PATENT SPECIFICATION

Inventor: EDMUND RAMSAY WIGAN.

651,500

Date of filing Complete Specification: Oct. 13, 1949.

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Index at acceptance:—**Classes 80(ii)**, C5b1, C5c(1 : 2 : 3); **and 108(iii)**, H5a.

PROVISIONAL SPECIFICATION

SPECIFICATION No. 651500

INVENTOR:- EDMUND RAMSAY WIGAN

By a direction given under Section 17(1) of the Patents Act 1949 this application proceeded in the name of National Research Development Corporation, a British Corporation, of 1, Tilney Street, London, W.1.

THE PATENT OFFICE,
8th May, 1951

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at right angles to the first, on the driven shaft. The two slots are coupled by a solid piece bearing two raised surfaces sliding in the two slots. Such an arrangement allows axial misalignment but the axes must be parallel; it has friction and backlash.

According to the invention the sliding action is provided by sine springs such as described in the provisional or complete specification of my patent application No. 32770/45, these springs being set to zero or near zero stiffness in one plane. Each shaft bears a sine spring with its " preferred line of motion " perpendicular to the shaft axis and the two preferred lines of motion perpendicular to one another, i.e., their clamping centres being on lines at right angles to each other. The centres of the springs are coupled by a clamp. Although the clamp is rigid the torsional compliance of the sine springs will allow some lack of parallelism of the two shafts, while the zero (or near zero) stiffness of the sine springs allows the equivalent of a sliding action of the clamp on two planes at right angles should the shaft be misaligned.

Where mechanisms possessing some bearing friction (which is increased, of course, by any shaft misalignment) are driven by a coupling, the mechanism will remain at rest while the driver shaft is rotated until the torque, stored in the (and a corresponding increase of friction) the " alignment-stiffness " must be low.

This means that the resistance to " sliding " of the coupler element must be low while its resistance to torque is high. Sine springs have this feature, the torque resistance increasing with thickness and width while the sliding resistance can be made almost independent of the thickness for small movements.

In the case of very small shaft misalignments (say, 0.05 inch errors in shaft centres) the springs can be short (say, $1\frac{1}{2}$ inch long) and set to have a " bow " (amplitude of the sine curve) of the order of 1 to 2 times the centres error.

The angle θ_0 between the end of the spring and the line joining the points of clamping should be set so that

$$\text{Tan } \theta_0 = 2\pi \frac{A}{D}$$

where A = bow of spring
D = distance between points of clamping.

If this is done and the angles at both ends of the spring are equal the spring blade will be symmetrically between its clamps and the incremental stiffness will be zero.

As the centre point moves away from this position the stiffness increases as the square of the distance. This increase is also proportional to the width and the

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PROVISIONAL SPECIFICATION

Improvements relating to Shaft Couplings

I, THE MINISTER OF SUPPLY, of Shell Mex House, London, W.C.2, do hereby declare the nature of this invention to be as follows:—

The invention aims to provide an improved form of shaft-coupling which can be employed in place of the well known "cross-slide" coupling. This latter coupling usually has a slot in a plate carried on the driver shaft facing a slot, at right angles to the first, on the driven shaft. The two slots are coupled by a solid piece bearing two raised surfaces sliding in the two slots. Such an arrangement allows axial misalignment but the axes must be parallel; it has friction and backlash.

According to the invention the sliding action is provided by sine springs such as described in the provisional or complete specification of my patent application No. 32770/45, these springs being set to zero or near zero stiffness in one plane. Each shaft bears a sine spring with its "preferred line of motion" perpendicular to the shaft axis and the two preferred lines of motion perpendicular to one another, i.e., their clamping centres being on lines at right angles to each other. The centres of the springs are coupled by a clamp. Although the clamp is rigid the torsional compliance of the sine springs will allow some lack of parallelism of the two shafts, while the zero (or near zero) stiffness of the sine springs allows the equivalent of a sliding action of the clamp on two planes at right angles should the shaft be misaligned.

Where mechanisms possessing some bearing friction (which is increased, of course, by any shaft misalignment) are driven by a coupling, the mechanism will remain at rest while the driver shaft is rotated until the torque, stored in the bending of the coupler elements, is sufficient to equal the frictional torque.

To obtain high accuracy of transmission of the rotation of driver shaft to the driven shaft the "torque-stiffness" of the coupler must be high.

On the other hand to allow the two shafts to be misaligned, and locked together mechanically by the coupling without serious strain to the mechanism (and a corresponding increase of friction) the "alignment-stiffness" must be low.

This means that the resistance to "sliding" of the coupler element must be low while its resistance to torque is high. Sine springs have this feature, the torque resistance increasing with thickness and width while the sliding resistance can be made almost independent of the thickness for small movements.

In the case of very small shaft misalignments (say, 0.05 inch errors in shaft centres) the springs can be short (say, $1\frac{1}{2}$ inch long) and set to have a "bow" (amplitude of the sine curve) of the order of 1 to 2 times the centres error.

The angle θ_0 between the end of the spring and the line joining the points of clamping should be set so that

$$\text{Tan } \theta_0 = 2\pi \frac{A}{D}$$

where A = bow of spring
D = distance between points of clamping.

If this is done and the angles at both ends of the spring are equal the spring blade will be symmetrically between its clamps and the incremental stiffness will be zero.

As the centre point moves away from this position the stiffness increases as the square of the distance. This increase is also proportional to the width and the

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cube of the thickness of the blade (i.e., to the stiffness of the blade in the ordinary sense).

It is not possible, therefore, to use extremely thick blades (so as to raise torsional resistance) without increasing somewhat the sliding resistance when the alignment errors are large.

However, the greater the bow (A) of the blade the less the stiffness increment due to displacement from the central position.

To judge the length of blade required to give a required value of bow (A) the following formula may be used:—

$$\frac{A}{D} = \sqrt{q}\,(0.3209 + 0.09\,q)$$

where $q = \dfrac{L-D}{D}$ and L = length of blade measured along the sine curve between point of clamping, D = straight line between clamping centres. (The quantity L—D is the " excess " length as it were).

However, the formula above refers to a blade without a central clamp, and this is needed to couple it to the second sine spring. When a clamp is applied to the middle of the blade it is probably safe to assume that the lengths to be inserted on the formula above should be the exposed surface of the blade and its equivalent along the straight line between clamping points; i.e., L and D must both be decreased to allow for the space occupied by the clamp.

There seems to be no limit set to the size of the clamp; the wider the clamp, the greater the torsional resistance of the coupling, but the greater L and D must be in order to obtain the required zero stiffness, and value of A.

It is recommended that the bow (A) of the springs should be 1 to 2 times the maximum expected shaft alignment error.

The bow should be kept down if the greatest possible torque resistance is essential, but the best design will probably be arrived at in each case by starting with a bow of the order of twice the shaft error and then thickening the material till the torque resistance is high enough.

If the shafts are not parallel the two halves of the coupling should be separated by an amount roughly proportional to the error angle and the clamp or coupler member joining the two springs, extended.

If the angle error is very large the coupler may have to incorporate a flat blade with its plane parallel to one of the sine spring fixings.

Dated the 21st day of October, 1948.

S. W. SLAUGHTER,
Agent for the Applicant.

COMPLETE SPECIFICATION

Improvements relating to Shaft Couplings

I, THE MINISTER OF SUPPLY, of Shell Mex House, London, W.C.2, do hereby declare the nature of this invention and in what manner the same is to be performed, to be particularly described and ascertained in and by the following statement:—

This invention relates primarily to a shaft coupling of a form which can be employed in place of the well-known Oldham coupling. Its application may, however, be extended to the coupling of shafts which are not parallel. The invention involves the use of springs of sinusoidal form (hereinafter referred to as sine springs) of the character described in Specification No. 617,076. In the present specification use will be made of the term " preferred line of motion " of a sine spring. In the case of the present invention this will be taken to be a line through the centre of the spring, perpendicular to the centre line of the support thereof and lying in the same plane as the " bows " of the spring.

According to the invention, a shaft coupling comprises two sine springs each restrained in a predetermined configuration by means such as a bracket or the like adapted for connection to one of the shafts to be coupled, the two springs being connected together at their centres so that their preferred lines of motion lie at right angles to each other in two parallel planes, the coupling being constructed and arranged so that when it is used for connecting two parallel shafts the said two parallel planes are perpendicular to the axes of the shafts. The springs are normally set to zero or near zero stiffness along their preferred line of motion and connected at their centres by a rigid clamp. The zero or near zero stiffness of the springs allows the equivalent of the sliding action in two planes at right angles which occurs in the Oldham coupling, while the torsional compliance of the sine springs will allow for some lack of parallelism of the two shafts to be coupled.

Where mechanisms possessing some bearing friction (which is increased, of course, by any shaft misalignment) are driven by a coupling, the mechanism will remain at rest while the driver shaft is rotated until the torque, stored in the bending of the coupler elements, is sufficient to equal the frictional torque.

To obtain high accuracy of transmission of the rotation of driver shaft to the driven shaft, the "torque-stiffness" of the coupler must be high.

On the other hand, to allow the two shafts to be misaligned and locked together mechanically by the coupling without serious strain to the mechanism (and a corresponding increase of friction) the "alignment-stiffness" must be low.

This means that the resistance to "sliding" of the coupler element must be low while its resistance to torque is high. Sine springs have this feature, the torque resistance increasing with thickness and width while the sliding resistance can be made almost independent of the thickness for small movements.

In the case of very small shaft misalignments (say, 0.05 inch errors in shaft centres) the springs can be short (say, $1\frac{1}{2}$ inch long) and set to have a "bow" (amplitude of the sine curve) of the order of 1 to 2 times the centres error.

The angle θ_0 between the end of the spring and the line joining the points of clamping of the spring to its restraining bracket should be set so that

$$\operatorname{Tan}\theta_0 = 2\pi\frac{A}{D}$$

where A = bow of spring
D = distance between points of clamping.

If this is done and the angles at both ends of the spring are equal the spring blade will lie symmetrically between its clamps and the incremental stiffness will be zero, that is to say, a vanishingly small force applied to the blade at right angles to the line of centres will result in a finite deflection in the same direction as the force.

As the centre point moves away from this position the stiffness increases as the square of the distance moved. This increase is also proportional to the width and the cube of the thickness of the blade (i.e., to the stiffness of the blade in the ordinary sense).

It is not possible, therefore, to use extremely thick blades (so as to raise torsional resistance) without increasing somewhat the sliding resistance when the alignment errors are large.

However, the greater the bow (A) of the blade the less the stiffness increment due to a given displacement from the central position.

To judge the length of blade required to give a required value of bow (A) the following formula may be used:—

$$\frac{A}{D} = \sqrt{q}\,(0.3209 + 0.09\,q)$$

where $q = \frac{L-D}{D}$ and L = length of blade measured along the sine curve between said clamping points, D = straight line between clamping points.

However, the formula above refers to a blade without a central clamp, and this is needed to couple it to the second sine spring. When a clamp is applied to the middle of the blade it is probably safe to assume that the lengths to be inserted on the formula above should be the exposed surface of the blade and its equivalent along the straight line between clamping points; i.e., L and D must both be decreased to allow for the space occupied by the clamp.

There seems to be no limit set to the size of the clamp; the wider the clamp, the greater the torsional resistance of the coupling, but the greater L and D must be in order to obtain the required zero stiffness, and value of A.

The bow (A) of the springs should be of the order of shaft alignment error.

The bow should be kept down if the greatest possible torque resistance is essential, but the best design will probably be arrived at in each case by starting with a bow of the order of twice the shaft error and then thickening the material till the torque resistance is high enough.

In order that the invention may be clearly understood, reference will now be made to the accompanying drawing.

Figures 1, 2 and 3 are end views of a practical form of coupling according to the invention connecting two parallel shafts and show the positions of the various component parts of the coupling after a rotation of the shafts through 45° (figure 2) and 90° (figure 3) from the initial position shown in figure 1. Figure 4 shows a suitable member for connecting the centres of the two sine springs.

Two parallel shafts are shown at 1 and 2. Sine springs 3 and 4 are clamped at the ends of the arms of **Y** brackets 5 and 6, the stems (not shown) of which are in the form of hollow bosses which are clamped by screws or other means to the ends of the shafts 1 and 2. The stiffness of the springs 3 and 4 along their preferred lines of motion is made small or zero by adjusting either the angles at which the springs are clamped to the arms or by altering the lengths of the spring blades or by both means. The member 7, slotted at both ends with the slot at one end at right angles to the other, is provided with suitable clamping screws or other means (not shown) to clamp the springs 3 and 4 rigidly in each slot.

In figure 1 the **Y** bracket 6 is at right angles to the line joining the centres of the two shafts 1 and 2, and in this position the spring 4 is deflected and the spring 3 undeflected. In the intermediate position shown in figure 2 both springs are deflected. After a 90° rotation of the shafts the spring 3 is deflected and the spring 4 undeflected (figure 3).

In the case of two shafts with centres offset, but which are not parallel, the member 7 is made long enough to couple the centres of the two springs without applying too great a twisting movement to them. Further, the member 7 may be secured to each of the blades by a single pin at right angles to the slot, allowing freedom of movement of the spring in the plane of the slot. Alternatively, the connecting member for the two springs may be in the form of a blade twisted through 90° at its centre and secured rigidly to the centres of the springs. The compliance of the blade, together with the compliance (to twisting forces) of the springs 3 and 4, suffices when the shafts 1 and 2 are but slightly offset and are very nearly parallel.

Having now particularly described and ascertained the nature of my said invention and in what manner the same is to be performed, I declare that what I claim is:—

1. A shaft coupling comprising two sine springs each restrained in a predetermined configuration by means such as a bracket or the like adapted for connection to one of the shafts to be coupled, the two springs being connected together at their centres so that their preferred lines of motion lie at right angles to each other in parallel planes, the coupling being constructed and arranged so that when it is used for connecting two parallel shafts the said two parallel planes are perpendicular to the axes of the shafts.

2. A shaft coupling according to Claim 1, in which each of the sine springs is restrained in its predetermined configuration by a **Y** bracket having means at the end of each fork for clamping the ends of the spring thereto, its stem being in the form of a hollow boss adapted for the reception and clamping of one of the shafts to be coupled.

3. A shaft coupling according to Claim 1 or Claim 2, in which the two springs are clamped together at their centres by a connecting element slotted at both ends with the slot at one end at right angles to that at the other and provided with a clamping means such as a screw to clamp the spring rigidly in each slot.

4. A shaft coupling according to Claim 1 or Claim 2, in which the two springs are connected together at their centres by a connecting element slotted at both ends with the slot at one end at right angles to that at the other, the springs being secured in each slot by a single pin or the like at right angles to the plane of the slot so that swivelling of each spring about the axis of the pin is permitted.

5. A shaft coupling according to Claim 1 or Claim 2, in which the two springs are connected together at their centres by a blade member twisted through a right angle.

6. A shaft coupling constructed and arranged substantially as described and shown in the accompanying drawing.

Dated this 13th day of October, 1949.
S. W. SLAUGHTER,
Agent for the Applicant.

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[This Drawing is a reproduction of the Original on a reduced scale.]

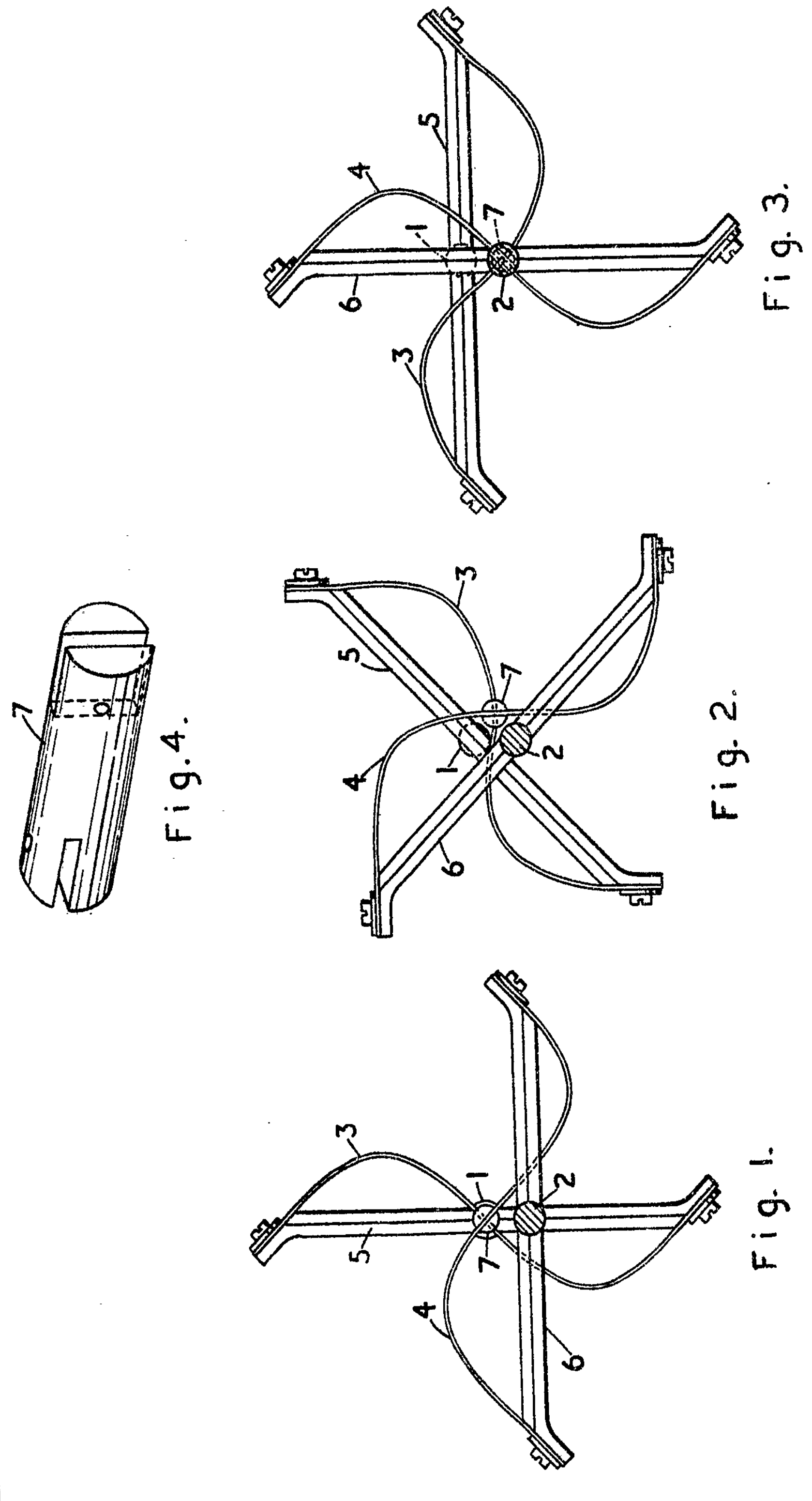

Fig. 1. Fig. 2. Fig. 3. Fig. 4.