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# THE 'CREEP' OF ROTATING STRUCTURAL COMPONENTS

## 0 Introduction

By 'creep' of rotating structural components is understood the *unintended* displacement in relation to each other of two components which *rotate together* and which, in terms of static equilibrium, are *not loaded in the direction of creep.*

It was not until only a few years ago that this phenomenon first received attention, as a result of difficulties encountered in connection with the bearing (pivoting)axle system of a balance structure for a drawbridge. In this type of bridge the balance is pivoted on two shafts rotating in bearings which are mounted on top of the portal columns (see Fig. 1).

Apart from wind load, the only forces acting upon a structure of this kind are vertical external forces. Yet it has been established that, in a number of these bridges, the bearing shafts have undergone horizontal displacement, in conjunction with considerable deformation of the web of the side girder of the balance and deformation of the diaphragm. In a few cases fracture of the inner securing collar of a shaft even occurred.

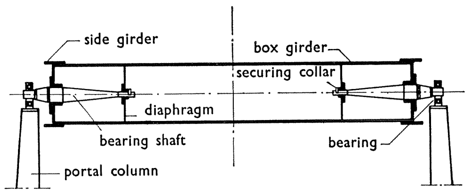

Fig. 1.

If measures are taken to prevent the displacements, a force will develop in the direction of creep, which force may be of very considerable magnitude in certain circumstances. In some cases it may even be as much as 30–40 % of the vertical force at the bearing, and values of substantially greater magnitude than this have been found to occur in tests.

It is a notable fact that this phenomenon was first detected only a short time ago. It has since come to be realised that it is most certainly necessary to take account of the phenomenon in the design and construction of such bridges.

An investigation into the causes of this so-called 'creep', their effects, and the remedial measures to be applied, has been conducted in the Stevin Laboratory.

## 1 Description of some tests performed

*Test 1*

A bush provided with two bearing surfaces is gripped in the chuck of a lathe. Inserned into this bush, with a certain amount of clearance, is a small shaft which projects a distance $L$ and is loaded by a weight $P$ suspended from a ball bearing (see Fig. 2). The shaft and the bush are of hardened and ground steel, in contrast with a previous test, in which they were both made of bright axle steel (in that test the results showed a considerable amount of scatter, however).

Two such small shafts, with different degrees of fit in relation to the bush, were tested: shaft I had a slightly looser fit than shaft II (see Figs. 2 and 3).

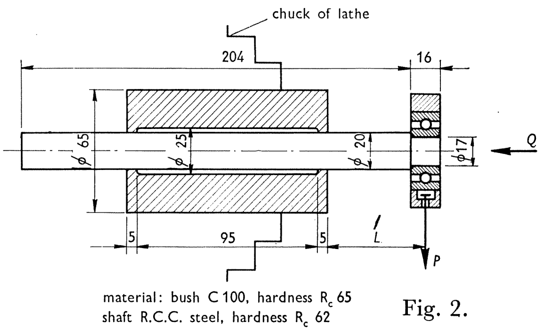


Fig. 2.

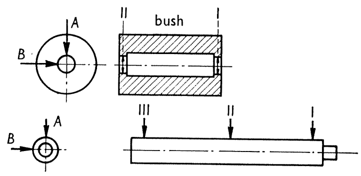


Fig. 3.

| | | A (mm) | B (mm) |
|---|---|---|---|
| Bush | I | 20.206 | 20.202 |
| | II | 20.196 | 20.220 |
| Shaft I | I | 20.176 | 20.176 |
| | II | 20.178 | 20.177 |
| | III | 20.180 | 20.180 |
| Shaft II | I | 20.192 | 20.193 |
| | II | 20.192 | 20.195 |
| | III | 20.189 | 20.194 |

By means of a mechanism comprising a crank and connecting rod the bush and shaft were rotated to and fro through an angle of 90° with a frequency of 22 to-and-fro movements per minute. In this test the load $P$ and the projecting length $L$ were varied.

The following quantities were measured:

a. the axial displacement $v$ of the shaft when the latter was allowed to move freely in that direction;
b. the axial force $Q$ that was needed to prevent displacement of the shaft.

It was anticipated that with the hardened and ground material the test results would show only a small amount of scatter, but this proved not to be the case: here again there was considerable scatter, while the shaft, despite its great hardness, became greatly roughened at the bearing surfaces.

Because of the diversity of the results obtained, there is no clear evidence of any effect of the variation of the projecting length $L$ and the load $P$. For the shaft I with $P = 10$ kg and $L = 5$ cm the greatest axial displacement measured per single oscillation was 1.55 $\mu$m, corresponding to approx. 3 $\mu$m per revolution.

For shaft II (with the tighter fit) the displacement as a function of the number of to-and-fro movements has been plotted in Fig. 4.

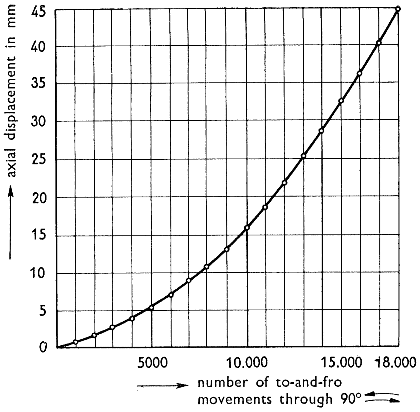


Fig. 4.

Under constant load $P = 62.5$ kg, and with a projecting length $L = 5$ cm at the start of the test, this length had increased to 9.5 cm after 18,000 to-and-fro movements.

As appears from the graph (Fig. 4), the displacement in the first 1000 alternations is, on an average, about 0.6 $\mu$m per to-and-fro movement. For the last 1000 alternations an average value of about 4$\mu$m per to-and-for movement was measured for the axial displacement.

The largest axial force needed to keep shaft I in its original position, for $P = 10$ kg and $L = 4$ cm, was found to be 0.74 kg, i.e., 7.4 % of $P$.

In the case of shaft II (with the tighter fit) an axial force of 41.5 kg, i.e., about 35% of $P$, was measured for $P = 120$ kg and $L = 5$ cm.

Further tests will have to show whether the higher percentage value of the axial force is due to the tighter fit or to the larger radial load $P$.

*Test 2*

In this test, too, a bush provided with two bearing surfaces is gripped in the chuck of a lathe, but now these surfaces are of quite different construction. A bearing diaphragm, situated nearest the load $P$, is formed by a 1.8 mm thick flexible perspex sheet, the object being to investigate the deformations of this diaphragm by means of the 'moiré' method. The other bearing is formed by a self-aligning ball bearing incorporating a so-called ball bushing (ball bearing

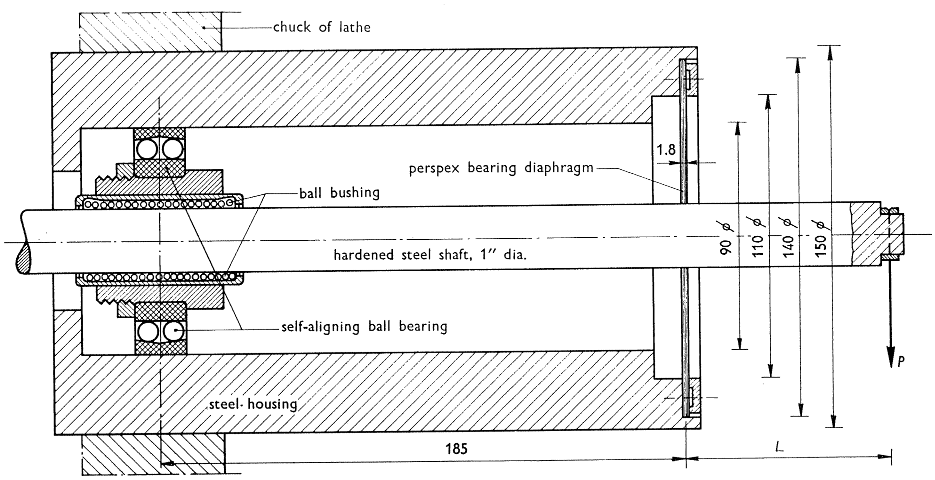


Fig. 5.

for axial movement), so that this bearing surface cannot exert moments or axial forces upon the shaft, but solely forces acting radially. In this way it is possible to investigate the effect of only one bearing surface, namely, the one formed by the perspex sheet.

Fig. 5 represents a section through the bush with the shaft and accessory parts.

In this test the following values were adopted for the load and the projecting lengh of the shaft respectively:

$$P = 25 \text{ kg}, \; L = 15 \text{ cm}.$$

With the aid of the moiré method the deformations of the perspex diaphragm due to the following causes were determined:

a. the load $P$ (stationary);
b. the load $P$ and 20 clockwise revolutions;
c. the load $P$ and 20 anti-clockwise revolutions.

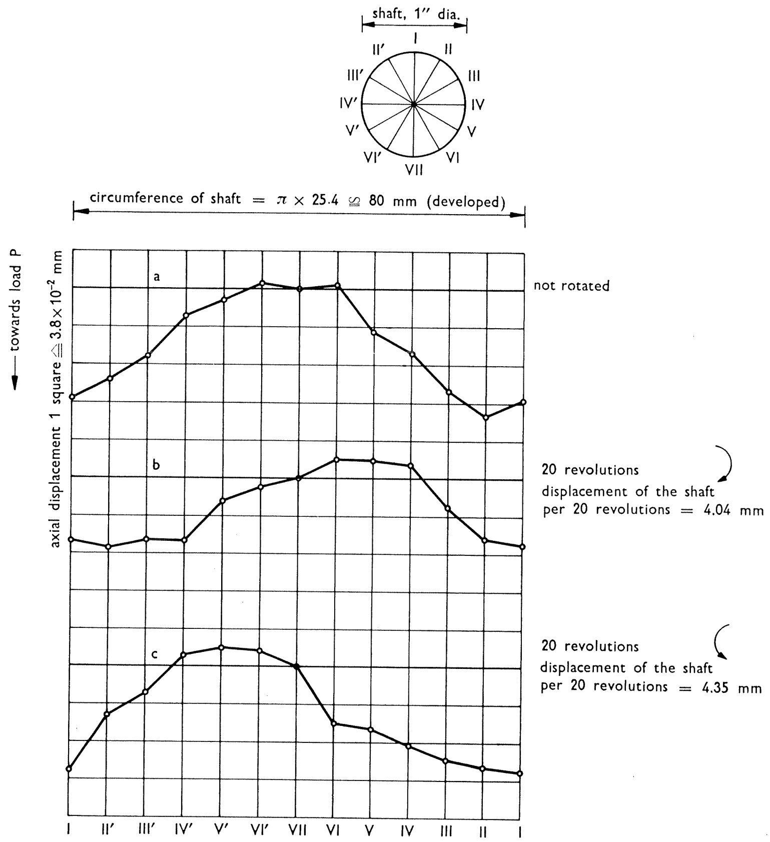


Fig. 6.

The displacements of the perspex sheet in the axial direction at the circumference of the shaft were deduced from the moiré photographs and have been plotted, in Fig. 6, for the three cases (a, b, c) in relation to point VII as the reference point.

From Fig. 6a it appears that, if only the load *P* is acting and the shaft is not rotating, the axial displacement of the perspex bearing diaphragm around the circumference of the shaft is practically symmetrical with regard to point VII (the lowest point of the shaft circumference).

It is apparent from Figs. 6b and 6c, however, that if the bush, with the loaded shaft end, is rotated, the perspex diaphragm undergoes an axial displacement which is asymmetrical with regard to point VII.

At the contact surface the perspex sheet adopts a slightly inclined position in relation to the shaft.

On comparing the two diagrams (Figs. 6b and 6c) it appears that the deviations of the perspex diaphragm from the purely transverse position at point VII are mutually contrary. Hence it follows that, as a result of the reversal of the direction of rotation, the slope of the perspex diaphragm in relation to the shaft is likwise reversed. This is in agreement with the theoretical explanation on page 36.

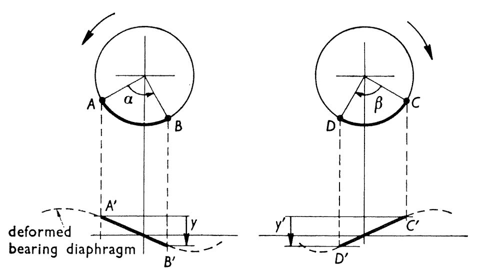


Fig. 7.

This accounts for the fact that, independently of the direction of rotation, the shaft always undergoes an axial displacement towards the load P.

The phenomenon can be clarified as follows (see Fig. 7).

Let A—B and C—D, for example, denote the arcs of contact between the diaphragm and the shaft. With anticlockwise rotation a point on the shaft will first come into bearing contact with the diaphragm, say, at A. If no slip between the diaphragm and the shaft occurs at the surface of contact, this point on the shaft will be carried along over the projected distance A′—B′ by the diaphragm, since the latter is not truly perpendicular to the shaft, but inclined in relation to it. Then this point on the shaft loses contact with the diaphragm at B. Thus, while rotating through an angle $\alpha$, the shaft has undergone a displacement $y$ in the axial direction. Similar considerations apply to clockwise rotation: in that case a point on the shaft will first come into contact with the diaphragm at C and will lose contact with it at D. The point in question thus moves along over the projected distance C′—D′. In the course of rotating through an angle $\beta$ the shaft has thus undergone a displacement $y'$ in the *same* direction as the displacement which previously occurred when the shaft was rotating in the opposite direction.

*Test 3*

In this test one end of the shaft is gripped in the chuck, while a bush (made of brass) is mounted freely, with a sliding fit, on the projecting shaft.

The load $P$ is carried by the shaft alone; the bush is not involved and will not be subjected to any forces except perhaps secondary forces and moments due to the curvature of the loaded shaft. In order to prevent damage to the shaft and also to achieve better gripping, the shaft end concerned is secured in the chuck with the interposition of a small steel bush.

The test arrangement is illustrated in Fig. 8.

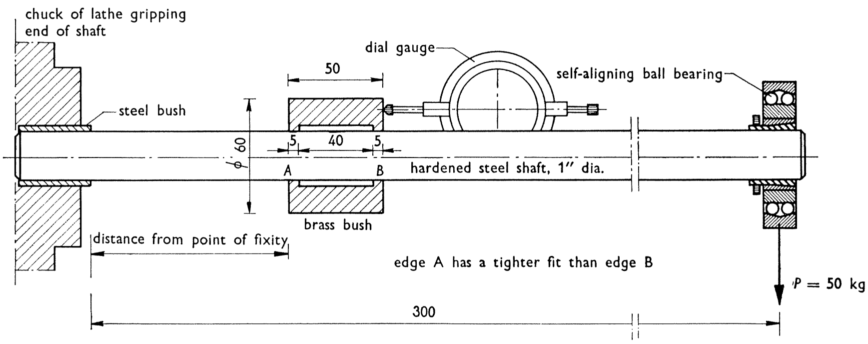


Fig. 8.

Although the brass bush is subjected essentially to the effect only of its own weight, an axial displacement of this bush in relation to the rotating shaft nevertheless occurred.

The explanation that can be advanced to account for this apparently strange behaviour is that secondary forces act upon the bush in consequence of the curvature of the shaft under the influence of the load. These forces are associated with the curvature of the shaft, the stiffness of the bush with its bearing surfaces, and the fit at these surfaces.

Suppose two thin discs are mounted some distance apart on a shaft. If a curvature is applied to the shaft, these discs will – since they are in no way hindered from doing so – retain their respective positions perpendicular to the shaft (see Fig. 9a).

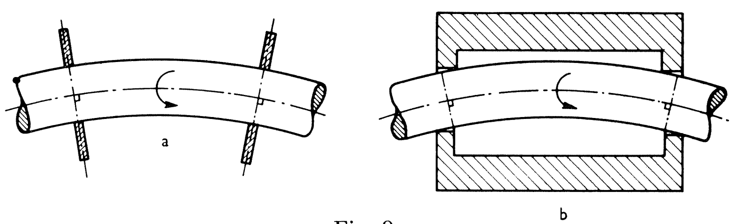


Fig. 9.

When the shaft rotates, the discs rotate in the same plane as the cross-section of the shaft at each disc.

However, if these discs are interconnected by a rigid element, as is the case with the bush envisaged here, then the rotation of the shaft will cause forces to act upon the discs, so that they (in this instance: the bearing surfaces) will acquire an inclined position in relation to the shaft (see Fig. 9b).

During rotation, the discs thus rotate in a different plane from that of the cross-section of the shaft at each disc. Consequently there is, besides the loading due to the dead weight of the bush itself, an additional set of forces at work. During rotation, these combined loads can result in an axial displacement: the bush, while taking part in the rotation, shifts along the (curved) shaft.

In this test it was endeavoured to determine the direction of creep of the bush placed with the bearing edge A on the side nearest the chuck and also with the edge B in that position. The edge A had a tighter fit than B.

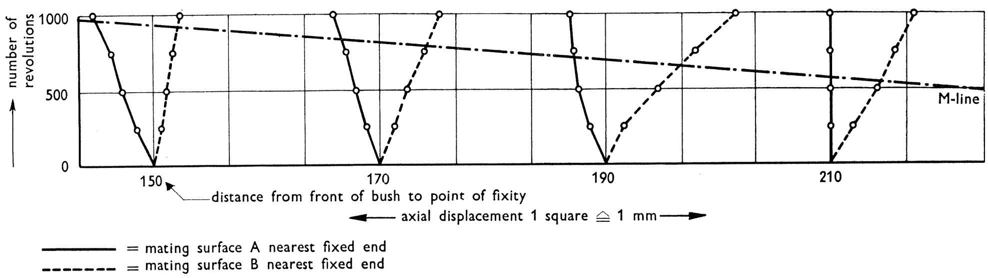


Fig. 10.

In Fig. 10 the measured displacements have been plotted in graph form. In this case the largest value that was observed for the axial displacement was about 1.8 $\mu$m per revolution. When the bush was placed in the reversed position on the shaft, axial displacement occurred in the opposite direction.

The test showed that the direction of creep can be reversed even when the difference in fit between the two bearing surfaces is very small. Having regard to tolerances, it will therefore hardly be possible, in a case like this, to predict the direction of creep in actual practice, unless the fit at one of the bearing surfaces is intentionally made looser than that at the other.

## 2 Theoretical explanation of the 'creep' of rotating structural components

Before going further into this problem, some simplifications will be introduced. It will be assumed that any frictional forces that occur between the shaft and the bush at the contact surfaces are directly proportional to the relative displacements of the points of contact from their position of equilibrium and that these forces are not affected by displacements of adjacent points.

To present this in a form that can be visualised, the bearing surfaces are conceived as consisting of radially disposed needles which are fixed in a rigid bush (see Fig. 11). Because of the curvature that the shaft acquires under the influence of the bending moment, a plane which intersects the shaft at right angles at the bearing surface will assume an inclined position in relation to the original vertical position.

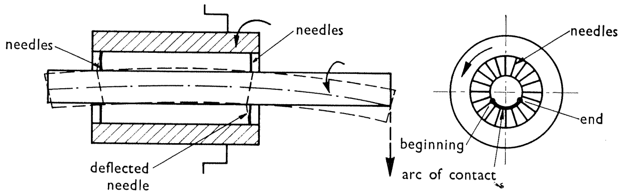


Fig. 11.

Let $\varphi_m$ denote this angle of inclination. Considering only the situation at the right-hand bearing surface, the condition shown in Fig. 12 will be obtained. As a result of the curvature of the shaft, the needles at the surface of contact are carried along by the rotating shaft and thereby undergo a certain deflection. If there were no axial displacement of the shaft, the maximum deflection would be $h_m - h_o$.

The deflection of the cylindrical surface along the lower half of the shaft, as indicated in Fig. 12, represents the pattern of the deflections and therefore of the forces acting upon the needles in the case where the shaft *undergoes no axial displacement.*

So the needles are subjected to a force directed to the left and the shaft is, consequently, subjected to a force acting in the opposite direction and undergoes a displacement to the right.

In consequence of an axial displacement of the shaft over a distance $s$ to the right per revolution, however, the needles will undergo deflections at the contact surface, as indicated in Fig. 13 representing the developed circumference.

With bending and axial displacement of the shaft these simultaneously occurring deflections must be superimposed. The resulting deflections of the needles will then be as shown in the lower diagram of Fig. 13.

On the assumptions that have been made, there will be equilibrium if the resultant of the forces acting to the left is equal to that of the forces acting to the right: the areas of the zones in which

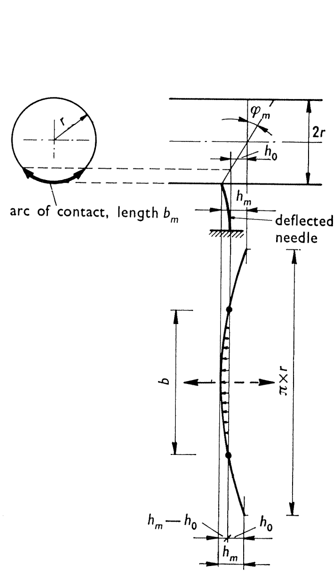


Fig. 12.

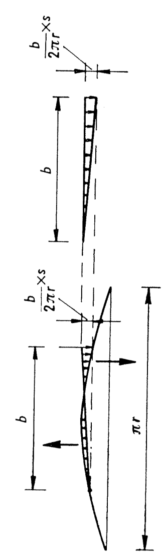


Fig. 13.

the small arrows have been drawn must therefore be equal. This is so if the area of the segment with base $b$ is equal to that of the triangle with the same length of base.

On equating these two areas, we obtain the pitch $s$ with which the shaft would, in the steady state, move in the axial direction. The following derivation is applicable to this case (see Fig. 14):

The sine curve is represented by:

$$y = h_m \sin \frac{x}{r}, \text{ while } x_0 = \frac{\pi}{2} r - \frac{b}{2}$$

Then: $$y_0 = h_m \cos \frac{b}{2r}$$

and the area of the segment is obtained from:

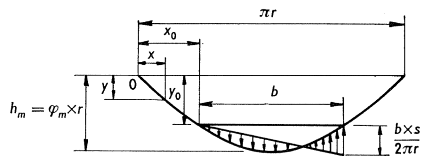


Fig. 14.

$$\int_{\frac{\pi r - b}{2}}^{\frac{\pi r + b}{2}} h_m \sin \frac{x}{r} \, dx - y_0 b = 2 h_m r \sin \frac{b}{2r} - h_m b \cos \frac{b}{2r}$$

For small values of $b/2r$ a sufficiently good approximation is obtained by considering only the first two terms of the expansion into a series, for a sine function and a cosine function respectively, whereby we obtain:

$$\text{area of segment} \approx \frac{1}{12} h_m \times \frac{b^3}{r^2}$$

The area of the triangle is:

$$\frac{1}{2} b \times \frac{bs}{2\pi r} = \frac{b^2 s}{4\pi r}$$

For an axial displacement of such magnitude that equilibrium of axial forces is achieved, the following relation applies:

$$\frac{1}{12} h_m \frac{b^3}{r^2} = \frac{b^2 s}{4\pi r}$$

Hence the pitch $s$ of the axial displacement is:

$$s = \frac{\pi \times b h_m}{3r}$$

## 3 Cases encountered in practice of axial movement manifested by structural components rotating together

### *3.1 Axial movement of the bearing shafts of the balance upon the portal columns of a drawbridge*

Fig. 15 shows a longitudinal section through the connecting box girder between the two I-section side girders, by means of which the balance structure is supported by journals resting in bearings mounted on the columns. When the bridge is raised and lowered (i.e., opened and closed), these journals and their shafts rotate with the balance. So this is a typical case of two components rotating *together*.

Although externally they are subjected only to radially directed forces (leaving wind load out of account), definite *axial* displacements towards the portal columns were nevertheless observed on these shafts.

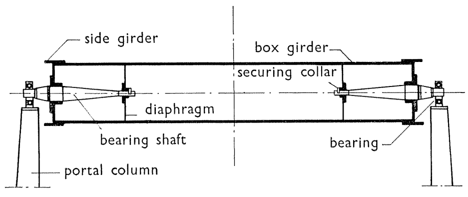


Fig. 15.

These displacements were attended by considerable deformations of the side girder web and of the diaphragm.

On inspection it was even found that the diaphragm had deflected about 15 mm at the shaft. The diaphragm was 30 mm thick, so that large axial forces had evidently been developed.

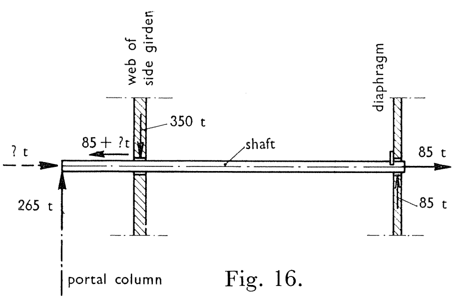


Fig. 16.

It was therefore necessary to take measures in order to ensure that these forces could in future be resisted without giving rise to objectionable or dangerous deformations. To this end, the bridge had to be dismantled, which provided an opportunity to carry out measurements on the diaphragms with a view to determining the magnitude of the axial force that had deformed them. The pattern of forces affecting the shaft, as deduced from these mesasurements, is indicated in Fig. 16.

### *3.2 Axial movement of the counterweight rope pulleys of a vertical-lift bridge*

Fig. 17 schematically illustrates a pulley, with its shaft and barrel-shaped roller bearings, which carried a wire rope for supporting the counterweight on a vertical-lift bridge and which underwent axial movement. There are eight such pulleys in all (four per lifting tower). After three months' service, the inner races of two of the barrel-shaped roller bearings were found to be fractured. On dismantling, it was established that the hub of the pulley had shifted on the shaft and had forced the inner race of the bearing forward on to the tapered thrust collar, causing fracture of the inner race.

The pulleys were originally not keyed to the shafts. After the occurrence of axial movement had been detected, three holes of 30 mm diameter were, in each pulley, drilled radially through the centre of the hub and to a certain depth into the shaft, cylindrical pins being inserted into these holes in order to secure the pulley to the shaft.

Because of the symmetry of construction, no axial movement should theoretically occur; that it did occur is attributable to the fact that in actual practice there never is perfect symmetry.

There are always asymmetrical loads, dimensional differences, unequal fits and variations in surface condition at the mating surfaces, etc., and these various differences act as disturbing factors.

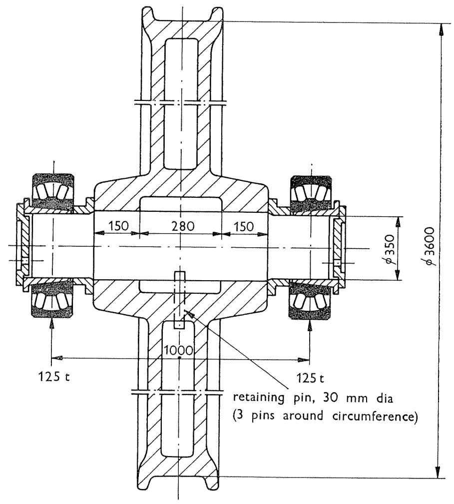


Fig. 17.

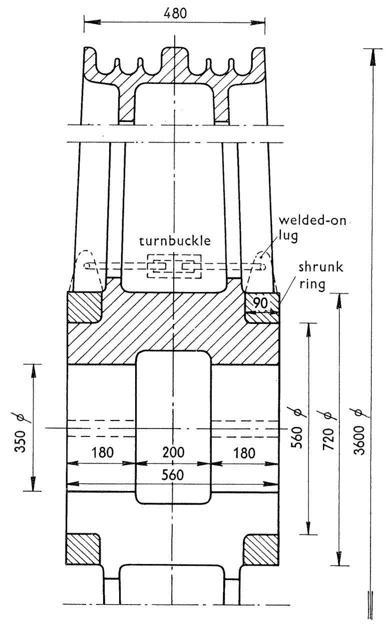


Fig. 18.

*3.3 Tangential and axial movement of a rope pulley on a vertical-lift railway bridge at Rotterdam*

On dismantling a pulley for the counterweight ropes on this vertical-lift bridge (see Fig. 18), the pulley was found to have undergone a tangential displacement on the shaft. As a result of this the keys and keyways in the shaft had sustained very severe damage (see Fig. 19).

The fact that here a tangential displacement occurred which gave rise to this damage is highly remarkable, considering that the pulley with the shaft can rotate freely in the barrel-shaped roller bearings (see Fig. 17), so that only the slight frictional moment of these bearings has to be transmitted from the pulley through the keys.

The tangential displacement of the pulley in relation to the shaft is probably due to a kind of 'rolling' of the shaft in the hub.

In the case where the shaft has a certain amount of play (clearance) within

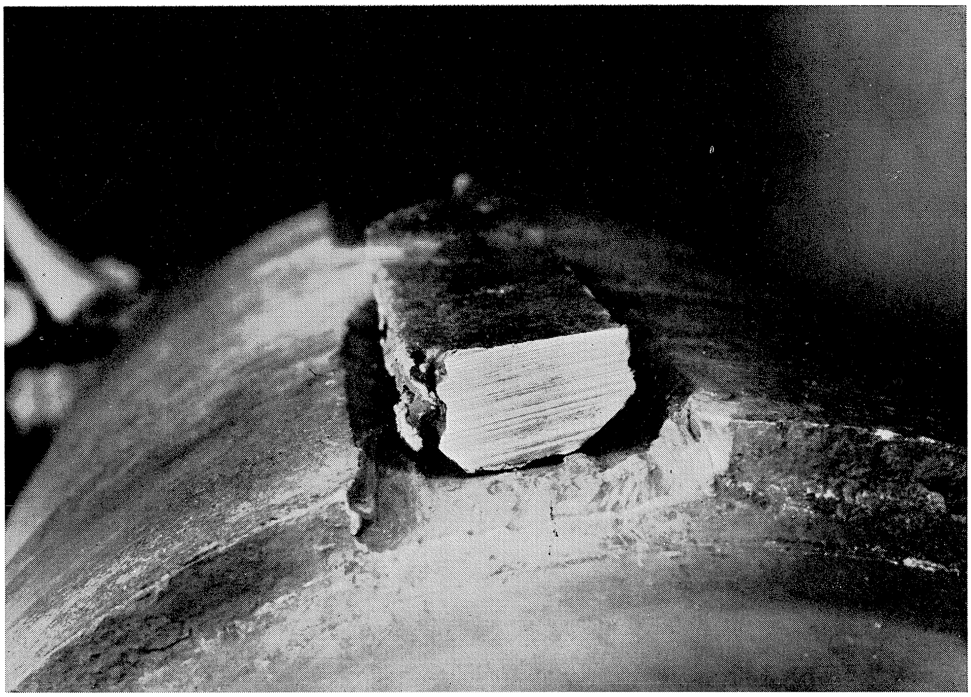

Fig. 19. Key in keyway. A very considerable amount of play developed between the key and the keyway.

the hub, the shaft will, if the two components rotate together and no slip occurs at the surface of contact, after a time have performed a larger number of revolutions than the pulley (because of the smaller circumference of the shaft). At first the key will prevent this displacement of the shaft in relation to the pulley (see Fig. 20). However, because of the large radial force and the high coefficient of friction (shaft surface attacked by corrosion in the course of years), a large frictional force is liable to develop at the surface of the shaft and to act upon the key. An approximate calculation yields the following result:

radial force 400 tons
coefficient of friction 0.4 (rusty surface)
frictional force $= 400 \times 0.4 = 160$ tons
force per key $= 80$ tons
pressure per unit area on key

$$= \frac{80{,}000}{1.2 \times 18} = 3.700 \text{ kg/cm}^2$$

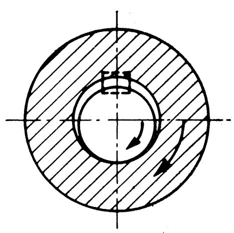

Fig. 20.

This is therefore indeed a high value.

Furthermore, the pulley first performs three or four revolutions in one direction and then three or four in the other direction. As a result of this rotation in alternate directions, some play between the key and the shaft probably developed in the course of years. Another possibility is that, in consequence of this reversal of the direction of rotation, the shaft, on attaining the maximum frictional force, comes into contact with the key sharply, so that an impact effect occurs.

This rope pulley manifested another unusual phenomenon. The hub is of split construction and shrunk on to the shaft with rings. The minimum shrinkage of these rings is 0.58 mm, so that it must be assumed that these rings are securely fixed to the hub. Yet some of the shrunk rings were found to have moved about 2 cm outwards, in the axial direction, on the hub. To prevent further movement, lugs were welded to the shrunk rings, and these lugs were joined together by means of threaded rods with left-hand and tight-hand screw threads interconnected by a turnbuckle.

## Summary

The following conclusions are to be drawn from the tests described in the foregoing and from the cases encountered in actual practice:

### *a. The direction of the axial displacement*

The direction of the displacement which two components, rotating together, may develop in relation to each other is in many cases difficult or impossible to predict in practice. The direction depends on factors whose magnitude is usually not known in advance and which include the difference in fit and surface condition of the bearing surfaces or the mating surfaces.

### *b. The magnitude of the axial force*

The axial force needed to prevent displacement may be fairly considerable. Indeed, in test 1 axial forces were measured which exceed 40% of the sum of the radial bearing reactions exerted by the bush upon the shaft.

In the case of movement in alternate directions the axial force that occurs will, under otherwise equal conditions, be larger than that occurring in the case of continuous movement.

### *c. Securing the components*

For components which rotate together one cannot rely entirely on the efficacy of a press fit or a shrink fit. Even if the only external forces involved are radial and/or tangential forces, it will be necessary to provide positive means of securing against axial displacement.

If the direction of the axial displacement is known, it is often possible to use merely a simple securing device, e.g., a fixed collar on the shaft (Fig. 21). At the mating surface of hub and shaft two forces are developed, one acting upon the shaft and the other acting upon the hub. As a result of the fixed collar, the two forces will be in equilibrium with each other (see dotted line).

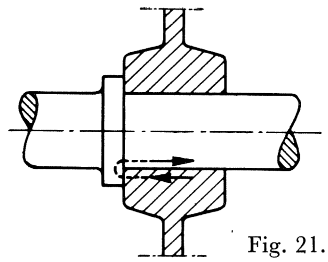

Fig. 21.

If the direction of the displacement is not known, then it will be necessary to secure the component by retaining devices on both sides.

In designing, it will in some cases also be necessary to take account of the possibility that the shaft may perform a 'rolling' motion within the hole (as in the case of the pulley on the vertical-lift bridge mentioned in the foregoing).

If a retaining or locking device is provided (e.g., a key), this phenomenon may, in certain circumstances, give rise to forces of considerable magnitude acting upon that device.