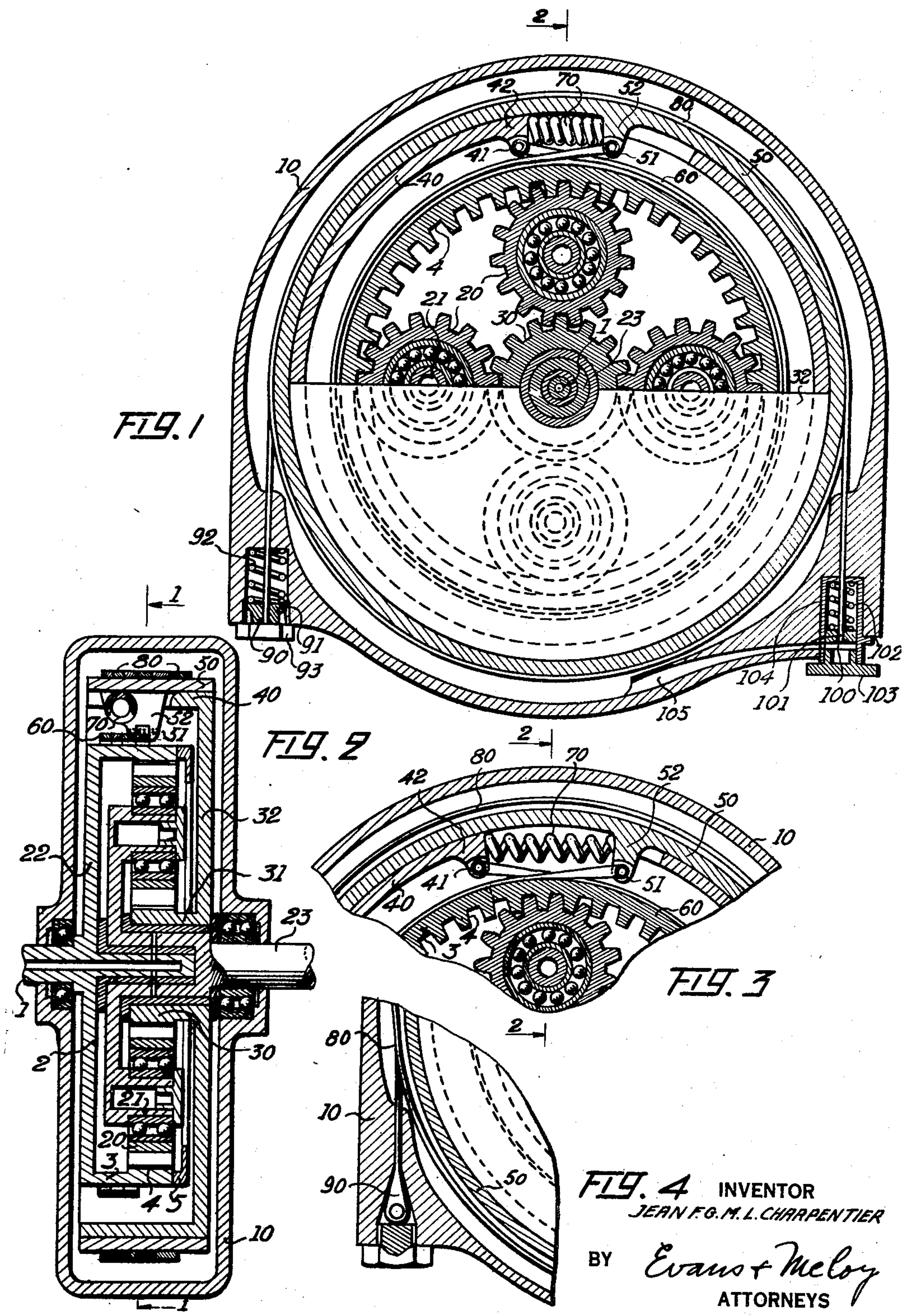
Sept. 29, 1953 J. F. G. M. L. CHARPENTIER 2,653,489

AUTOMATIC TORQUE TRANSFORMER

Filed July 5, 1949 3 Sheets-Sheet 1



INVENTOR
JEAN F.G.M.L. CHARPENTIER

BY Evans & McCoy
ATTORNEYS

AUTOMATIC TORQUE TRANSFORMER

Filed July 5, 1949 3 Sheets-Sheet 2

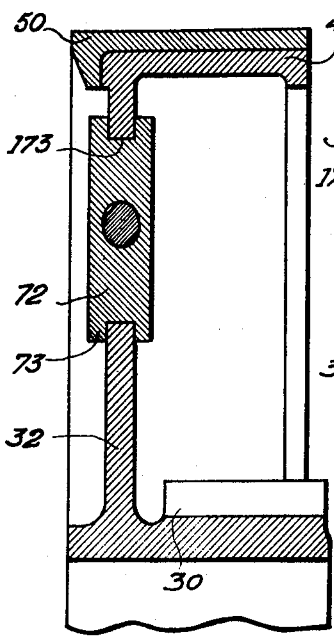


FIG. 6

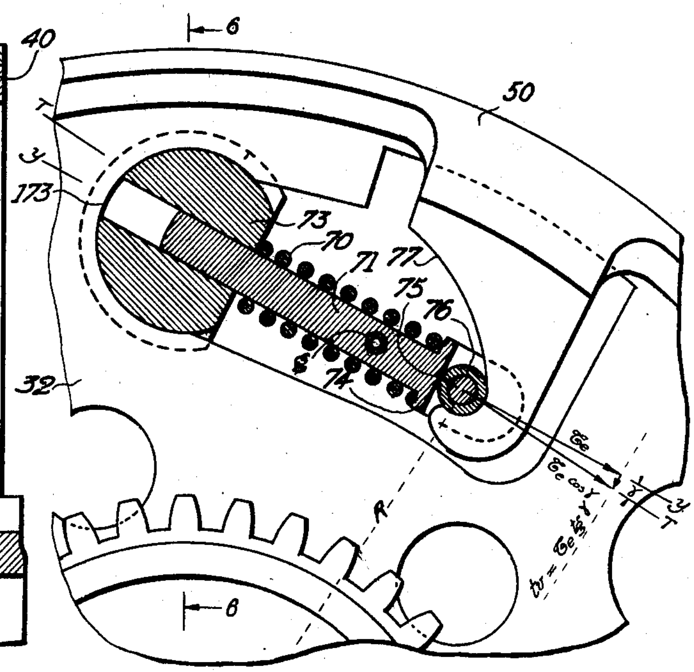


FIG. 5

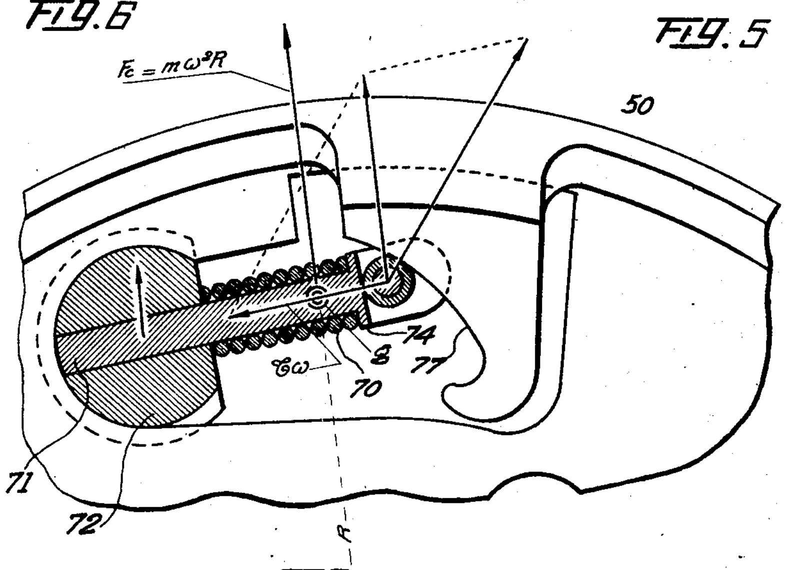


FIG. 7

INVENTOR
JEAN F. G. M. L. CHARPENTIER
BY Evans & McCoy
ATTORNEYS

AUTOMATIC TORQUE TRANSFORMER

Filed July 5, 1949 3 Sheets-Sheet 3

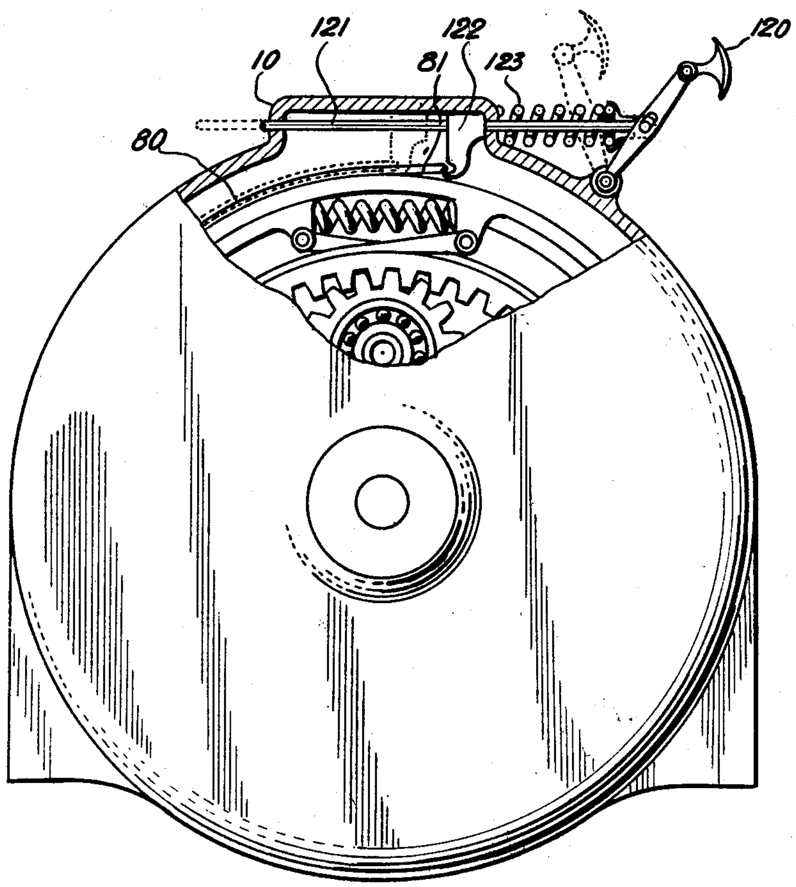


FIG. 8

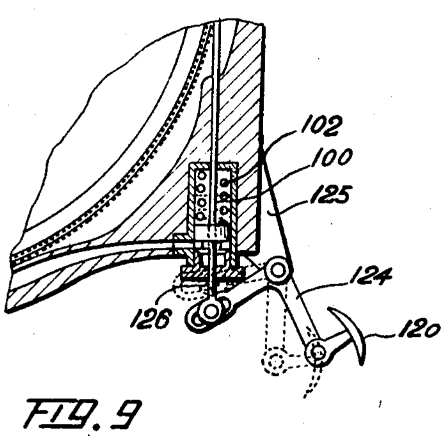


FIG. 9

INVENTOR
JEAN F.G.M.L. CHARPENTIER

BY Evans & McCoy
ATTORNEYS

Patented Sept. 29, 1953

2,653,489

UNITED STATES PATENT OFFICE

2,653,489

AUTOMATIC TORQUE TRANSFORMER

Jean F. G. M. L. Charpentier, St. Cloud, France

Application July 5, 1949, Serial No. 102,992
In France July 12, 1948

7 Claims. (Cl. 74—751)

1

This invention relates to torque transforming gears and more especially to a gear unit capable of transforming torque automatically between a motor or other driving element and a driven element in this sense that it automatically maintains stability of the equilibrium of operation when the opposed turning moments (torques) vary in respect of their relative intensities.

For the purpose of achieving this, the new device according to the invention utilizes only the variations of intensity of the intrinsic forces which are developed on its organs by the influence of the movement.

The variations are:

1. The variations of the intensity of the driving torque;
2. The variations of the intensity of the driven torque;
3. The variations of the intensity of the centrifugal force;
4. The variations of the intensity of the inertia forces;
5. The variations of the intensity of friction.

These variations are utilized in both directions for enabling the device connecting the driving and driver groups, viewed from a state of equilibrium of operation, to fulfill the following reversible function:

1. The intensity of the driven torque rises.

In this case the function of the new device consists therein, to automatically lend to the driving element parallelly to the rise of intensity of the driven torque, one or a plurality of supporting points and adding their actions, thereby to enable it to multiply the intensity of its torque as many times as required by the driven torque to become equilibrated.

2. The intensity of the driven torque drops.

In this case the function of the device reverses itself so as to automatically and parallelly to the drop of this intensity withdraw the supporting systems which multiplied the intensity of the driving torque until it became overabundant.

In the device according to the invention the supporting systems constitute the elements for the transformation of the torque and, consequently, of the angular speed, these elements being similarly constituted as far as the principle of the invention goes.

Each of these elements enables the driving torque to be transmitted, as to intensity, either identically to itself or in a definite ratio of multiplication.

The device comprises a number of elements which is equal to the number of the factors of multiplication, the product of which imparts to the final transformed driving torque that intensity which corresponds to the maximum intensity attainable by the driven torque.

2

In the drawings affixed to this specification and forming part thereof several embodiments of this invention are illustrated diagrammatically by way of example.

It should however be understood that in these examples every torque transforming gear unit comprises parts well known as such and other parts which are specific of this invention, and that these specific parts may as well form part of other assemblies without thereby leaving the orbit of this invention.

In the drawings:

Fig. 1 is an elevational view from the direction of the arrows 1—1 of Fig. 2, with parts broken away and parts shown in section of a transforming gear unit embodying my invention; the position of the members assures transmission of the driving torque at a multiplication greater than 1 and smaller than 2.

Fig. 2 is a vertical cross sectional view along the line 2—2 of Fig. 1.

Fig. 3 is a view of a portion of the unit shown in Figs. 1 and 2 with the spring in elongated position as in high speed or direct drive.

Fig. 4 is an elevational view partly in section of a portion of a modified form of transforming gear unit showing a rigid anchor for the brake means instead of the spring or resilient anchor shown in Fig. 1.

Fig. 5 is an elevational view with parts broken away from and parts partly in section of a portion of a modified form of a transforming gear unit wherein the torque controllable clutch actuating means such as the parts shown in section is controllable by centrifugal force. The spring is shown in elongated position as at rest or at very small driven torque.

Fig. 6 is a sectional view on the line 6—6 of Fig. 5.

Fig. 7 is a view similar to that of Fig. 5 of the portion shown in Fig. 5 with the spring of the torque controllable clutch actuating means in compressed position but with the driving and driven shafts locked together as when the driven torque is relatively low and the rotational speed is high.

Fig. 8 is an elevational view with parts broken away of a torque transforming gear unit equipped with pedal operative clutching means.

Fig. 9 is a view partly in section of a portion of my transforming gear unit with the pedal attached at another position.

Referring to the drawings, the torque transforming gear unit comprises a train of hypo-epicycloidal gears in combination with the specific members forming part of this invention. In this combination the driving torque is transmitted to the device located inside the casing 10 by means of the driving shaft 1 which carries along the drum 2 of cylindrical circumference 3 which carries the internal gear ring 4. This gear ring 4 meshes with a number of pinions 20 mounted on axles 21 which are secured to a disc 22 which is keyed on the driven shaft 23 that extends through the casing 10. The system of gears hereabove described is the hypo-cycloidal part of the train.

The pinions 20 are in mesh as planet gears with a sun gear 30. This system constitutes the epicycloidal part of the train.

The sun gear 30 is mounted on the hub 31 of a disc 32 carrying the cylindrical drum 40.

The hub 31, the sun gear 30 and the drum 40 can freely rotate around the driven shaft 23. The drum 40 of the sun gear 30 embraces with a clearance the driving drum 3. A second drum 50 embraces this drum completely and is free to turn about it. The internal driving gear 4, the hub 31 of the sun gear 30, the driven shaft 23 and the drums 40 and 50 are arranged coaxially.

The two drums 40 and 50 are coupled on the one hand by means of a clutch band 60 and on the other by a spring 70. The band 60 is wound in a spiral around the cylindrical outer surface of the driving drum 3, its ends 41, 51 are fixed to arms 42, 52 mounted on the drums 40 and 50, respectively. The ends 41, 51 of the band are crossed as shown in Figs. 2 and 3 so that the tension of the spring 70, which is being compressed between the arms 41 and 51 of the drums, tightly pulls the band 60 around the drum 3 of the driving ring 4.

The cylindrical surface of the drum 50 is spirally embraced by a band 80, both ends 90 and 100 of which are fixed to the casing. The end 90 may either be fixed to the casing rigidly, as shown in Figure 4, or by means of a spring 92 as shown in Fig. 1, while the other end 100 may either be free or attached to the casing by a spring 101. The operation of this device is analyzed beginning with the starting phase when the driven torque C_r has an intensity always greater than that of the driving torque. As mentioned above, Figs. 2 and 3 represent the device as viewed from the side of the driven shaft 23 and the direction of rotation of the driving shaft if defined by that of the tangential driving force t_m is in this case the positive trigonometric direction.

The driving torque C_m which is transmitted to the torque transforming gear thru the annulus 4 with the radius pitch R_3, exerts on the teeth of each of the n planet gears 20 a fraction t_m of the tangential driving force T_m

$$t_m=\frac{C_m}{n.R_3}$$

The planetary gears 20 carry along their support such as the disc 22 (Fig. 2), to whose shaft 23 is applied the resistance torque C_r which exerts on each of the "n" axes 21 of the planetary gears 20 a resistance R to being carried along. ($C_r=n.R.R_2$) where R_2 is the radius of the circle passing thru all of the axes of said planetary gears.

For the carrying along of their axis 21 in the direction of movement impressed on them by the ring 4, the pinions 20 take support from the sun gear 30, while exerting on it the tangential force

$$-t_m=\frac{-C_m}{n.R_3}$$

and receiving the reactions

$$r'=-(-t_m)=t_m$$

Under the influence of the thrust which corresponds to the tangential actions, $-t_m$ of the planet gears, the sun gear 30 tends to rotate in the direction opposed to that of the driving element. In this tendency to move it carries along the drum 40 fixed to it. The drum 40 then exerts on the spring 70 a tension

$$-T=-\Sigma t_m\frac{R_1}{R_4}$$

Where R_1 and R_4 are respectively the pitch radius of the sun gear and the radius of the circle to which the center line of spring 70 is tangent wherein R_4 is the effective radius of the drum, which acts on the spring 70, while R_1 is the pitch radius of the sun gear 30.

The spring 70 transmits the tension T to the drum 50 which also tends to turn in a direction counter to that of the driving member. At that point the band 80 enters into action, which is a brake whose winding is so arranged that its braking effect is abolished by unwinding, when the drum 50 tends to rotate in the same sense as the driving shaft. However, in order that this braking effect become automatically and instantaneously active and block the drum 50, when it tends to rotate in a sense counter to that of the driving shaft, which corresponds exactly to the case here in question, the spring 70, being subjected to the compression force T, simultaneously creates an effect proportional to the intensity of the force t, which effect enables the points 41 and 51, at which the ends of the band 60 (the clutch band) are attached, to approach each other. This approach of the two points 41, 51 results in a relief of the clutch band 60. Being thus relieved, the clutch band also relieves the drum 40 (which is the drum of reaction) and the drum 50 (which is the supporting drum) of all effort, by the driving drum 3, to carry them and, together with them, the sun gear 30 along which is integral with the two drums. Under these conditions the internal driving gear 4, which is moving at an angular speed ω_m corresponding to the driving torque C_m, carries along the planet gears 20, which act on the immobile sun gear 30 with the effect of carrying along the disc 22, on which they are mounted, and the driven shaft 23 fixed to it, at the speed

$$\omega_r=\frac{\omega_m}{1+\frac{R_1}{R_3}}$$

under the condition that

$$(0.5\times\omega_m)<\omega_r\omega_m$$

where ω_r is the rotational speed of the driven shaft and ω_m is the speed of the driving shaft transmitting to the shaft 23 a transformed driving torque

$$C_r=C_m\left(1-\frac{R_2}{R_3}\right)\text{ with }C_m\leq C_r\leq 2C_m$$

This first explanation presents the torque transforming gear, which operates as a multiplier of the driving torque, an operation which occurs in the same way as that of a classical train of gears of the same composition.

Fig. 3 illustrates the operation of the torque transforming gear, when the intensity of the driven torque C_r has been reduced until it has become equal to that of the maximum intensity of the driving torque C_m.

Starting from the case of Fig. 1, when the intensity of the driven torque drops, the spring **70** (Fig. 3) expands, reducing its effect and moving asunder the attaching points **41** and **51** of the clutch band **60**, whereby this band is forced to approach the driving drum **3** which it embraces, when the equality of the intensities of the driving and driven torques is established. The spring **70** then attains its maximum elongation, to which corresponds the tension t_0, which is designed to assure to the clutch band **60** the winding pressure required to obtain the relative blocking of the elements of the train of gears. The blocking, by abolishing the relative speeds of all the members which constitute the torque transforming gear unit, determines the carrying along of the driven shaft **23** at the angular speed ω_m of the driving shaft **1**.

Obviously the device can operate only if the initial tension t_0 of the spring, which is necessary for effecting the blocking of the train of gears when the intensities of the opposing torques are equal in absolute values, is lower than the tension produced by the reaction drum **40**. The intensity of the driven torque then becomes greater than the driving torque. This condition is necessary in order that the band **60** can relieve the drum **4** from its tension. By utilizing a spiral clutch band **60** and by using a larger or lesser number of turns in the spiral one may obtain any desired clutching force between the band **60** and the drum **40** and still have a spring which has the desired tension, t_0, to be compressed at the desired torque so that the drum **40** have release for rotation at a speed different from the rotation speed of the driven shaft.

With the tension t_0 the relative blocking of the train of gears can be obtained in the following manner:

The axis of the spring extends tangentially to the device. Consequently the effect of its tension t_0 on each of its two points of fixation is affected by a sign as follows:

On point **41** there acts the tension $+t_0$,
On point **51** the tension $-t_0$.

Since the system is carried along by the drum **4** in the positive sense, the relative tension $+t_0$ which acts on the point **41** of the band **60**, exerts on the opposed point **51** and on the drum **50** a positive traction effect T which as function of a figure $\alpha=\pi.n$ of the encircling windings, has the intensity: $T=t_0e^{f.\alpha}$.

This positive traction $+$T which assures the carrying along of the entire reaction system (pinion **30**, drums **40** and **50**) by the driving ring **4**, for the coupling traction T.

The coupling traction T is determined in the folowing manner:

The transformer device is considered as functioning as a multiplier of the driving torque C_m,μ being the multiplication factor of the driving torque C_m, such as $C_r=C$, in the present case, where the ring **4** is the driving element,

$$\mu \text{ is}=1+\frac{R_1}{R_3} \text{ with } 1<\mu<2$$

Supposing that the device has a stable function, satisfying the condition,

$$\frac{dC_m}{dC_r}>0$$

and supposing further that the intensity of the receptive torque $-C_r$ drops until $-C_{r1}$ becomes equal as absolute value to the driving torque C_m, then the driven shaft **23** opposes to the driving shaft **1** a torque of an intensity $-C_m$ equal to its own $(-C_r)$. However, owing to the equation of the planet gears **20**, the opposing torque corresponding to $-C_r$ and effectively opposed to the driving torque C_m on its shaft **1**, is a torque:

$$-C_r''=-\frac{C_r}{\mu}=\frac{-C_m}{\mu}$$

The driving torque C_m thus is relieved of a fraction of the driven torque, whose intensity, equal to ΔC_m, is expressed as follows

$$\Delta C_m-C_r',-C_r''=C_m-\frac{C_m}{\mu}=C_m\left(1-\frac{1}{\mu}\right)$$

Consequently the term

$$\frac{1}{\mu}$$

in reversal of the multiplying factor of the driving torque C_m, is the demultiplying factor ζ of the angular speed of the driven shaft with $0.5<\zeta<1$, which results in $\Delta C_m=C_m(1-\zeta)$.

Since the driving torque is $C_m=\Sigma t.R_3$, the coupling traction is expressed as

$$T=\frac{\Delta C_m}{R_3}=\frac{C_m}{R_3}(1-\zeta)=\Sigma t_m(1-\zeta)=T_m(1-\zeta)$$

The second explanation furnished hereabove presents the mechanism of passage of the initial operation from the torque transforming gear as a multiplier of the driving torque (and, in consequence thereof, a demultiplying factor of the angular speed of the driven shaft **23**) to the operation in direct connection with identical torques and angular speeds on the driving and driven shafts.

It has thus been established that the equipment comprising the band **60**, the reaction drum **40** fixed to the sun gear **30**, the spring **70** and the supporting drum **50**, fulfills the purpose of utilizing the difference of intensity which may arise between the driving torque and the driven torque, when this latter becomes equal to the former, for restoring the equilibrium of the intensities of these torques by the application of a complementing or "compensating" supercharge ΔC_m to the driving torque C_m. This supercharge is the clutching traction obtained simply by suppressing automatically therein the action of the multiplying work of the planet gear **20**.

The automatically operative equipment described above constitutes the "compensating" system of the transformer device according to the invention.

The mechanisms of passage from the condition of the torque multiplier to that of a coupler in direct connection which was defined hereabove is obviously reversible. It was in fact described in the beginning of the first explanation. If from the moment when operation in direct connection sets in, the driven torque rises, the driving shaft is braked, the reaction on the sun gear rises also and movement of the equipment slows down until complete standstill is reached, while the compensating spring **70**, which was compressed from the beginning of the brake action by the main force of the equipment to which it is secured, releases the ends of the band **60**, allowing them to approach each other, whereby the compensating system of the driving drum **4** is declutched and operation by multiplication of torques rendered possible again, so that then the working conditions described in the first explanation are reestablished.

In the foregoing exposition of the gear unit forming the operation of the auto-torque transformer it was assumed implicitly that the driving element operated with a constant opening of the intake port of the motor.

In this case the initial tension of the compensating spring 70 (Figs. 1, 2 and 3) can easily be determined by the condition that the rate of relief of the operation as multiplier be slightly higher than that corresponding to the highest value of the driving torque. This satisfactory condition allows avoiding the risk, for the driving element, of being carried along at the highest speed of the zone of stable operation by the braking of the driven torque.

In reality, however, there exists an infinite number of openings for the fuel intake of a motor between the end positions of opening and closure. To this infinite number corresponds an infinite number of curves of torque producing the torque diagram of the driving element. Consequently there corresponds to each of these curves a point analogous to the one above defined which lies near a rate slightly superior to that of the maximum value, from which on the release of the operation of the auto-torque transforming gear must be effected. This is the condition which must be fulfilled in order that the auto-torque transforming gear be enabled to constitute an integral solution of the problem here in view. This condition requires that there be as many tensions t_0 as there are curves. The condition is fulfilled if there is imposed to the compensating spring 70 a rule of tension depending from the rate of operation, t_0 becoming a function of the variable n.

This condition is sufficient, for the other values here in view are functions of n.

The material performance permitting to obtain this result consists in the utilization of the action of the centrifugal force acting on a mass which communicates to the compensating spring a tension which this spring transmits to the clutch band in following the function imposed to it, viz.

$$t=F_c.\lambda \text{ with } F_c=m\omega^2R$$
$$t=m\omega^2R^\lambda$$

an expression in which F_c is the centrifugal force acting on the mass m when it rotates at a speed ω around the general axis of the device situated at a distance R from its center of gravity, while λ is a function of ω which permits of transforming the centrifugal parabola into an appropriate curve within the range of speeds utilized.

In Figs. 4 to 7, is illustrated an application of the principle here set out. The compensating spring 70 is mounted on a cylindrical support 71, on which it is free to slide. One end of the support is engaged in a bore 172 provided inside a joint 72, in which the support 71 is free to slide. The joint 72 is fitted in a notch 173 formed in the disc 32 which renders the reaction pinion 30 integral with its drum 40. The joint 72 is free to turn in its notch, in which it is held transversely by means of flanges 73 (Fig. 4). The support 71 is formed at its free end with a fork 74 carrying a roller 75 which is free to turn about an axle 76.

The spring is engaged between the joint 72 and the fork 76, being held between them and tending to force them asunder. The roller 75 is applied against a cam face 77 on the drum 50, the profile of this cam being designated as λ. It is engaged in an opening of the disc 32 and thereby held within its plane, as are also all the other members here described, and able to oscillate within this opening.

Fig. 5 shows this device in its condition of rest. The repulsive force which maintains, by the distance between the drums 40 and 50, the tension acting on the brake band 60, is the initial tension t_1 of the compensating spring 70. In order to facilitate the adjustment of the transformer, the connection 51 of the band 60 with the drum 50 is provided with means (not shown) for regulating the tension, which is accessible from without the casing 10. This regulating means might also be mounted on the connection 41 of drum 40.

The mobile equipment comprising the support 71 and its fork 74, the spring 70, the roller 75 and its axle 76, its washer and cotter (not shown) has a mass m calculated and situated at its center of gravity g, spaced by the radius R from the general axis of rotation. In the case where the driving drum of the driven shafts are in direct connection, under the influence of the speed of rotation this equipment is thrown outwardly by centrifugal force as shown in Figure 6, and turns about the axis of the joint 72 which forms the center of articulation, while the roller 75 climbs on the face of the cam 77, compressing the spring 70 in order that the tension t_0 at each one of its positions assume the intensity imposed by the corresponding speed of rotation.

Fig. 6 illustrates this device in the extreme position corresponding to the highest rotation speed, the spring 70 being fully compressed.

The following points are to be noted with respect to the structural embodiment here described.

The exterior brake 80 on the drum 50 is connected to the casing 10 by its ends 90 and 100 (Figs. 2 and 3). The end 90 may be connected to the casing by elastic means such as the spring 92 (Fig. 2), the end 100 by the spring 102. The end 90 is fixed to a small piston 91 with a spring 92 exerting on the piston a constant tension equal and opposed to the tension of the maximum torque before it is transmitted by the torque transforming gear. This tension is impressed on the piston by a nut 93. The other end (100) is fixed to a piston 101 which is acted upon by the very low tension of a small spring 102 which merely serves the purpose of avoiding loosening of the band 80. The piston 101 does not contact the nut 103 on the cylinder 104.

If, with the several members arranged as described, a supercharge or extra heavy load is imposed on the brake band 80, for instance in the sense of a declutching by the instantaneous application of the maximum torque, the negative traction of the brake 80 which is carried along by the drum 50, imposes on the spring 92 a force which permits the brake and its support 50 a negative rotation which induces the opposite end 100 of the band to apply its piston 101 against the nut 103 of the cylinder 104, whereby the band is unwound from the cylindrical surface of the drum 50 to the extent of allowing it to dampen by a negative rotation without outward effect the impulse received, the period of time consumed thereby being equal to the oscillation period of the spring 92.

As illustrated in Figs. 8 and 9 provision is made for controlling the supercompression of the spring 92 or 102 by actuation of a pedal and for utilizing this device in the place of the classical declutching device, to which it is superior insofar as it does not force mechanical pieces in motion to weight on each other.

Figs. 9 and 10 illustrate two settings permitting to let loose brake band 80 by operating upon one of its ends. The setting by Fig. 9 is in relation with the instance of a free end 81 of brake band 80. A pedal 120 actuating a groove 121 pushes a pin 122 which distends the brake band. A spring 123 draws the pedal back in a neutral position.

Setting refered to in Fig. 10 concerns the exertion of a thrust upon the ties of brake band 80. Pedal 120 articulated upon a support 125 affixed to the stationary casing controls a send back device 124 which pushes piston 101 by crushing the spring 102, which retains the extremity 100 of brake band 80 through the medium of a running rod 126 and penetrating into cylinder 104, container of piston 101.

Same setting may be employed for the opposite tie 90, but spring 92 is quite stiffer than spring 102.

As mentioned above, the piston 101 does not contact the threaded cylinder cap 103. The gap between them communicates with the space at the bottom of the casing by way of a conduit 105 situated in the plane of rotation of the disc 32 of drum 40 and opening in a negative sense.

While the gear unit constituting the torque transformer operates in direct connection with the driving and driven shafts, the drums 40 and 50 rotate at the general speed of the system. The brake 80 then does not play any role and it is advisable to separate it from the cylindrical surface of drum 50. The oil in the casing is projected under pressure by the circumferential acceleration imparted to it by the drums 40 and 50 toward the conduit 105 and acts on the small piston 101 which in rising in its cylinder 104 (Fig. 3) unwinds the brake band 80 and thereby lifts it from the drum surface.

When the torque transforming gear unit goes through an ascending phase, the drum 50 is braked, the oil pressure which is a function of ω^2, drops rapidly and the brake band 80 has returned to its position to become operative before the drum 50 is arrested. The drum may be provided with vanes to build up the oil pressure required in the cylinder 104.

While the torque transforming gear unit comprises amongst its specific members a free-wheel device, it does not possess a free-wheel action itself.

Propulsion by the kinetic energy of the vehicle in which the transformer may be mounted, converts the planet-wheel carrier into a driving element which will act in two ways:

1. The motor is slowed down, starting from the operation in direct connection, i. e. at high speed. The compensator is then subjected to the locking action of the transformed centrifugal force $Fc+\lambda$, the locking continues and the vehicle is braked by the motor in the unitary sense.

2. The motor is slowed down, starting from the multiplying of the torque, and the braking effect is then produced in corresponding multiplication.

The application of the principle of the direct torque transforming gear unit according to this application also comprises the provision of a torque de-multiplying and speed multiplying element which can be mounted in series with other torque transforming gears. In that case the driving torque is led into the element by the planet wheel carrier, the reaction sun gear is fixed and the internal gear ring receives and conveys to the outside the transformed torque and speed.

The device here described presents many different possibilities of use of its different elements. For instance—

1. The coupling drum which in the example described is constituted by the driving drum, may be a drum integral with the planet-gear carrier. This means one more member, however the angular speed is weaker and the start more progressive.

2. The planet-gear carrier may comprise the clutch member and its compensator and can engage either the driving or the reacting drum.

3. The trains of gears may be formed with helicoidal teeth and the variation of the axial thrust of the planet gears or of the reactor may be utilized together with an appropriate compensator for causing the coupling of the element selected.

I wish it to be understood that I do not desire to be limited to the details of construction shown and described, for obvious modifications will occur to a person skilled in the art.

I claim:

1. In a mechanical automatic torque transformer of the kind described in combination, a stationary casing, a driving shaft and a driven shaft arranged coaxially in, and extending outwardly through the walls of said casing, a first drum fixed to said driving shaft, an internal gear ring fixed to said first drum, a gear carrier disc fixed to said driven shaft, planet gears rotatably mounted on said gear carrier disc and meshing with said internal gear ring, a sun gear freely rotatable only in the driving sense, mounted on said driven shaft in mesh with said planet gears, a second drum integral with said sun gear surrounding said first drum, and a third drum mounted on said second drum for angular displacement relative to same, elastic means arranged between said second and third drums for controlling their relative angular displacement and rotation in common, an automatic braking means attached to said stationary casing for braking said third drum in a sense counter to the revolution of said driving shaft, and an automatic clutching means capable of coupling said internal gear ring with said second drum, said clutching being controlled by a relative angular displacement of said second and third drums.

2. The device of claim 1 in which the automatic braking is constituted by a brake band affixed by one of its ends to the stationary casing and spirally wound about said third drum, the sense of such winding counter-acting the driving sense.

3. The device of claim 2 in which the tie of one of the ends of said brake band on the stationary casing comprises a deadening spring.

4. The device of claim 2 in which the second end of same said brake band may be actuated outwardly, from the stationary casing, in order to loosen said brake band and to procure declutching action between the motor and the automatic torque transformer.

5. The device of claim 1 wherein said automatic clutch is constituted by a clutch band spirally wound about said first drum, one end portion of said clutch band being attached to said second drum and the other end being attached to said third drum.

6. The device of claim 1 wherein said elastic means between said second and third drums is a spring whose reaction is tangentially exerted on said second and third drums and is transmitted

by these drums to the far ends of the clutch band to brake the first driving drum.

7. The device of claim 1 in which the tension of the spring forming the elactic means inserted between second and third drums is a function of rotational speed and comprises a cylindrical joint formed with a bore and capable of turning in a cylindrical notch formed in the first drum, a piston movable in said bore, the said spring tending to push said piston away from said joint and a cam face on said second drum being in contact with said piston and tending to increase said spring action.

JEAN F. G. M. L. CHARPENTIER.

References Cited in the file of this patent

UNITED STATES PATENTS

Number	Name	Date
917,729	Henriod	Apr. 6, 1909
1,170,980	Levedahl	Feb. 8, 1916
1,780,293	Christie	Nov. 4, 1930
2,120,832	Cotterman	June 14, 1938
2,136,971	Fleischel	Nov. 17, 1938
2,244,133	Taylor	June 3, 1941
2,248,133	Snow	July 8, 1941