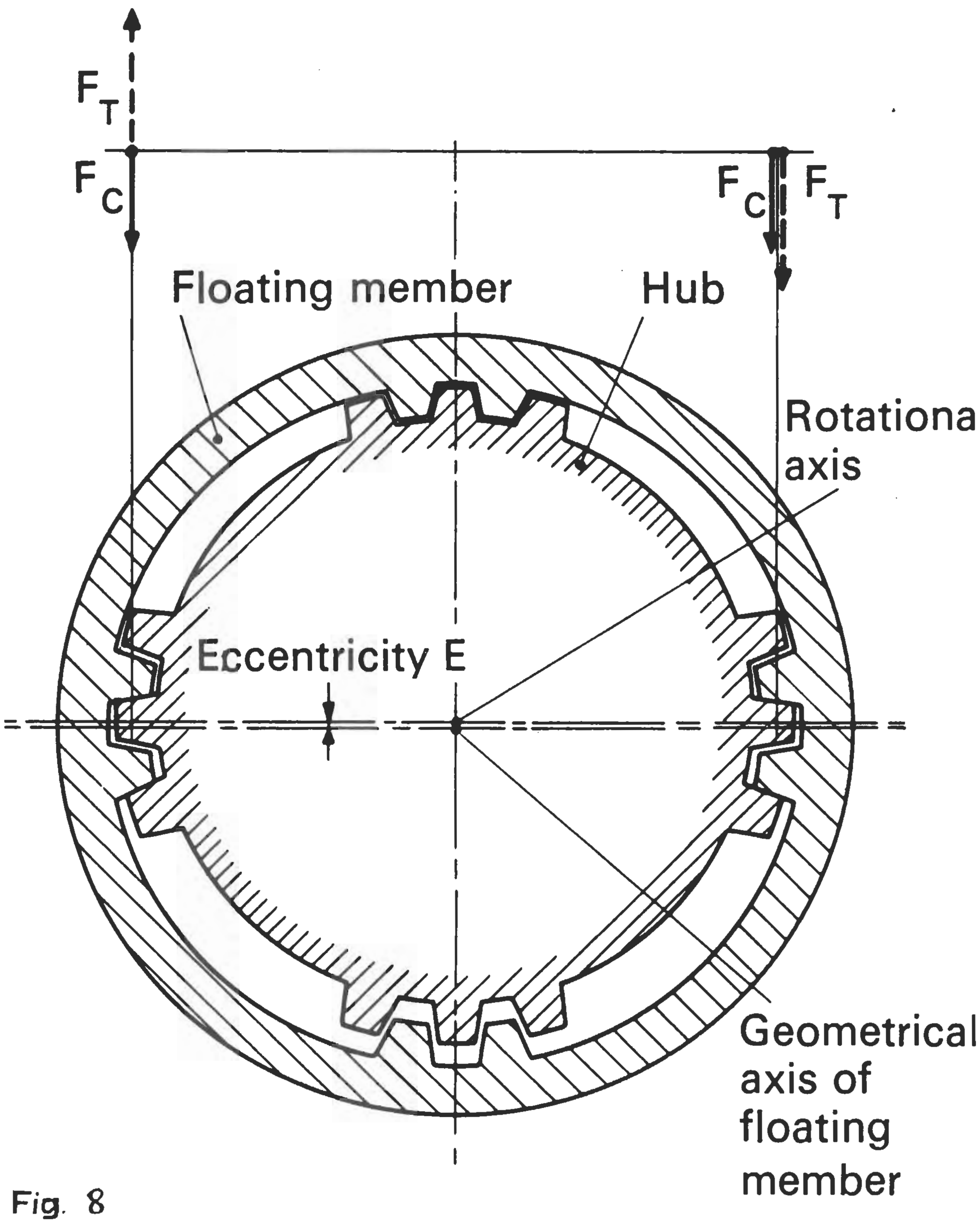


Fig. 8

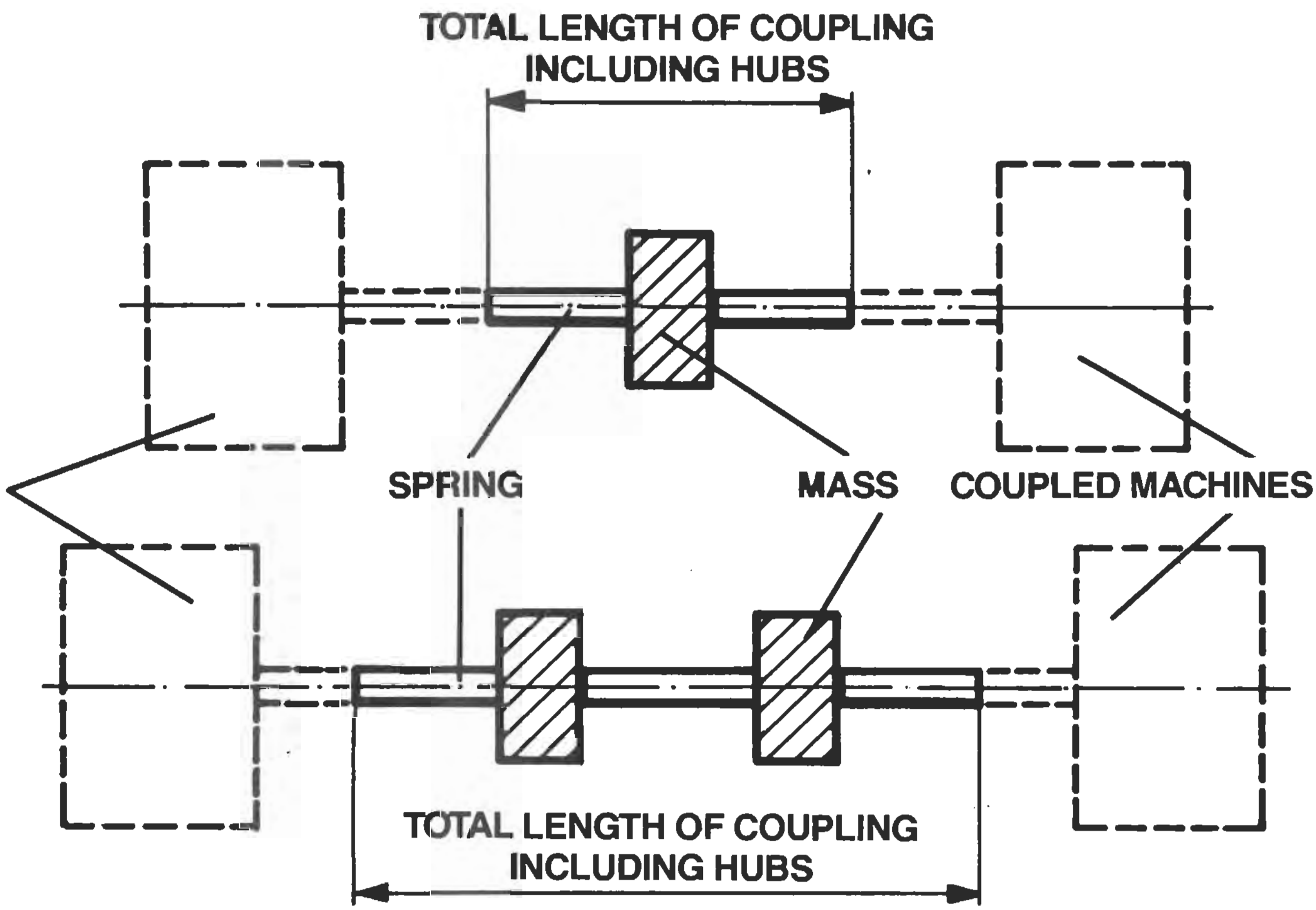

TOTAL LENGTH OF COUPLING
INCLUDING HUBS
SPRING
MASS
COUPLED MACHINES
TOTAL LENGTH OF COUPLING
INCLUDING HUBS

FIG. 9

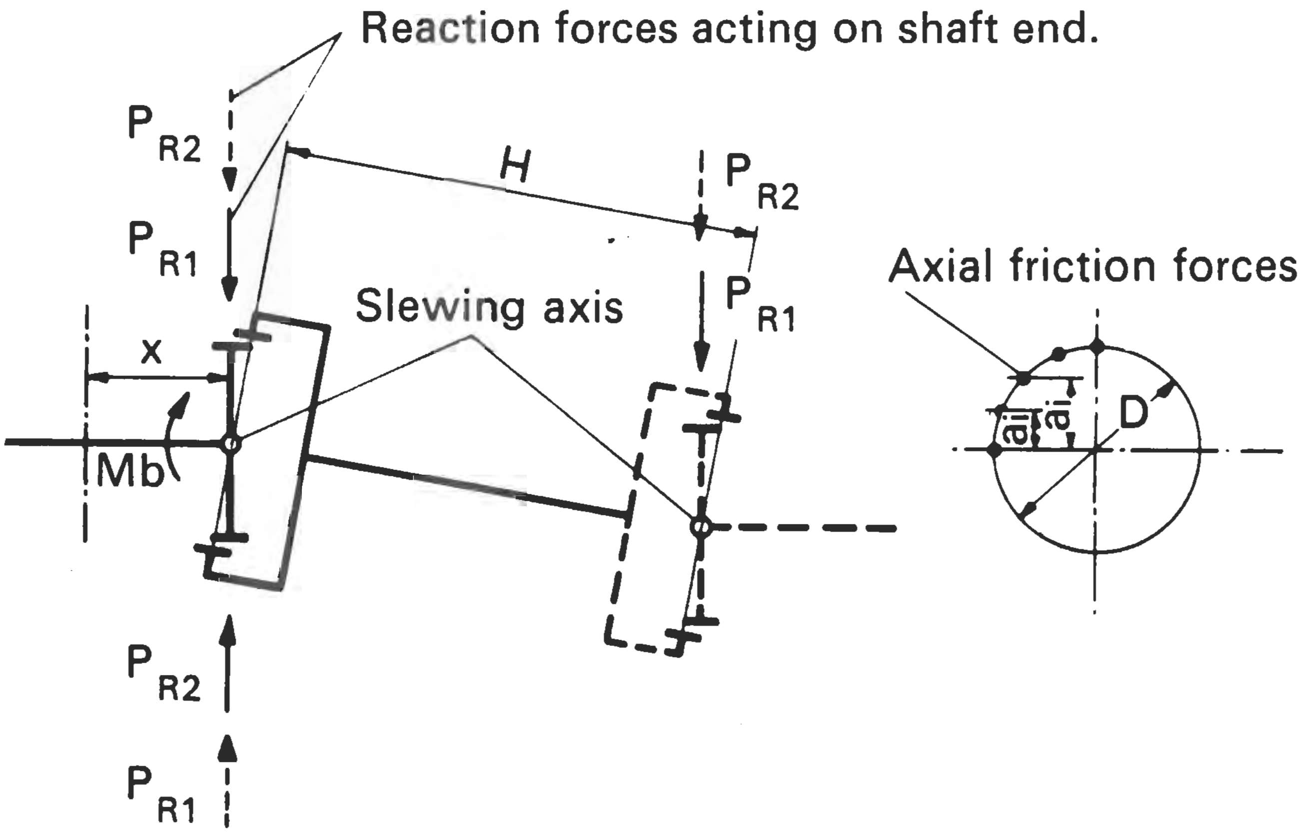

Reaction forces acting on shaft end.
P R2
P R1
H
P R2
P R1
Axial friction forces
Slewing axis
x
Mb
ai
ai
D
P R2
P R1

Fig. 10

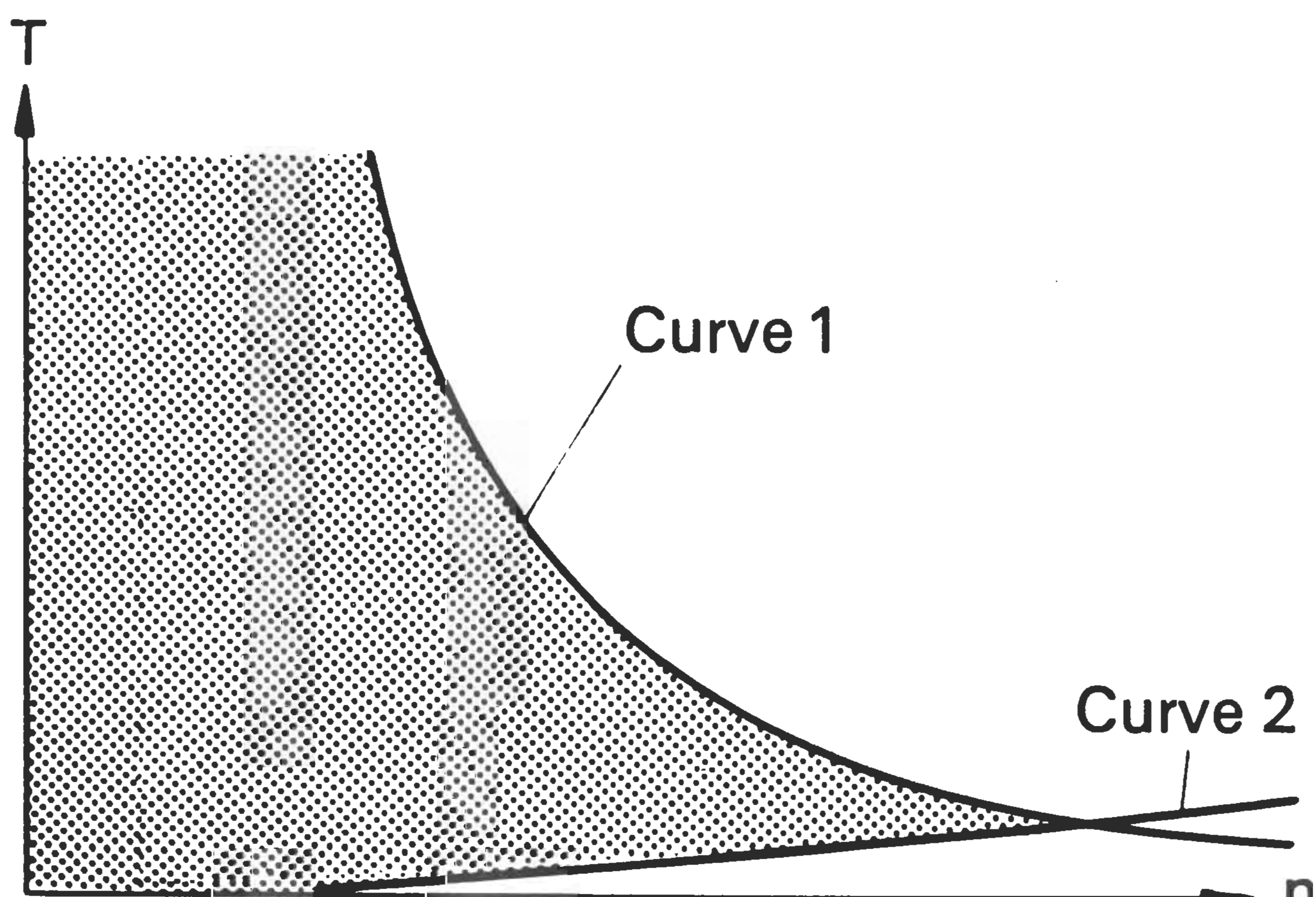
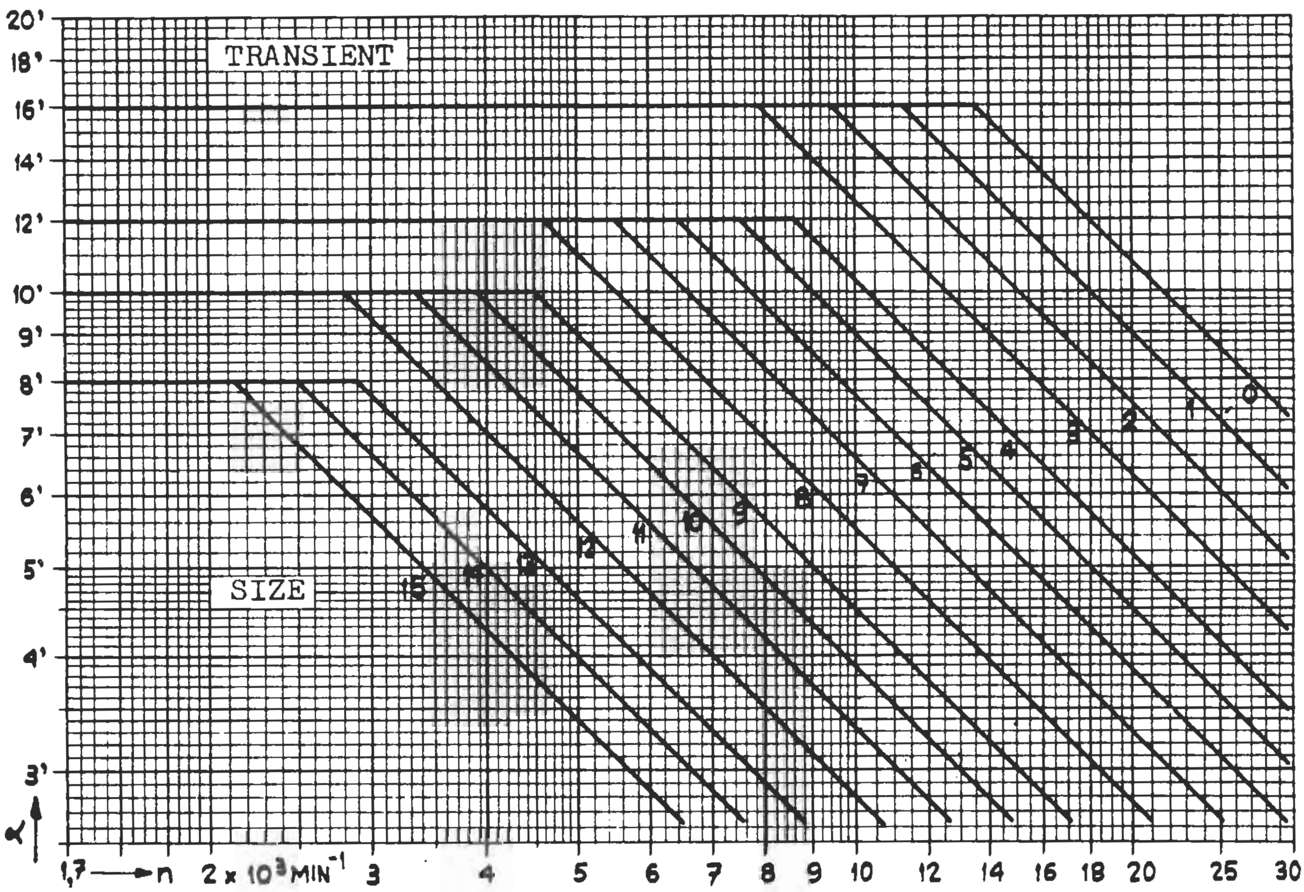

T
Curve 1
Curve 2
n

Fig. 11



α = Angular misalignment, minutes

n = Service speed, min^{-1}

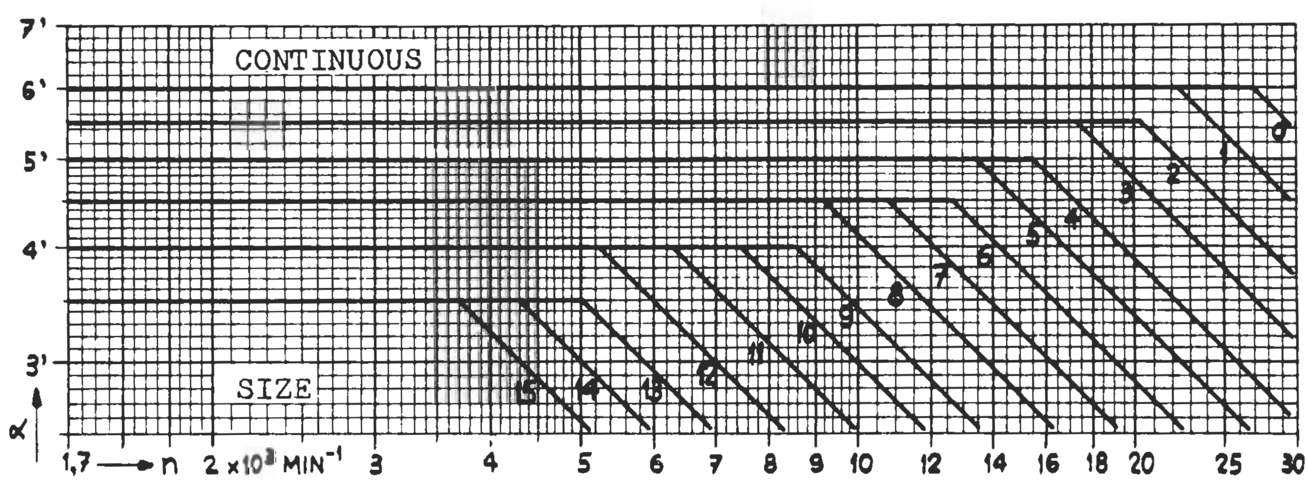


Fig. 12

Fig. 13

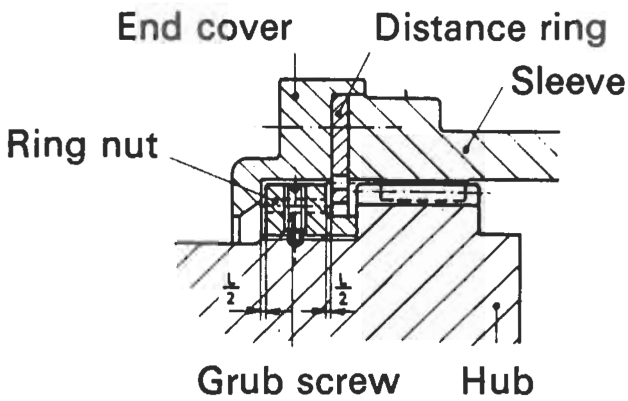


Fig. 14

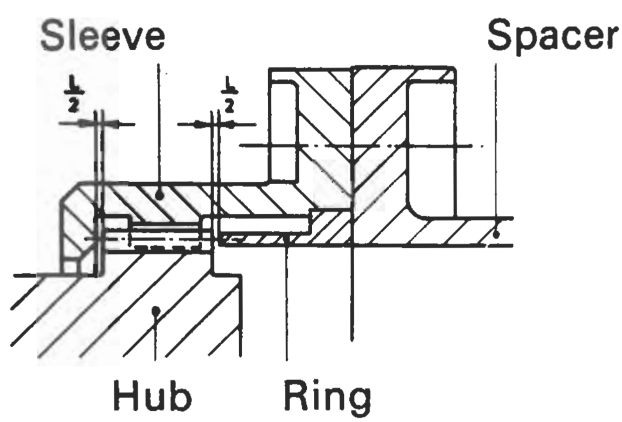


Fig. 15

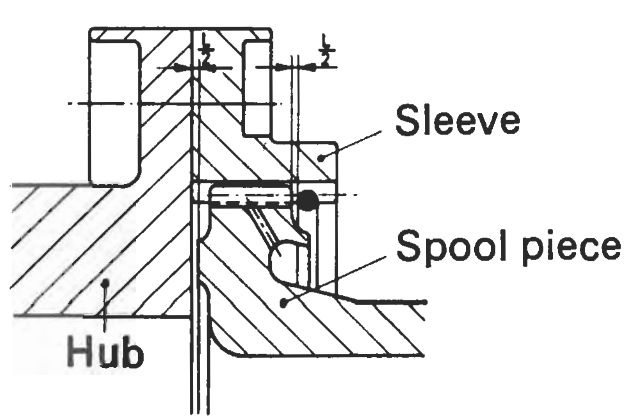


Fig. 16

Size	$L_{tot} = 2 \cdot L$ Minimum Value
0	1.2
1	1.2
2	1.2
3	1.6
4	1.6
5	1.6
6	2
7	2
8	2.4
9	2.4
10	2.8
11	3.2
12	3.6

Fig. 17

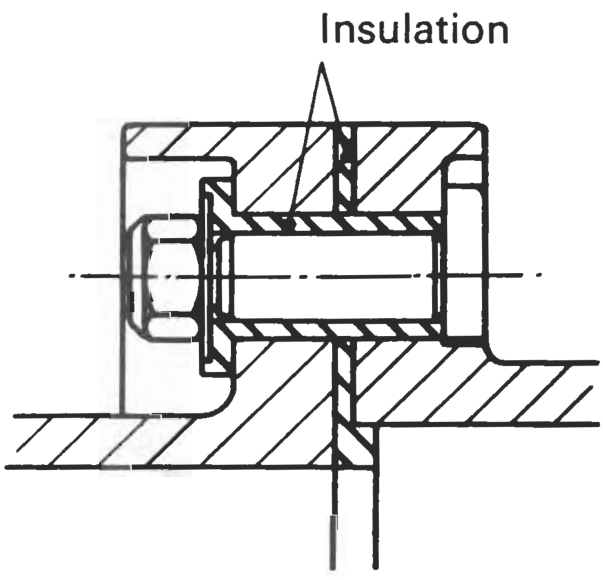


Fig. 18

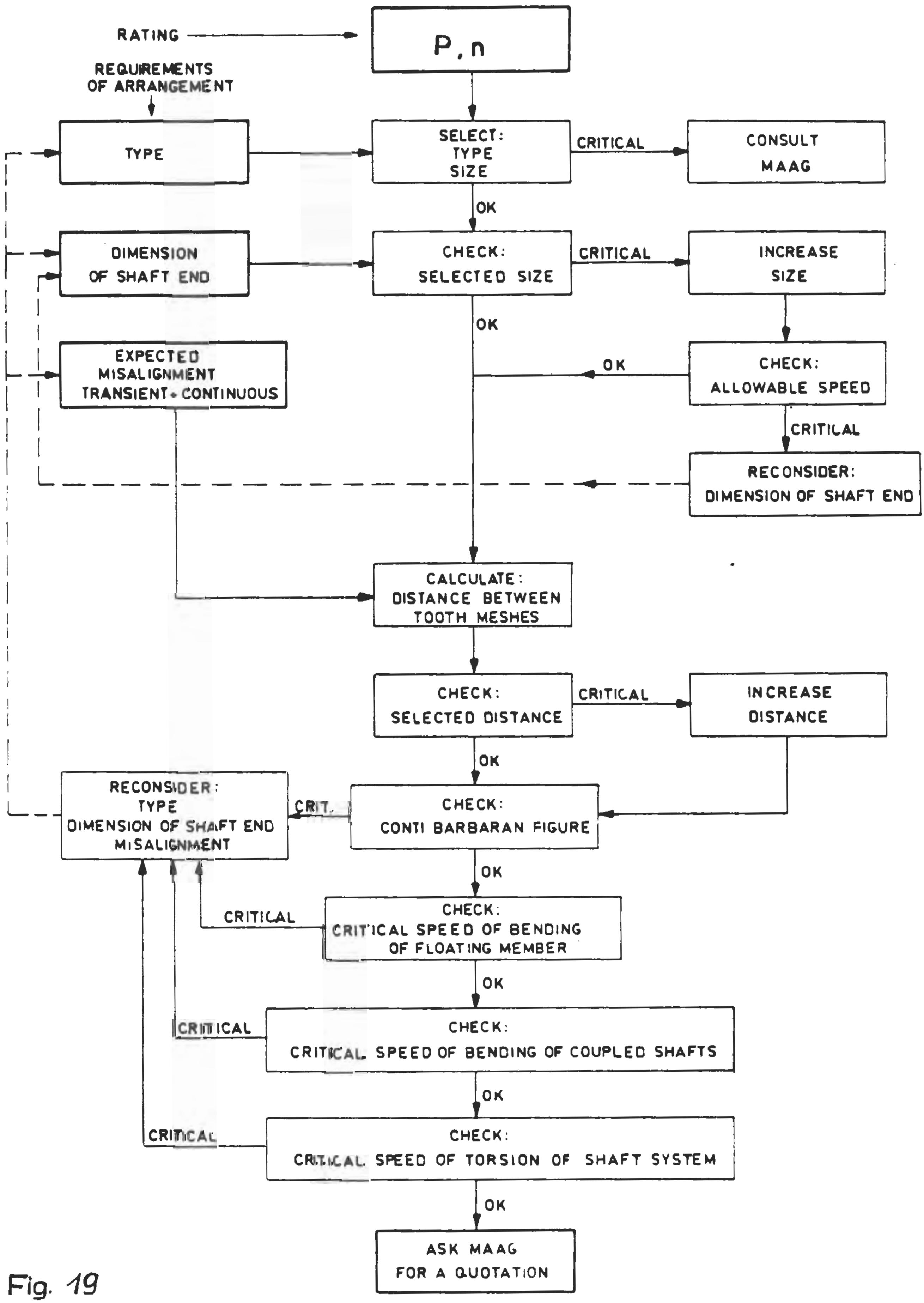

RATING
P, n
REQUIREMENTS OF ARRANGEMENT
TYPE
SELECT: TYPE SIZE
CRITICAL
CONSULT MAAG
OK
DIMENSION OF SHAFT END
CHECK: SELECTED SIZE
CRITICAL
INCREASE SIZE
OK
EXPECTED MISALIGNMENT TRANSIENT + CONTINUOUS
OK
CHECK: ALLOWABLE SPEED
CRITICAL
RECONSIDER: DIMENSION OF SHAFT END
CALCULATE: DISTANCE BETWEEN TOOTH MESHES
CHECK: SELECTED DISTANCE
CRITICAL
INCREASE DISTANCE
OK
RECONSIDER: TYPE DIMENSION OF SHAFT END MISALIGNMENT
CRIT.
CHECK: CONTI BARBARAN FIGURE
OK
CRITICAL
CHECK: CRITICAL SPEED OF BENDING OF FLOATING MEMBER
OK
CRITICAL
CHECK: CRITICAL SPEED OF BENDING OF COUPLED SHAFTS
OK
CRITICAL
CHECK: CRITICAL SPEED OF TORSION OF SHAFT SYSTEM
OK
ASK MAAG FOR A QUOTATION

Fig. 19

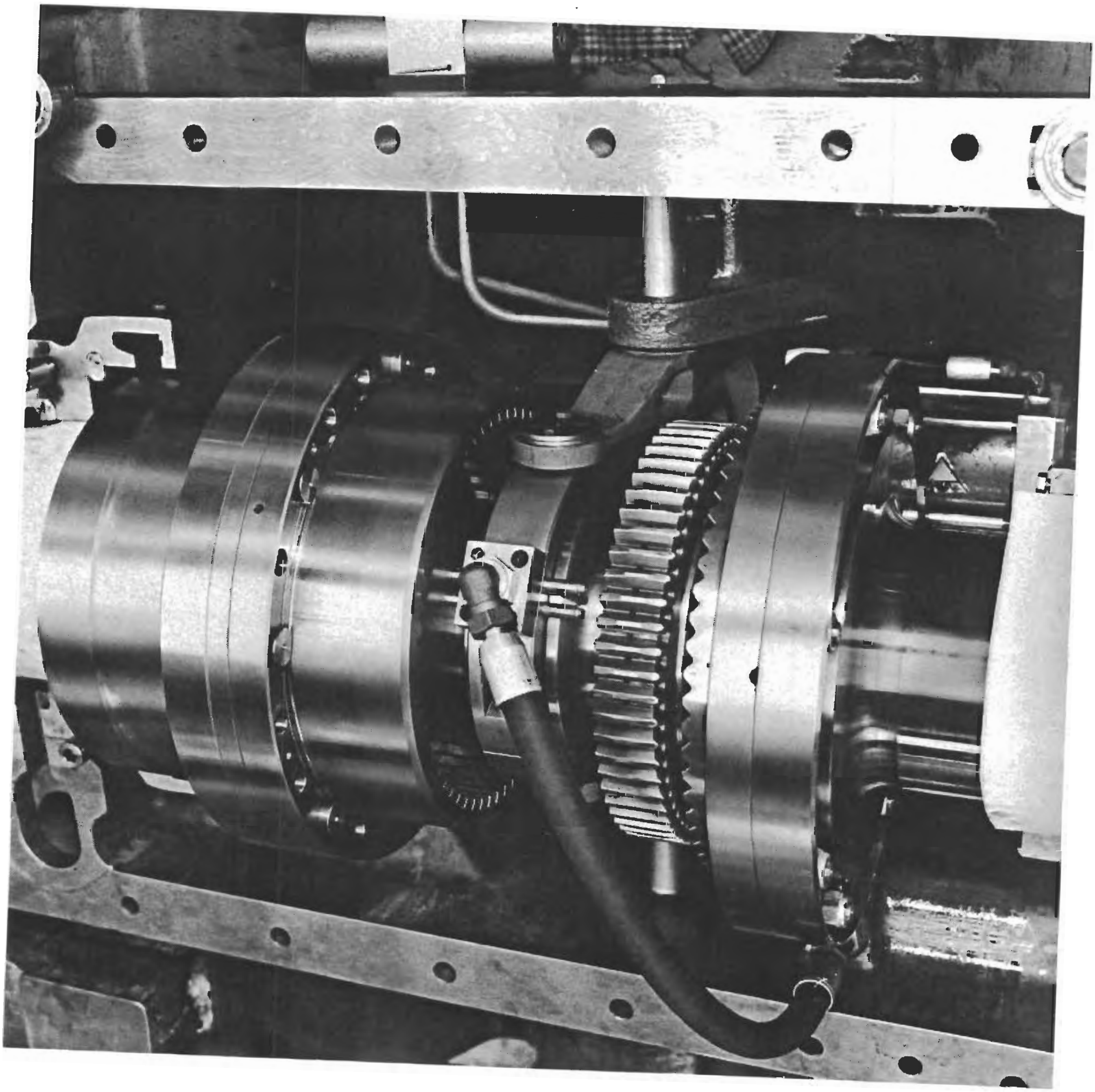
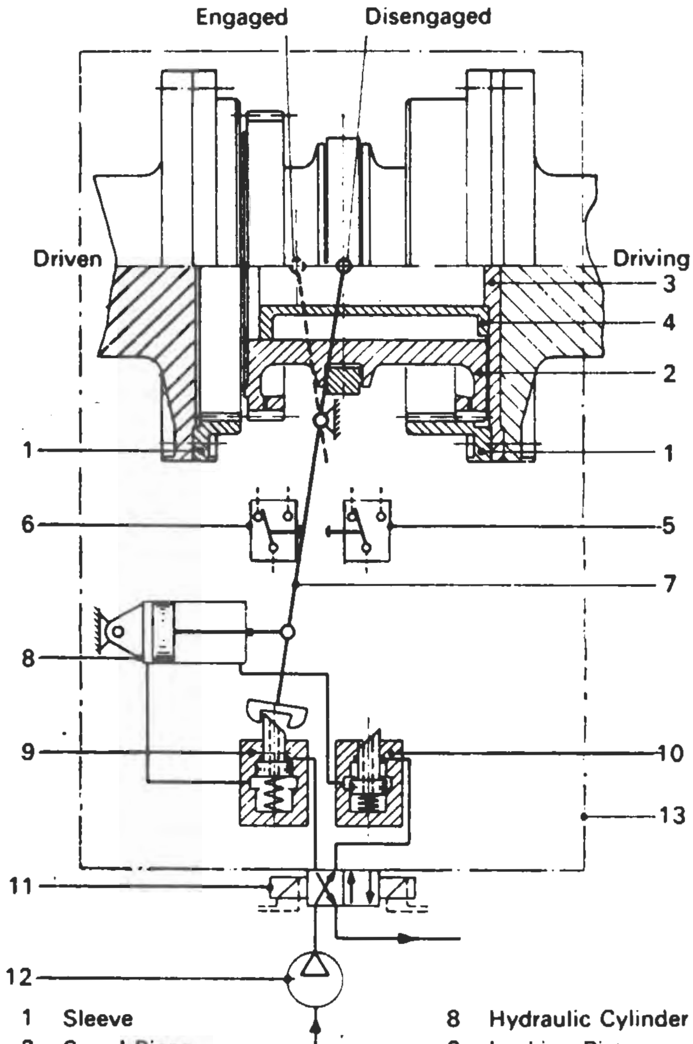


Fig. 20



1 Sleeve
2 Spool Piece
3 Intermediate Flange
4 Support Sleeve
5 Limit Switch "Engaged"
6 Limit Switch "Disengage"
7 Lever
8 Hydraulic Cylinder
9 Locking Piston
10 Locking Piston
11 Solenoid Valve
12 Oil Pump
13 Casing

Fig. 21

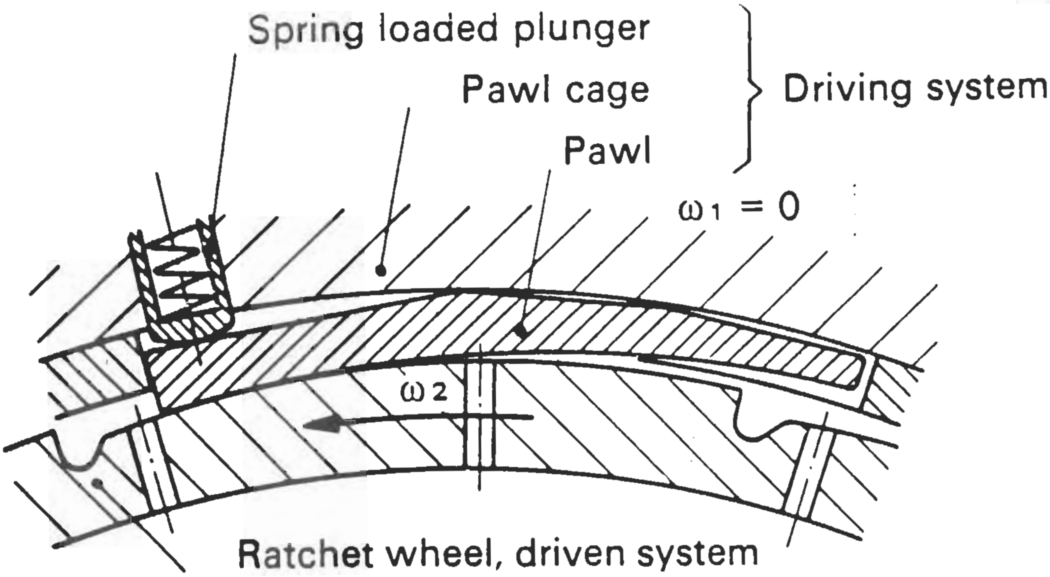


Fig. 22

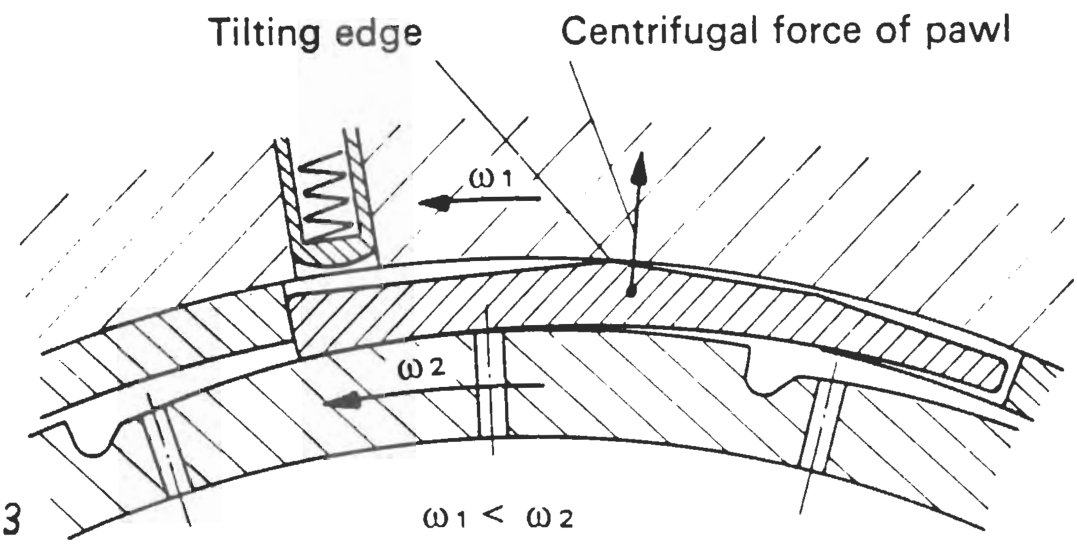


Fig. 23

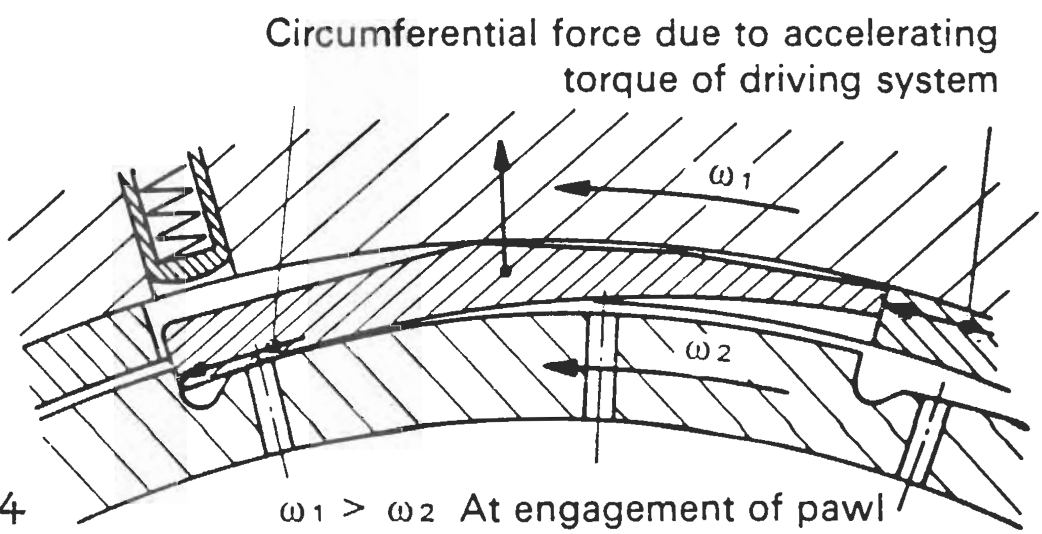


Fig. 24

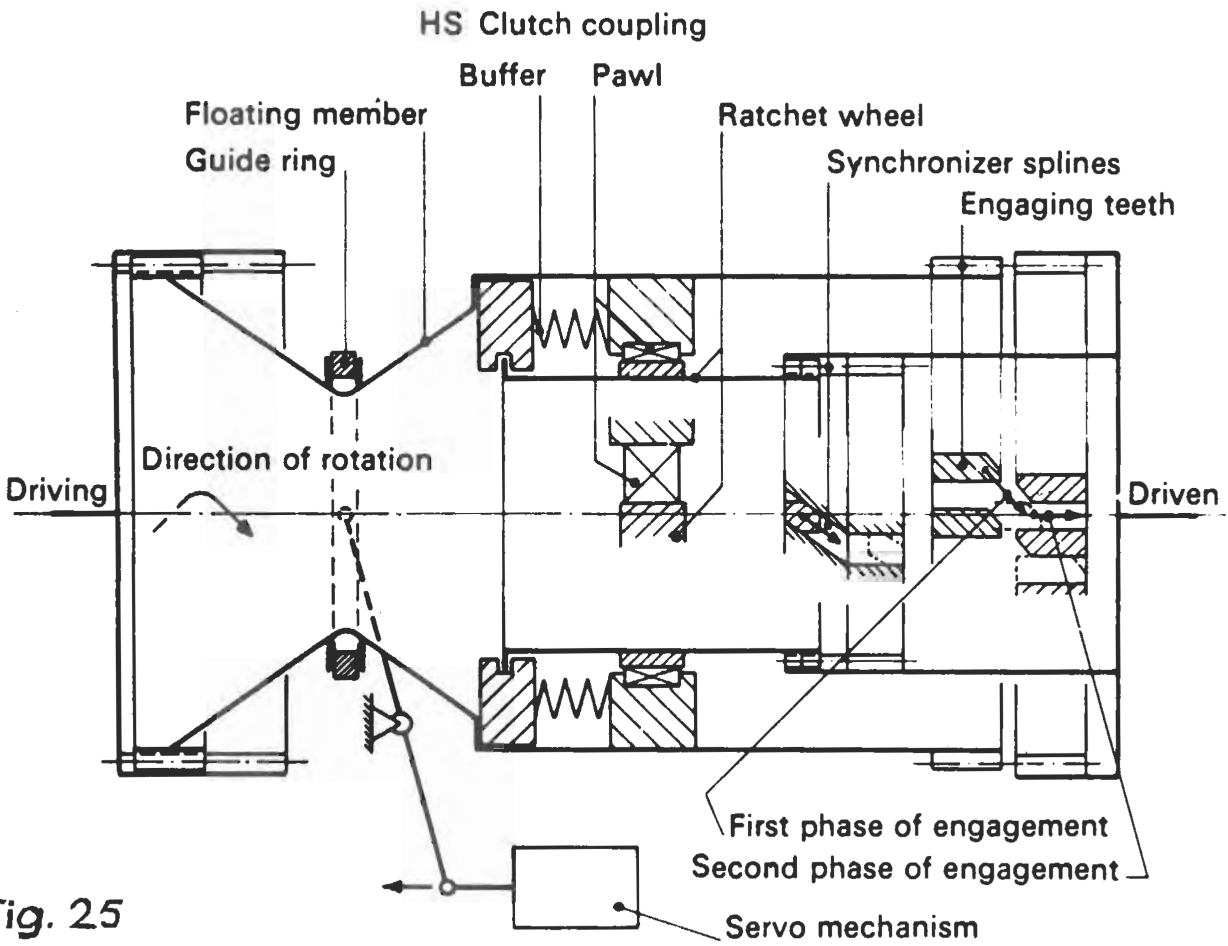

HS Clutch coupling
Buffer
Pawl
Floating member
Ratchet wheel
Guide ring
Synchronizer splines
Engaging teeth
Direction of rotation
Driving
Driven
First phase of engagement
Second phase of engagement
Servo mechanism

Fig. 25

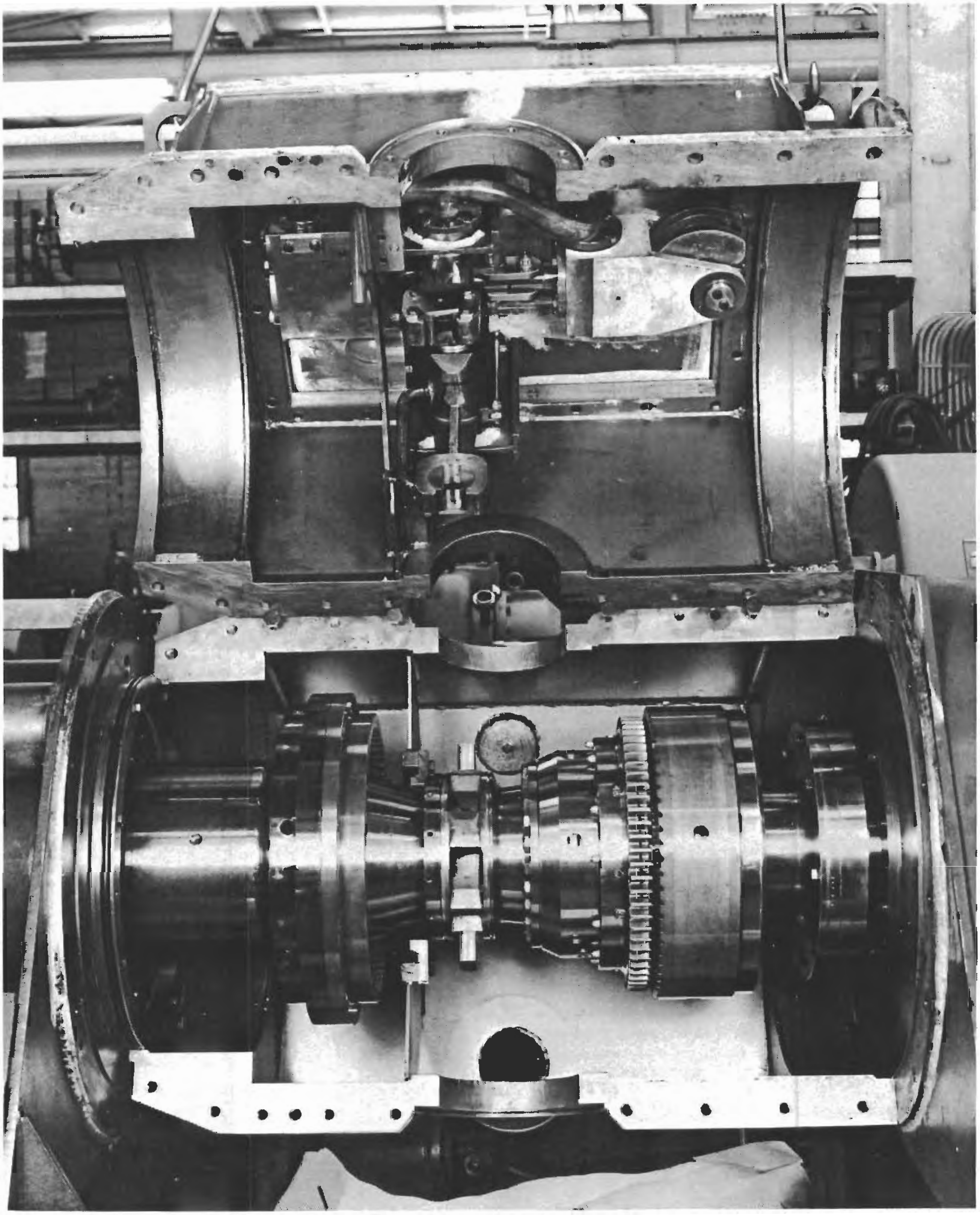


Fig. 26

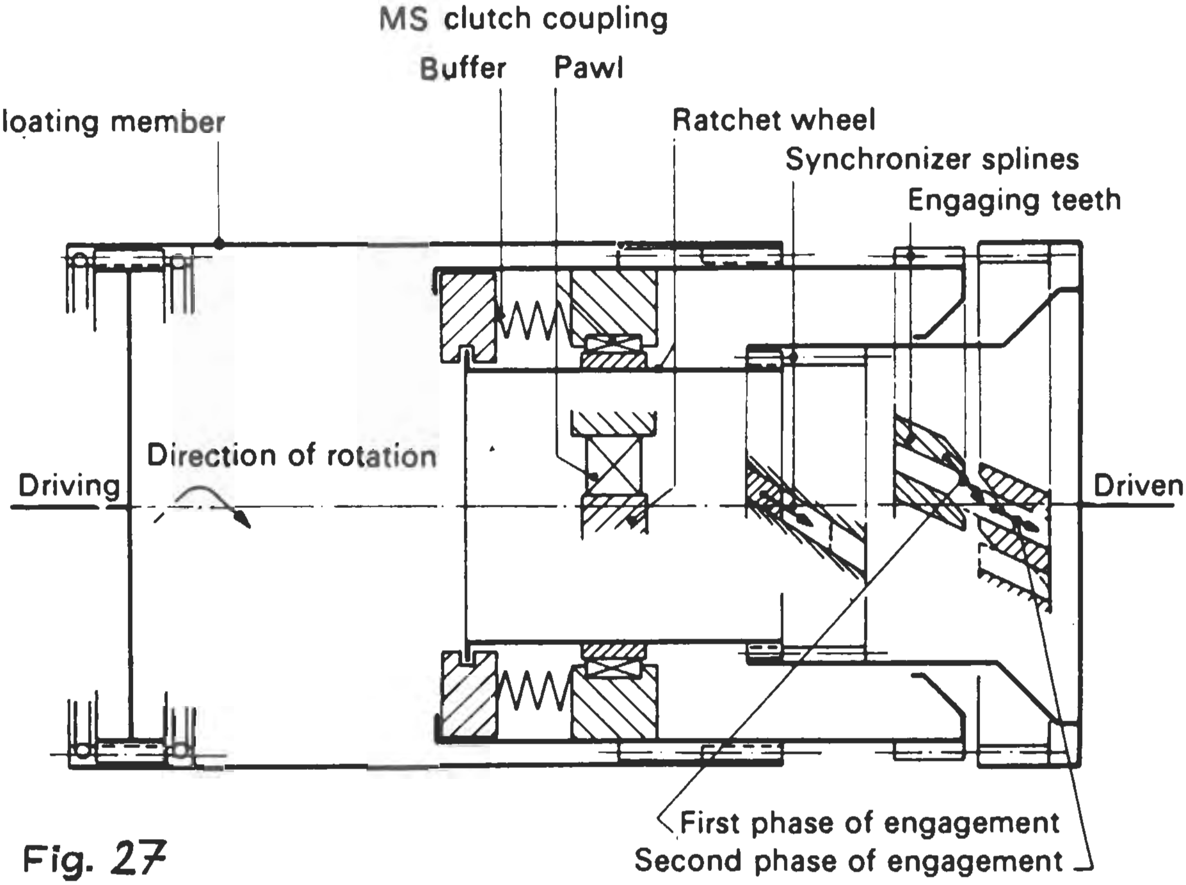

MS clutch coupling
Buffer
Pawl
Floating member
Ratchet wheel
Synchronizer splines
Engaging teeth
Direction of rotation
Driving
Driven
First phase of engagement
Second phase of engagement

Fig. 27

Fig. 28

oil flow required, as only a part of the entire oil quantity reaches the tooth flanks which are to be lubricated.

The arrangement according to Fig. 7 can only be applied together with tip centered teeth. With regard to centrifuging, this design is practically identical to Fig. 5. The drain is similar to the design according to Fig 6. A substantial disadvantage is that the oil level is governed by the tooth clearance. A slight increase of the clearance, say due to minor wear which would have no adverse effect on the performance of the coupling, would no longer guarantee sufficient lubrication. The initial negligible wear would then very rapidly reach unacceptable values. This is the reason why MAAG does not apply this lubrication method.

To summarize, there is no design which is absolutely free from accumulating impurities. The toothed coupling acts like a centrifuge. To achieve optimum performance of the coupling, not only the design must be impeccable, but the lube-oil system and the maintenance must also cope with the requirement of minimizing the presence of impurities in the oil.

2.4 Calculations

2.4.1 Calculation of the Allowable Torque

The tooth flanks transmit the torque. The allowable torque may be calculated using the following formula:

$$T = \frac{d \cdot z \cdot b \cdot h}{2} \cdot P_a$$

This formula is based on an even load distribution between all the teeth.

When the sleeve is misaligned relative to the hub, the load distribution becomes uneven, a factor on which the amount of barreling has a considerable influence. The greater the misalignement and barreling, the greater the difference between individual tooth load (6). However, the load distribution is also dependent on the pitch errors in the teeth. It is a very complex problem to determine the actual load distribution (2). The

The minimum torque required to centre the sleeve can be calculated by using the following formula (7):

$$T_{min} = m_H \cdot E \cdot \omega^2 \cdot \frac{d}{4}$$

The increase in clearance due to difference in expansion between the floating member and the hub, at high speed, must be considered when determining the eccentricity E.

However, experience has shown that the torque must be greater to positively centre the floating member during running. From a statistical investigation into the performance of a large number of couplings the following formula, known as Conti-Barbaran-formula, has been derived (5):

$$\varepsilon = \frac{P \cdot 1.36 \cdot 10^9}{G_H \cdot d^2 \cdot n^3}$$

For $\varepsilon > 10$ a stable and troublefree performance of the coupling can be expected. If $\varepsilon < 10$ but > 5 difficulties may arise if the running conditions are unfavourable. If $\varepsilon < 5$ problems are most likely to occur.

It must be noted that these values have been found empirically. The formula does not consider the actual eccentricities nor the production methods. Nevertheless, the formula is very valuable for quickly estimating the behaviour to be expected.

b) Lateral natural frequency

With couplings having a long floating member, the lateral natural frequency of the spacer may come close to the operating frequency. For the calcultaion of the first order lateral critical speed, it is assumed that the floating member has a uniform section over the whole length H and that the supports are links free of friction. The critical speed can then be calculated according to the following formula:

allowable surface pressure is in addition dependent on surface finish, surface hardness, and the material combination. All these factors are allowed for in practice by assuming an allowable surface pressure, which has proved suitable for a given make of coupling according to operational experience. A generally valid figure for the allowable surface pressure cannot be defined, since the coupling manufacturers use different production methods. However, it can be said that the harder the materials and the better the surface finish and the accuracy of the teeth, the higher is the allowable surface pressure (4).

2.4.2 Dynamic Behaviour

When considering the dynamic behaviour of a coupling, two categories can be defined:

1) The "internal" dynamic behaviour
2) The "external" dynamic behaviour

However, the two categories are somewhat interdependent, i.e. the "internal" behaviour may influence the "external" and vice versa.

1) "Internal" dynamic behaviour

a) Minimum torque

At standstill the sleeve is eccentric to the rotational axis of the shafts due to the clearance in the teeth. When the coupling starts to rotate, the sleeve will revolve eccentrically due to the centrifugal force, as long as no torque is transmitted. The magnitude of the generated out-of-balance depends on:

a) Speed
b) Weight of the floating member
c) Eccentricity, i.e. backlash or clearance of tip centering.

Therefore, a free rotational force is generated which will be transmitted by the hubs to the shafts. To bring back the geometrical axis of the floating member into the rotational axis of the shafts a torque has to be transmitted, see Fig. 8.

$$n_{crit} = \frac{C \cdot 10^3 \cdot i}{H^2}$$

C = 4.82 for components made of steel

C = 4.75 for components made of Al-alloy

$$I = \frac{da}{4} \quad \text{for solid shafts}$$

$$i = \sqrt{\frac{da^2 + di^2}{16}} \quad \text{for hollow shafts}$$

The calculated critical speed should be at least 20% above the maximum service speed.

2) "External" dynamic behaviour

Under "external" dynamic behaviour, the influence of the coupling on the vibrational characteristics of the shaft will be discussed.

a) Out-of-balance

Even a perfectly centered coupling shows some imbalance. The allowable residual imbalance is calculated from the allowable eccentricity between mass axis and rotational axis and the weight of the component to be balanced:

$$U = e \times W$$

It is practically impossible to carry out the dynamic balancing of an assembled coupling. To positively centre the floating member on the hubs, the coupling would have to be balanced in a torqued condition. The loading mechanism would be far too heavy relative to the coupling weight, and the balancing accuracy would be impaired. In the case of tip centered couplings, certain specifications call for a tip clearance which does not disturb the balancing. This is not achievable for high speed couplings. The allowable excentricities are in the range of fractions of 1/1000 mm whereas the tip clearance is within a few 1/100 mm.

b) Overhang moment

When calculating the lateral critical speeds of the coupled shafts, the mass and the centre of gravity of the "half coupling" acting on the shafts have to be considered.

c) Coupling factor

The calculation of the lateral critical speed of coupled shaft systems is difficult since the actual coupling factor is unknown. Therefore, it is assumed, for the calculations, that the tooth meshes of the coupling are links free of fraction. A misaligned coupling however induces always forces and moments on the coupled shafts. An attempt has been made to determine the coupling factor empirically (8).

d) Torsional vibration

When computing the torsional critical speeds, the coupling has to be regarded as a member of the entire shaft system. The coupling can be assumed to be a one-mass two-spring system. However, to calculate with reasonable accuracy the critical speeds of higher orders it is necessary to replace the coupling by a two-mass three-spring system, see Fig. 9.

The coupling is insensitive to torsional vibration as long as, during continuous operation, the lower value of the vibratory torque does not impair the centering of the coupling, and the stresses caused by the vibratory torque are acceptable for the components.

2.4.3 Induced Forces

Mechanics tell us that when two bodies are pressed against each other and have to be moved relative to each other, a frictional force is generated. The same is valid for a torsionally loaded coupling in which the hubs are to be moved axially relative to the floating member. The frictional force, the so called coupling thrust, is calculated according to:

$$F_S = \frac{2 \cdot T \cdot \mu}{d}$$

When two machines, each one provided with a thrust bearing, are coupled by a toothed coupling then the thrust bearings must be capable of carrying this thrust because the shafts will always expand axially when the machines are operating. If only one machine is equipped with a thrust bearing, a coupling with limited axial float may be fitted. The entire shaft system is then axially located, by the one thrust bearing, and the coupling with the adjacent shaft will follow the expansion and thus no frictional force will act on the thrust bearing.

In chapter 2.3.2 we have seen that, when a coupling is misaligned, a relative axial movement in the teeth occurs (slewing movement). In a torsionally loaded misaligned coupling which is rotating, frictional forces will be continuously induced.

The coupling slews around an axis which is normal to the slewing plane. At each tooth, a frictional moment is generated by the frictional force acting on the tooth and the distance a_i of the tooth to the slewing axis see, Fig. 10. The frictional moment, with which the teeth oppose misalignment, is the sum of all individual tooth frictional moments.

The frictional moment is then:

$$M_R = \sum_{i=0}^{i=z} \frac{2 \cdot T}{d \cdot z} \cdot Ai \cdot \mu = T \cdot \mu \cdot 0{,}637$$

This moment is overcome by a force couple applied in the axis of tooth meshes and acting in the plane of slewing. The value of the force is:

$$F_R = \frac{M_R}{H}$$

The same happens in the opposite set of teeth of a double link coupling. However, the slewing plane of this second set of teeth may lie in any direction relative to the first plane.

The shaft end of the coupled machines are therefore stressed by the frictional moment M_R and the radial forces F_{R1} and F_{R2}. The radial forces act in their respective planes.

The maximum load on the shaft end occurs when the misalignment conforms with Fig. 10. F_{R1} and F_{R2} then act in the same plane and direction. The overall bending moment acting on the shaft and at the point x has the value of:

$$M_b = M_R + (F_{R1} + F_{R2}) \cdot X = M_R + 2F_R \cdot X = M_R(1 + 2 \frac{X}{H})$$

These considerations are valid only for an absolutely even load distribution in the teeth and for small angular misalignments. For high speed couplings this can be assumed because of the high standard of accuracy demanded and the small allowable misalignments imposed for other reasons. The unknown in all these calculations is the coefficient of friction μ. Tests have been carried out to determine its values (6). The friction is dependent on:

Misalignment
Tooth geometry
Tooth load
Lubricant
Surface finish
Material combination
Lubrication

At the present stage of technical knowledge only an approximate value can be given. For a coupling which is run-in and working under normal conditions the coefficient of friction may be taken as $\mu = 0{,}01 - 0{,}15$. Should however the shaft alignment accidently be such that the coupling has no misalignment at all then there is no movement in the teeth, the lubrication of the teeth is impaired and the coefficient of friction rises to 0.2 - 2.25. This condition is also known as "torque-lock".

It is also possible for the running conditions to change unintentionally for the worse, for the coupling to become overloaded and consequently for the surface of the teeth to wear. The coefficient of friction under such circumstances may rise to $\mu = 0{,}3$.

The reason why some specifications stipulate to calculate with a coefficient of friction of $\mu = 0{,}3$ may be to prevent failure of a machinery component which is loaded by the coupling reaction in the event of coupling teeth surface deteriorating.

2.4.4 The Operation Limits of Toothed Couplings

The toothed coupling has, as any other technical product, no unlimited range of application. Torque and speed cause stresses in the coupling components and set natural limits. The torque carrying capacity follows approx. the relationship

$$T = C_1 \cdot d^2$$

The tangential stresses in the coupling parts are dependent on diameters and speed and follow approx. the equation

$$T = C_2 \cdot d^2 \cdot n^2$$

Torque and speed can now be brought into relationship

$$T_{max} = \frac{C_3}{n^2} \quad (1)$$

A second limit is the "inner" dynamic behaviour. The equation of Conti-Barbaran can be written

$$T_{min} = C_4 \cdot n^2 \quad (2)$$

Equation (1) and (2) are shown graphically in Fig. 11

This graph is only of qualitative value, and valid for a certain type and make of coupling. If the running conditions are within the shaded area a coupling of this particular design can be successfully applied. If the conditions are above curve 1 then the coupling components will be overstressed. If they are below curve 2 then the floating member of this particular coupling is too heavy. Curve 1 can be shifted upwards either by using a steel with higher ultimate tensile strength or by improving the production methods for the teeth, allowing a higher specific surface load. Curve 2 can be lowered by using a material with lower specific weight.

2.5 Conclusions

Only an ideal link can satisfy the requirements as stipulated in paragraph 2.2. We have seen that, due to natural laws, a toothed coupling can not work like an ideal link. However, the experience shows that if, at the design stage, all the factors mentioned are considered, a satisfactory performance of the toothed coupling can be expected. The large number of MAAG toothed couplings which have been supplied, particulary for turbo installations, and which are working troublefree, proves that toothed couplings can be successfully applied. With the trend to higher power and speed in the field of turbomachinery, it will become necessary, in future when designing new machinery plants, to regard the coupling as an integral part of the entire shaft system.

3 The MAAG-Toothed Coupling

3.1 The MAAG Technique

Basically, all our couplings are manufactured with the same quality; that is all the different types can be applied in high performance machinery. The combination of surface hardened with through hardened teeth has been chosen to reach a high load carrying capability, a high wear resistance and good sliding conditions. One exception is the high torque and low speed coupling, for example used in the cement mill drivers, for which the external teeth are not surface hardened.

3.1.1 The Teeth

Generally all external teeth are surface hardened and ground with slight barrelling, see Fig. 3 (exception as mentioned in 3.1). With the small amount of barrelling a high load carrying capacity is achieved. Heavy barrelling is not advantageous for high speed couplings, since the allowable angular misalignment is limited by the sliding velocity. The internal teeth are cut into the through hardened sleeve. To avoid distortions, normally no further heat treatment is carried out. For special applications the internally toothed components may be nitrided, attention is then paid to minimise the dimensional changes and distortions.

3.1.2 Materials

For the manufacture of the toothed coupling components, case hardening or direct hardenable steels are used. Spacers may be made of aluminium (anticorrosive) or titanium alloy if a low coupling weight is required. The external teeth are either induction hardened, case hardened, or nitrided. For high speed application alloay-steels are used exclusively for toothed components.

3.1.3 Dynamic Balancing

The coupling hubs and the floating member are balanced individually. If however, they are a bolted unit consisting of different pieces, then they will be balanced in the assembled condition. When designing the coupling components it is desirable to achieve direct support of the components on the rollers of the balancing machine. Balancing mandrels have to be used for balancing the hubs and certain coupling sleeves. Hubs with one keyway only are not balanced, these have to be balanced together with the shaft. But if such a hub has to be replaced it is practically impossible to re-balance the shaft on site. Therefore, two keys should always be used for shafts having a speed greater than 1500 min^{-1}.

Balancing mandrels are available or will be manufactured if required for:

1) Tapered bores with cones of 1:10, 1:16 (Nema cone), 1:20 and 1:24.
2) Cylindrical bores with standard diameters and ISO tolerances class H 6.

The balancing accuracy corresponds to the requirements of VDI-2060 quality class Q1.

3.1.4 Allowable Angular Misalignment

The allowable angular misalignment is dependent on backlash, sliding, velocity barrelling and loading. Since the tooth characteristics are fixed for the given standard sizes, the maximum allowable angular misalignment can be given for each size as a function of the speed, see Fig. 12.

The allowable angular misalignment is given for two running conditions:

1) Transient condition, for example during run up of the machinery.
2) Continuous condition, that is when all the expansion has settled.

The horizontal line in the graph represents the geometrical limit; the declining line is the limit given by the allowable sliding velocity.

For the transient condition, the barrelling has been taken into consideration when determining the maximum allowable misalignment, as allowable sliding velocity 0.16 m/s has been assumed. For the continuous condition the barrelling and the Hertzian defomation have been considered, and as allowable sliding velocity 0.12 m/s has been assumed.

Should the operating conditions require a larger misalignment then a more pronounced barrelling can be applied, within certain limits. The maximum allowable barrelling depends on the power, speed, and coupling size, therefore a generally applicable value cannot be given.

For low speed couplings, of the type ZCF (cement mill drive), the allowable misalignment is 4 mm per 1 m axial distance between the teeth (H).

3.2 Non-disengageable Toothed Couplings

These couplings are applied for connecting shafts which remain coupled during opearation. Fig. 13 shows a MAAG toothed type ZUD, with spacer and "anti sludge" lubrication, for turbo-machinery.