A variable-speed transmission for use between two substantially parallel shafts for permitting continuous drive engagement during continuous speed-variation, and during power-variation of one shaft and corresponding change in operation of the other shaft.
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Fig. 2
Fig. 10
VARIABLE SPEED TRANSMISSION MECHANISM

The invention relates to a variable speed transmission mechanism between two substantially parallel shafts which each carry a guide wheel over which a friction ring passes. A pair of flanks of the ring engage a pair of flanks of at least one of the guide wheels, while a change in the spacing between the flanks of a pair of flanks causes a change in the position of the ring relative to the guide wheel and hence in the transmission ratio.

The shafts to be coupled carry split guide wheels the two parts of which are axially displaceable against spring force. The friction ring is mounted for rotation in a frame which is displaceable by means of an operating mechanism.

Similar transmission mechanisms are known which are fitted with an operating mechanism which enables the spacing between the parts of the guide wheels and hence the position of the ring and consequently the transmission ratio to be varied.

A disadvantage of these known variable-speed transmission mechanisms is the complicated and hence expensive and vulnerable construction of the mechanism for varying the transmission ratio. Moreover, the forces exerted on the ring by the moment to be transmitted from one shaft to the other frequently have to be absorbed entirely or largely by the said mechanism so that the mechanism must be of heavy construction and bulky.

It is an object of the invention to avoid the said disadvantages, and the construction according to the invention is characterized in that in order to vary the position of the ring relative to one of the guide wheels, the ring is adapted to be tilted about the other guide wheel.

The variation of the transmission ratio may be adjustable at will by means of a construction which is characterized in that the friction ring is also passed around an adjusting wheel which is displaceable by means of an operating mechanism.

The transmission mechanism may be made self-adjusting in an embodiment which is characterized in that the friction ring is exclusively supported by the guide wheels on the two shafts and is automatically adjustable by forces exerted on it by the guide wheel flanks.

A preferred embodiment is characterized in that the guide wheel about which the ring is pivotal takes the form of an externally toothed gearwheel, the friction ring being provided with corresponding internal teeth.

A particular embodiment is characterized in that the friction ring is composed of two annular elements which together form an annular unit and are resiliently joined together.

The annular element may be joined together either by resilient elements which are uniformly distributed around the circumference or by a rim made of a resilient material.

A construction which is continuously adjustable within very wide limits is characterized in that the transmission mechanism is combined with a differential gear, the motion of the input shaft being divided into two motions, namely that of the output shaft and that of a third shaft, which motions are in a variable ratio to one another, the third shaft being coupled to one of the two other shafts by the variable transmission mechanism.

Such a combination with a differential gear may in particular take a form which is characterized in that the differential mechanism comprises an epicyclic gearing with a sun gear on the input shaft and at least one planet gear coupled to the output shaft, the annulus having external teeth which mesh with a gearwheel on the third shaft, the guide wheels of the variable transmission mechanism being mounted on the third shaft and on the shaft coupled to the third shaft respectively, while the friction ring passing over the guide wheels occupies a position which determines the transmission ratio and which is variable under the influence of the moment to be transmitted by the third shaft.

Embodiments of the invention will now be described, by way of example, with reference to the accompanying diagrammatic drawings:

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is an axial sectional view of a variable transmission mechanism,
FIG. 2 is a sectional view taken at right angles to the shaft on the line II—II of FIG. 1,
FIG. 3 shows schematically the friction ring of FIG. 1 in two positions,
FIG. 4 is a sectional view of a variable transmission mechanism including a separate adjusting wheel, taken at right angles to the shafts,
FIG. 5 is a sectional view of part of a specific embodiment of a split guide wheel and a fitting friction ring,
FIG. 6 is a sectional view in a plane containing the shafts of a variable transmission mechanism having a friction ring composed of two annular elements which are resiliently joined together,
FIGS. 7 and 8 are sectional views of modifications of the guide wheel and the friction ring,
FIG. 9 is a sectional view taken in a plane containing the shafts of a transmission mechanism according to the invention in which an epicyclic gearing is combined with a variable-speed transmission mechanism having a self-adjusting friction ring passing over guide wheels,
FIG. 10 is a sectional view taken on the line X—X of FIG. 9,
FIG. 11 is a schematic perspective view of the transmission mechanism of FIG. 9,
FIG. 12 is a graph in which the relationship between the moment applied to the output shaft and the ratio between the peripheral velocities of the input and output shafts is plotted.

OF THE PREFERRED EMBODIMENT

Referring now to FIGS. 1 and 2, guide wheels 1 and 2 are mounted on shafts 3 and 4 respectively to be coupled. A metal friction ring 5 passes over the guide wheels 1 and 2. The ring 5 has internal teeth which mesh with corresponding external teeth of the guide wheel 2. The guide wheel 1 is composed of two parts 7 and 8 between which the ring 5 is disposed. The radial distance of the shaft 3 from the points of contact 9 between the ring 5 and each of the parts 7 and 8 determines the transmission ratio in transmitting the rotation movement from the shaft 3 to the shaft 4 and vice versa. The part 8 of the guide wheel 1 is integral with the shaft 3. A shaft 10 is aligned with the shaft 3. The shafts 3 and 10 are interconnected by a bolt 11 so as to be axially adjustable. A spring abutment 12 is rigidly secured to the shaft 10. Cup-shaped spring elements 13 are tensioned between the spring abutment 12 and the
part 7 of the guide wheel 1. As a result, the part 7 is adapted to slide axially on the shaft 3, so that the position of the ring relative to the two parts 7 and 8 of the guide wheel 1 and hence the locations of the contact points 9 which determine the transmission ratio are variable. This change in position of the ring 5 relative to the split guide wheel 1 may be considered as a pivoting movement of the ring about the guide wheel 2. During the pivoting movement the center M of the ring 5 moves along the arc of a circle having its center in the axis of the shaft 4. As will be described more fully hereinafter with reference to FIG. 3, the transmission mechanism may be designed so that this change in position is automatically effected under the influence of the forces exerted on the ring by the guide wheels.

FIG. 3 shows schematically the ring 5 in two positions relative to the guide wheels 1 and 2 and also the forces exerted by the guide wheel 1 on the ring 5. One of these two positions is shown by a broken line and the reference characters relating to this position are distinguished by a prime. Assuming the shaft 3 which carries the guide wheel 1 to be the driving shaft and the direction of rotation to be as indicated by the arrows, in the first position of the ring 5, in which position the center of the ring is at M, the guide wheel 1 exerts a tangential force $K_4$ and a radial force $K_r$ on the ring 5 in the contact point 9. In the stationary condition the moment of the resultant of the forces $K_4$ and $K_r$ about a theoretical point 14 of contact between the ring 5 and the guide wheel 2 is zero, i.e. $K_r A = K_4 b$. If the force $K_r$ increases, for example because the resisting torque applied to the driven shaft 4 is increased, the moment $K_r A$ will become greater than $K_4 b$ and the ring will move to a new equilibrium position, for example in the position in which the center is located at $M'$. Thus, the contact point 9 has been moved to 9' at a smaller radial distance from the shaft 3. The two parts 7 and 8 of the guide wheel 1, which in a cross-sectional view enclose a groove which tapers towards the shaft, are thrust apart against the action of spring elements 13 (cf. FIG. 1 also). A new position of equilibrium is reached because with respect to the new contact point 14' of the ring 5 and the guide wheel 2 the moment arm $a'$ of the force $K_r$ is smaller than the moment arm $a$ for the force $K_4$, while the moment arm $b'$ of the force $K_4$ has increased with respect to the moment arm $b$ of the force $K_r$. Moreover, in general the force $K_r$ will change on movement of the ring 5 relative to the parts 7 and 8, and in the embodiment shown in FIG. 1, $K_r'$ will be greater than $K_r$. However, the connection between the parts 7 and 8 by means of spring elements may be differently designed, for example in a manner such that the force which these spring elements exert on one another is constant, irrespective of the relative positions of the parts 7 and 8.

Thus a self-adjusting transmission mechanism is obtained which on increase of the resisting torque applied to the driven shaft 4, will automatically adjust itself to a lower speed for this shaft. Consequently, an increase in the power to be transmitted is counteracted, which is also due to the fact that the greater force $K_r'$ has a smaller moment arm with respect to the shaft 3 than has the force $K_r$. Conversely, a decrease in the power to be transmitted will also be counteracted, so that in this embodiment the transmission mechanism will tend to maintain constant the power to be transmitted and hence in general the speed of the driving shaft 3.

In addition to the above effect tending to automatically maintain constant the speed of the driving shaft, the transmission mechanism may also be used for maintaining constant the speed of the driven shaft. In this case the shaft 4 carrying the guide wheel 2 is the driving shaft and the directions of rotation of the guide wheels 1 and 2 are opposite to the directions of the arrows. An increase in the force $K_r$, for example due to an increase of the resisting torque of the shaft 3, again causes a displacement of the contact point 9 to 9', which in general will be accompanied by a decrease in the speed of the shaft 4. A corresponding decrease in the speed of the shaft 3 is counteracted, because the contact point 9' is spaced from the shaft 3 by a smaller radial distance than is the contact point 9.

The guide wheel 2 need not be in the form of a gearwheel, but it may alternatively be a roller or pulley having a groove for the ring formed in it, the movement being transmitted from the ring 5 to the guide wheel 2 and vice versa by means of frictional forces. The only condition is that the ring is adapted to pivot about this guide wheel 2.

Alternatively, both guide wheels 1 and 2 may be split in the manner described hereinbefore with respect to the guide wheel 1. When the guide wheels 1 and 2 are identically shaped, changing conditions will theoretically cause the ring to change its position relative to the two guide wheels in the same sense and in the same degree, so that the transmission ratio remains unchanged. If, however, the two guide wheels 1 and 2 differ from one another, for example, in the slopes of the friction flanks or in the properties of the springs by which the guide wheel parts are coupled to one another, in general automatic adjustment of the transmission ratio by the ring will nevertheless be possible. In this case the change in position of the ring may be described as a combination of a translation and a rotation about one of the guide wheels.

The self-adjusting transmission mechanism described with reference to FIGS. 1 to 3 is distinguished by its simple construction. There is no need for special mechanisms for displacing the ring, for intercoupling the guide wheels or guide wheel parts or for displacing one or both shafts. Hence, the afore-described variable transmission mechanism may be built in a highly compact form.

FIG. 4 shows an embodiment in which the ring automatically adjusts itself but in which the position of the ring may also be determined by means of a roller 23 over which the ring runs and which may be displaced from the exterior. The roller 23 is mounted on an arm 24 of a bell-crank 25 which is pivotable about a pivot 26. The position of the roller 23 is adjustable by means of a Bowden cable 27, which passes through a casing 28 and is secured to the end of the other arm 29 of the bell-crank 25. The roller 23 may alternatively be disposed so as to engage the outer surface instead of the inner surface of the ring 5. If the inner surface of the ring 5 carries teeth, the roller 23 may be replaced by a gearwheel. Another alternative is for the roller 23 to run in a circumferential groove traversing the teeth. If the limb 24 carries an additional roller disposed opposite the roller 23 and engaging the outer surface of the ring 5, the position of the ring 5 and hence the transmission ratio between the shafts 3 and 4 is completely determined by the position of the bell crank 25. The position of the bell crank may be manually adjusted.
FIG. 5 shows an embodiment using a friction ring 15 of U-shaped cross-section. Flanks 16 and 17 at the inner surface of the limbs of this ring 15 engage flanks 18 and 19 of the parts 20 and 21 of a guide wheel. The guide-wheel parts 20 and 21 are joined together by spring elements 22 and are axially movable relative to one another, so that the position of the annular transmission member 15 is again variable. A ring 15 having a cross-section as shown in FIG. 5 may, for example, be used in a variable transmission mechanism of the afore-described type. The guide wheel composed of parts 20, 21 and 22 will then be substituted for the guide wheel 1 of FIGS. 1 to 4.

If large powers are to be transmitted, several pairs of guide wheels 1 and 2 may be mounted on the shafts 3 and 4 respectively, a ring 5 or 15 running over each pair. The self-adjusting transmission mechanism shown in FIGS. 1 to 3 is particularly suited for such a parallel arrangement. In such an arrangement the rings 5 or 15 will automatically adjust themselves to the same transmission ratio because, if a ring tends to depart from it, forces will act on this ring which will immediately correct the deviation in the afore-described manner.

FIG. 6 is a cross-sectional view of an embodiment taken in a plane containing the shafts, which embodiment includes a friction ring consisting of two annular elements 30 and 31 joined together by spring elements 32. In this embodiment, a guide wheel 33 on the shaft 3 needs not comprise two relatively displaceable parts and has a groove formed in it which tapers towards the center of the wheel 33. The shaft 4 carries a guide wheel 34 in which likewise a groove for the friction ring has been formed. The transmission of motion between the guide wheels and the ring is effected by means of the frictional forces between the flanks of the ring and the walls of the grooves in the guide wheels. Moreover, the inner surfaces of the annular elements 30 and 31 are in contact with the bottom 35 of the groove in the guide wheel 34 and this contact also will produce frictional forces. Hence the grooved guide wheel 34 on the shaft 4 may be replaced by a friction ring.

Also, the annular elements 30 and 31 may be provided with internal teeth which mesh with a gearwheel on the shaft 4.

The annular elements 30 and 31 are adapted to tilt about the guide wheel 34, so that in complete correspondence with the principle of the embodiments described with reference to FIGS. 1 to 4 the transmission ratio is variable.

FIG. 7 shows an embodiment in which the friction ring likewise is composed of two annular elements 36 and 37 which again are joined together by spring elements 38. These spring elements 38 exert forces directed towards each other on the elements 36 and 37, so that these elements are clamped onto sloping flanks 39 and 40 of a guide wheel 41 on the shaft 3. FIG. 8 shows a modification of the ring of FIG. 7. In this modified ring, the two annular elements 42 and 43 are joined together by a resilient ring 44. The elements 42 and 43 and the ring 44 form an integral unit made of a material having elastic properties, for example, a synthetic material. The elements 42 and 43 are reinforced by metal cores 45 and 46 respectively. A variable transmission mechanism according to the principle described with reference to FIGS. 1 to 4 may again be readily realized by means of the embodiments shown in FIGS. 7 and 8.

A particularly compact transmission mechanism the transmission ratio of which is variable within very wide limits is obtainable by combining a variable transmission mechanism of the aforedescribed type with a differential gear. Such a combination is shown in FIGS. 9 to 11.

The transmission mechanism includes an epicyclic gearing which intercouples shafts 47, 48 and 49. The input shaft 47 is driven, for example by an electric motor 50, and its other end carries a sun gear 51 of the epicyclic gearing. A planet gear 52 of the gearing meshes with the sun wheel 51. The planet gear 52 is mounted for rotation about a spindle 53 one end of which is secured in an end part 54 of the output shaft 48, which end part 54 has a greater diameter than the remainder of the shaft 48. An annulus 55 of the epicyclic gearing has internal teeth and external teeth which mesh with the planet gear 52 and with a third gear wheel 56 on the third shaft 49 respectively. The epicyclic gearing further has three supporting gearwheels 57 each mounted for rotation about a corresponding spindle 58. One end of each spindle 58 is rigidly secured in the end part 54 of the output shaft 48.

The shafts 47 and 49 are also coupled to one another by a metal friction ring 59 which passes over two guide wheels 60 and 61. The guide wheel 60 is rigidly mounted on the shaft 47 and has the form of a gearwheel meshing with internal teeth of the ring 59. The guide wheel 61 is split and the two parts 62 and 63 are axially displaceable with respect to another in the manner described with reference to the embodiments shown in FIGS. 1 to 4.

The output shaft 48 carries a gearwheel 64 for driving a tool. In response to the resisting torque acting on the shaft 48 the transmission mechanism will automatically adjust itself. If the variable-speed mechanism is set to a given transmission ratio, the ratio between the speeds of the shafts 47, 48 and 49 is uniquely determined. In the stationary condition, the ratio between the moments to be transmitted by the shafts 47, 48 and 49 is also uniquely determined by the dimensions of the sun gear 51, the planet gear 52, the annulus 55 and the gearwheel 56. If the moment to be transmitted by the shaft 48 changes, for example because the driven tool requires another moment, the moment to be transmitted by the shaft 49 will be proportionately changed.

The change in the moment acting on the shaft 49 is used for automatic adjustment of the position of the ring 59 between the parts 62 and 63 of the guide wheel 61 and hence of the transmission ratio in the aforedescribed manner. FIG. 11 shows the ring in two positions which correspond to different transmission ratios, the ring in the position shown by broken lines being designated by 59'. The location of the variable speed transmission mechanism which automatically responds to the moment to be transmitted is variable. Instead of being interposed between the shafts 47 and 49 it may be used to couple the shafts 48 and 49. This permits of obtaining transmission mechanisms having different properties, which may be adapted to the requirements to be satisfied by the tool to be driven. In the graph shown in FIG. 12, a curve 1 shows the relationship between the moment $M_y$ acting on the output shaft 48 and the transmission ratio between the input shaft 47 and the output shaft 48 in the embodiment described with reference to FIGS. 9 to 11. The transmission ratio is defined as $\omega_o/\omega_i$, where $\omega_o$ and $\omega_i$ are the angular ve
locities of the input and output shafts 47 and 48 respectively. The curve I shows that a finite moment acting on the output shaft is maintained even if the speed of the output shaft is zero. This property may advantageously be used for driving different devices such as mixers, mills, drills, lawn mowers, boats, motor cars, motor bicycles and the like. If the outgoing shaft meets an obstruction such that its speed becomes zero, the input shaft and the power source coupled to it will remain revolving, whilst a finite torque keeps acting on the output shaft. This prevents the transmission mechanism from breaking itself up in the case of the output shaft being obstructed. Since at zero speed of the outgoing shaft this shaft does not transmit power, the input shaft need not supply power apart from the losses produced in the transmission mechanism itself. The mechanism contains a so-called “circulating power” which is circulated via the input shaft 47, the differential gearing the third shaft 49 and the variable-speed transmission mechanism.

If in the embodiment described with reference to FIGS. 9 to 11, the drive is coupled to the third shaft 49, which as a result actually becomes the input shaft whilst the former input shaft takes the place of the third shaft, then curve II of FIG. 12 is obtained. This curve shows an initial increase of the transmission ratio with increase of the moment acting on the output shaft, whilst beyond a given value the transmission ratio continues to increase with decrease of the said moment.

What is claimed is:

1. A transmission mechanism for continuous and variable speed change between first and second shafts which are in substantially parallel fixed relative positions, comprising a first guide wheel keyed to said first shaft, a friction ring substantially in said plane and encircling the first wheel, with an inner edge of the ring in drive engagement with an adjacent outer edge of said wheel, a second guide wheel comprising two disc parts axially spaced apart, at least one disc spring-biased toward the other, the discs rotatable on said second shaft with at least one disc keyed to said second shaft, said ring having a portion thereof situated between and in frictional engagement with said two discs, said ring being pivotable about said first wheel with an edge of the ring always in contact with said first wheel, and a second portion of the ring engaging said discs along varying radially spaced parts thereof, said ring automatically varying its radial position relative to said second wheel according to the forces exerted on said ring by said wheel, with a resulting change in the speed ratio between said first and second shafts.

2. A variable-speed transmission mechanism as claimed in claim 1 further comprising an adjusting wheel rotatably mounted in said frame, said ring circumscribes the adjusting wheel, the mechanism further comprising means for displacing said adjusting wheel to pivot said ring to a different position.

3. A variable-speed transmission mechanism as claimed in claim 1 characterized in that the first guide wheel about which the ring pivots comprises an externally toothed gearwheel, and the friction ring has corresponding internal teeth.

4. A variable-speed transmission mechanism as claimed in claim 1 wherein the second wheel further comprises spring elements uniformly distributed around the circumferences of said discs.

5. A variable-speed transmission mechanism as claimed in claim 1 wherein the second wheel further comprises a rim made of a resilient material for biasing together said discs.

6. A transmission mechanism for continuous, variable speed change between first and second shafts, which are mounted in substantially parallel, fixed relative positions in a frame, comprising: first and second wheels respectively keyed to said shafts, a friction ring circumscribing the first wheel in constant driving engagement therewith, and frictionally engaging the second wheel at positions spaced at variable radial distances from the axis of said second wheel, the radial position of said ring relative to said second wheel being automatically varied as compensation for variation of the forces exerted by the second wheel on said ring.

7. A variable-speed transmission mechanism between two substantially parallel shafts rotatably mounted in a frame, comprising: first and second wheels keyed to said first and second shafts respectively, the two wheels in a plane that is generally normal to said axes of said shafts and wheels, the first wheel having a circumferential contact edge, the second wheel having a friction surface generally in said plane, and a friction ring that circumscribes the first wheel in drive engagement with said contact edge, and contacts the second wheel in frictional engagement with said friction surface, a moment-arm distance being established from the two points of engagement of said ring with said two wheels, said ring being pivotable about said first wheel such that a remote part of the ring contacts said second wheel's frictional surface at a variable radial distance from its axis with the frictional forces from said contact being operative on the resulting moment-arm distance to the pivot point on said first wheel, said frictional contact point and corresponding forces being automatically variable with a change in speed or load between said shafts.

8. A variable-speed transmission mechanism as claimed in claim 1 characterized in that the first wheel about which the ring pivots comprises an externally toothed gearwheel, and the friction ring has corresponding internal teeth.
UNited States Patent Office
Certificate of Correction

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It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

In the heading "[30] Foreign Application Priority Data"
"Mar. 26, 1970" should be --April 1, 1970--

Signed and sealed this 9th day of April 1974.

(SEAL)
Attest:

EDWARD M. FLETCHER, JR. G. MARSHALL DANN
Attesting Officer Commissioner of Patents