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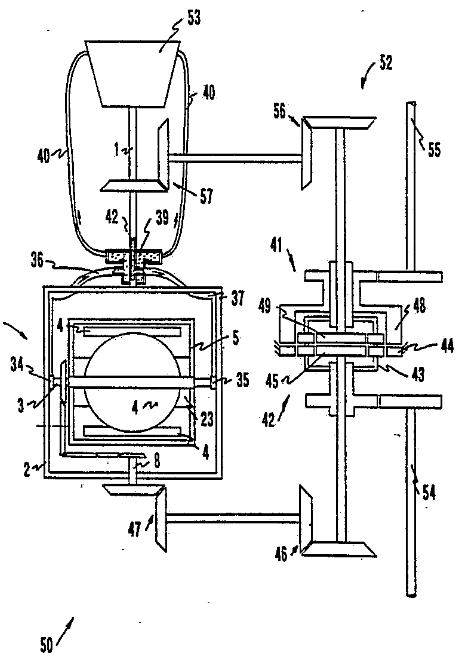
(54) Title: CONTINUOUSLY VARIABLE TRANSMISSION

(57) Abstract

A continuously variable transmission comprising an input shaft (1), an output shaft (8), a rotatably mounted main frame (2), a sub-frame (5) rotatably mounted within the main frame (2) substantially perpendicularly to the rotational axis of the main frame (2), and a mass distribution (4) mounted within the sub-frame (5) movable about an axis lying in a plane substantially perpendicular to the rotational axis of the sub-frame (5), wherein the main frame (2) is driven by the input shaft (1) and the output shaft (8) is connected, by way of a right angled gear train, to the sub-frame (5) rotational axis. Coupling between the input shaft (1) and output shaft (8) is achieved using the variable coupling torque generated by an inertial reaction from the oscillating mass distribution (4). The coupling torque fluctuates are utilised to generate coupling energy which is withdrawn from the transmission in a manner allowing the torque fluctuations to be synchronized with the sub-frame (5) rotation to achieve a net coupling advantage.



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CONTINUOUSLY VARIABLE TRANSMISSION

TECHNICAL FIELD

The present invention relates to a continuously variable transmission and, more particularly, to a continuously variable transmission in which inertial effects derived from an oscillating mass distribution are used to generate a coupling torque and enable a net transfer of power from the input of the transmission to the transmission output.

BACKGROUND ART

Conventional transmissions, such as those presently in wide spread use in applications such as motor vehicles, typically operate at any one of several discrete speed/torque input:output ratios which have been specifically designated during the transmission design. While the adoption of discrete speed/torque input:output ratios generally facilitates design, such transmissions have significant drawbacks in situations where varying output conditions are experienced.

In this connection, certain types of power source, such as for example internal combustion engines, only function at maximum efficiency within a narrow range of operating speeds. If a transmission of conventional configuration is coupled to such a power source substantial inefficiencies will result under output conditions of the sort which exist during start, stop, acceleration and deceleration, and also when the drive system is called upon to operate at a constant speed which does not correspond to a speed falling within the efficient operating range of the power source.

Moreover, resulting from the fixed nature of the input:output ratio of the transmission is a requirement that an additional mechanism, generally comprising a dry plate clutch or the like, for disengaging the transmission from the power source be provided. Inclusion of such a mechanism invariably leads to further significant inefficiencies resulting from slip etc.

A variety of transmissions capable of virtually continuous variation of input:output ratio are known. In the main, these utilise hydraulic systems for torque transfer. Again, however, drive systems which incorporate such transmissions tend to suffer inefficiencies due to slip under the transient conditions of start, stop, and acceleration and deceleration, and further, are prone to high internal fluid frictional losses and slow response capability under rapidly changing output demands.

The applicant is aware of a number of prior proposals which endeavour to utilise gyroscopic rotors to achieve a continuously variable transmission. The device disclosed in US Patent Specification 4,169,391 is one such transmission. The basic elements of the transmission of this US Patent Specification comprise a rotatable main frame, two identical and separate sub-frame members which are rotatably mounted within the main frame, four identical gyroscopic type rotors which are coaxially spin mounted in pairs within the sub-frames, and two stators, one for each sub-frame, with each stator circumferentially positioned around main frame adjacent the sub-frame member.

Operationally, the moment of inertia of the rotors is varied by automatically varying the radii of gyration in predetermined sequence in concert with the rotation of their sub-frame members. Rotor spin is effected and controlled by individual hydraulic turbine drives. It is alleged that when the rotors are spinning and a precession is applied from an input power source such as to impart contra-rotation to the subframes the paired rotors will generate an effective output torque to the output shaft via the main frame.

It has been noted that use of hydraulic turbine drives, together with varying the radius of gyration of each rotor in the manner disclosed is not practical due to the small response time available, high energy losses due to fluid friction and the difficulty in accurately controlling the system. This situation is greatly aggrevated by the oscillating inertia torque acting on the rotors. A further shortcoming of the invention disclosed is that the output power and the input power are matched by a hydraulic feedback control. Due to feedback delays energy losses result. Furthermore, power transmission cannot take place in the reverse direction because of the worm gear drive train and, therefore, regenerative braking is not possible.

U.S. Patent Specification 3,851,545 also discloses a continuously variable transmission. This transmission includes a main, fixed, housing, a rotatable frame in the housing carried by coaxial driving and driven shafts extending into the housing, gyroscopic cage shaft means extending perpendicular to the axis of the coaxially driving and driven shafts, gyro cage means rotatably mounted on the gyro cage shaft means, bevel gears mounted on the driving and driven shafts and a bevel gear therebetween and meshing therewith mounted on the gyro cage shaft means, a pair of gyro support shafts mounted on the gyro cage means, gyroscopic rotors having spin axles mounted on each of the gyro support shafts, the spin axles being substantially perpendicular to the gyro support shafts and means connecting the gyro supports shafts together and relative to the gyro cage shaft means to maintain the spin axles in parallel relation.

The differential arrangement incorporated in the above detailed transmission is very similar to conventional differential drive systems. There is no sub-frame/main frame combination and the frame rotation is dependant on both input and the output rotations. Accordingly, to achieve rotor spin some external spin generating means is required to drive and maintain the rotors in spinning orientation.

The need for an external power input to provide and maintain the rotors in spinning orientation adds significant complexity to the invention disclosed and the lag involved in adjusting the spin speed of the rotors to achieve a change in speed or torque transfer potentially cause problems, especially under rapidly fluctuating input or output conditions.

Thus, it will be appreciated that there is still a need for a fast response continuously variable transmission capable of operating efficiently through a wide range of speed/torque ratios and capable of net transfer of power in either direction through the transmission.

Accordingly, it is an object of the present invention to provide a continuously variable transmission in which the input power and the output power are substantially the same under all operating conditions, and/or in which for any given input speed the output speed may vary over a wide range of values, and/or in which for any given output speed the input speed may vary over a wide range of values, and/or which can provide net transfer of power in either direction at a continuously variable speed ratio, and/or to at least provide the public with a useful choice.

Further objects of the invention will become apparent from the following description.

SUMMARY OF THE INVENTION

The present invention essentially involves a differential type drive arrangement, in that the input shaft and the output shaft can spin at a continuously variable speed ratio. Coupling between the input shaft and output shaft, and control of the relative shaft spin speeds, is achieved by way of a variable coupling torque generated by an inertial reaction from an oscillating mass distribution.

In order to create a net coupling torque transfer of coupling energy is necessary between the transmission and some other device, dissipated to the surroundings or, preferably, is recirculated within the transmission.

In an oscillating mass distribution the inertial reaction generating the coupling torque fluctuates. By transferring coupling energy into or withdrawing coupling energy from the transmission in an appropriate manner the torque fluctuations can be sychronised to achieve a net coupling advantage.

To satisfy one or more of the aforestated objects the present invention provides, in its most general form, a continuously variable transmission comprising an input shaft, an output shaft and a rotatably mounted main frame which have aligned rotational axes, a sub-frame rotatably mounted within the main frame, having an axis of rotation substantially perpendicular to the rotational axis of the main frame, and a mass distribution mounted within the sub-frame movable about an axis lying in a plane substantially perpendicular to the rotational axis of the sub-frame, characterised in that the main frame is fixed to the input shaft and is rotatable therewith, and the output shaft is connected, by way of a right angled gear train, to the sub-frame at the sub-frame rotational axis, such that when the input shaft and the output shaft rotate at different rotational velocities the sub-frame is caused to rotate about its rotational axis, the transmission further comprising a fluctuation means independant of the output shaft to vary the angular momentum of the mass distribution to be a maximum and/or minimum once every predetermined number of rotations of the subframe so that any rotation of the main frame results in the generation of a net torque at the output shaft.

Preferably, the fluctuation means is driven or activated by the rotation of the main frame and the sub-frame, and oscillates the mass distribution about its rotational axis so that the angular velocity of the mass distribution is a maximum and/or minimum once every rotation of the sub-frame.

Alternatively, the fluctuation means is driven or activated by the rotation of the main frame and the sub-frame, and oscillates the mass distribution abouts its rotational axis so that the moment of inertia of the mass distribution is a maximum and/or minimum once every rotation of the sub-frame.

Desirably the mass distribution is a gyroscopic rotor.

Conveniently the fluctuation means includes a double acting positive displacement pumps driven by the oscillating inertial torque acting on the rotor.

For greatest net torque transfer the fluctuation means must maximise the angular momentum of the mass distribution when the axis of the mass distribution is substantially perpendicular to the main frame and minimise the angular momentum of the mass distribution when the axis of the mass distribution has turned through 180 degrees.

A variant to the above noted general form of the invention may be provided so that the transmission comprises an input shaft, an output shaft and a rotatably mounted main frame which have aligned rotational axes, and a sub-frame, rotatably mounted within the main frame, having an axis of rotation substantially perpendicular to the rotational axis of the main frame, characterised in that a mass is reciprocally mounted on the sub-frame to move substantially radially towards and away from or back and forth parallel to the rotational axis of the sub-frame, further, the main frame is fixed to the input shaft and is rotable therewith, and the output shaft is connected, by way of a right angled gear train, to the sub-frame at the sub-frame rotational axis, such that when the input shaft and the output shaft rotate at different rotational velocities the sub-frame is caused to rotate about its rotational axis, the transmission also including a fluctuation means independent of the output shaft to vary the radial distance of the mass from the rotational axis of the sub-frame to be a maximum and/or minimum sychronous with revolution of the subframe so that rotation of the main frame generates a net torque at the output shaft.

In this form of the invention when movement of the mass is radial greatest net torque transfer occurs if the fluctuation means maximises the radial distance of the mass when the angle between the plane containing the main-frame and the plane containing the mass and the axis of the sub-frame is substantially 45 degrees and minimises the radial distance of the mass when the said sub-frame plane has turned through 90 degrees about its axis. Maximum net torque transfer occurs when the maximum and the minimum radial distances are achieved as the plane containing the sub-frame makes an angle of 45 degrees with the plane containing the main frame.

Preferably the fluctuation means and the mass consist of a double acting piston pump, the piston being the reciprocating mass or being movable under the influence of the oscillating inertial force on the mass due to the sub-frame rotation and/or main frame rotation.

In both the general form and variant of the general form of the invention, as noted above, the fluctuation means can comprise a device on the input shaft, adjacent its attachment to the main frame, to either take or supply the energy required for or received from fluctuation of the momentum of the rotationally oscillating mass distribution or reciprocating mass,, a device mounted on the sub-frame to oscillate the angular momentum of the mass distribution or the linear momentum of the reciprocating mass and a transfer means for transferring energy between the said devices.

Preferably the transfer includes a hydraulic system.

Desirably the device on the input shaft removes energy generated by the fluctuating momentum and transfers it to the input shaft.

Conveniently, the device on the input shaft is a turbine. Alernatively, the device is a pelton wheel.

The net torque transferred by a single rotationally oscillating mass distribution or reciprocating mass fluctuates cyclicly. By using a minimum of four masses or mass distributions disposed around the sub-frame at 90 degree phase intervals, so that each is dynamically identical with any other of the four when occupying any given position with respect to the plane of the main frame, the fluctuations can be significantly eliminated. If more than four are used then for dynamic balance the number should increase in increments of four, all equispaced.

Provided the fluctuation means are independent of the output shaft rotational oscillation of the mass distribution or reciprocation of the may be driven by either the sub-frame and main frame rotation or by an independent drive. It may be noted that in the latter case, for example, where the mass distribution is a rotor and rotor speed fluctuation is used the fluctuation means and the driving means are one and the same.

Thus, it is apparent that power transfer from the input shaft to the output shaft involves the variation of angular momentum of a rotationally oscillatable mass distribution, such as a gyroscopic rotor, by varying either the spin speed or the moment of inertia or, in the case of a reciprocatable mass, by varying the linear momentum, which can be achieved by several methods. It is also apparent that the variation should be synchronous with the rotation of the sub-frame about its axis.

To summarise, practical methods to vary the momentum include use of oscillating inertial torque on a rotor or equivalent mass distribution rotationally mounted on the sub-frame; or, in the case of a reciprocatable mass, by use of fluctuating inertial force the oscillating torque or fluctuating inertial force being caused by the combination of the subframe rotation and the main frame rotation or main frame rotation alone. This oscillating torque or fluctuating inertial force is synchronous with the sub-frame rotation and therefore is suitable as an acutating means.

As noted above, to achieve a net coupling torque requires transfer to or from the transmission of a certain amount of coupling energy. It is the coupling energy in which the energy to vary the momentum in the desired manner and is synchronous with the rotation of the subframe is embodied.

While it is possible to achieve the net coupling torque by adding a coupling energy component to the transmission it is significantly more desirable to transfer coupling energy from the transmission especially in transmissions of the kind described above.

The reason for this is that the oscillating torque or inertial force fluctuations generate a fluctuation in the momentum vector of the mass distribution or mass, respectively, which must be timed to vary in a desired manner and in synchronous with the sub-frame rotation. This necessitates generating a phase shift in the momentum vector (angular for a rotationally oscillating mass distribution, linear for a reciprocating mass). Instead of overriding the momentum fluctuations generated by the oscillating torque/inertial force fluctuations by transferring energy into the transmission the momentum fluctuations can be utilised to do work, at the same time providing the means for their control in a desired manner.

As indicated above, transfer of the coupling energy may be effected using a medium such as electricity, hydraulic fluid or the like. For practical purposes if the coupling energy is to be transferred from the transmission hydraulic fluid is the preferred medium, in which case the torque oscillations/inertial force fluctuations can be controlled and utilised to pump hydraulic fluid.

While the coupling energy taken from the transmission may be used elsewhere, for maximum efficiency the energy should be returned or recirculated to the transmission. This may be achieved by incorporating a pelton wheel, turbine or like arrangement on the input shaft.

The recirculated energy is proportional to the oscillating torque/inertial force fluctuations and not to the power transmitted. Under steady conditions and 100% transmission efficiency the energy recirculation required between the main frame and the fluctuation means is equal to the product of the output torque and the slip, slip being the difference between the input speed and the output speed.

Therefore the transmission of the present invention may be designed so that the maximum rate of energy recirculation required is only a fraction of the maximum power transmitted. The methods to minimise the rate of energy recirculation may include the following:

1. Low main frame speed or slip at starting conditions.
2. Low output torque at starting conditions.

The present invention is such that in general for all operating conditions the net torque on the input shaft is proportional to the speed of the output shaft and the net torque on the output shaft is proportional to the speed of the input shaft, assuming no losses and energy recirculation.

The roles of the input and output shafts of the transmission are not interchangeable, although reversibility of power transfer is always possible. In this connection, transfer of power can be reversed by reversing the output rotation or automatically when the output speed exceeds an equivalent input speed. When the output shaft rotation is reversed the load, after giving up its energy gradually to the input shaft (or to an energy storing device), will come to rest or continue to rotate in the opposite direction, if allowed, by deriving energy from the input shaft (normal transfer).

The coupling torque is created as a function of the input/output speeds and the output torque-speed characteristics can be predetermined irrespective of the other output conditions. The hydraulic system is automatically adjusted by the actuating inertial reaction as the output speed varies and this adjustment does not necessarily require any feed back control.

Where the mass distribution comprises a freely rotatable gyroscopic rotor the spin is generated by torsional vibration and the forcing function, which is the actuating inertial reaction, is determined by the input/output speeds. The amplitude of vibration and hence the effective spin speed are thus predetermined as a function of the input/output speeds unless otherwise controlled by damping.

Coloumb damping to obtain control over the torsional vibration is effected, as noted above, by utilising the oscillating rotor to pump hydraulic fluid. Thus, the control of the hydraulic fluid velocity or pressure can be used to control the vibration of the rotor.

Feed-back control is neither required nor practical except at the inlet nozzle of the turbine, pelton wheel or like device used to recirculate the coupling energy.

It has been found that with appropriate design the efficiency of the transmission is high, even without the nozzle adjustment, over a limited operating range, and that step changes to the nozzle dimension are sufficient to maintain high efficiency over a wide operating range.

Control over the turbine nozzle has a further advantage, in that by closing the turbine nozzle transfer of power by the transmission can be decoupled.

The hydraulic system variables (pressure, flow rate and other dimensions) can be optimised independently without affecting the torque-speed characteristics of the transmission.

The present invention may also include a speed controller to enhance the operation of the transmission. This device enables the speed of rotation of the sub-frame (about the rotational axis of the subframe) to be controlled. The speed controller is based on an epicyclic differential drive and operates by cancelling or reducing the effects of change in the output speed (or input speed in reverse transmission mode) on the rotation speed of the subframe. Control of the sub-frame speed is useful, for example, in designing the fluctuation means.

BRIEF DESCRIPTION OF THE DRAWINGS

Further aspects of the invention will become apparent from the following description, given by way of example only, of possible embodiment of the invention, described with reference to the accompanying drawings, in which like elements are like numbered:

FIGURE 1 illustrates schematically the principles relating to gyroscopic torque due to single precession;

FIGURE 2 details, again schematically, a system which provides double precession through a differential drive arrangement;

FIGURE 3A shows an end view of a sub-frame in a system similar in principle to Figure 2, multiple rotors;

FIGURE 3B shows an end view of an alternative sub-frame configuration to Figure 3A;

FIGURE 4 illustrates in part the preferred embodiment of the invention, detailing in particular one of four rotor assemblies mounted in the sub-frame;

FIGURE 5 shows an end view of the fluid accumulator of the arrangement of figure 4 where it interfaces with the torque disc;

FIGURE 6 illustrates schematically an overview of the preferred embodiment, outling the frequency controller and the turbine arrangement for recirculation of the coupling energy;

FIGURE 7 shows schematically an alternative frequency controller arrangement;

FIGURE 8A illustrates graphically the non-stop motion boundaries within which vibration of the rotors of figure 4 must be confined for controlled operation if no viscons damping occurs;

FIGURE 8B illustrates similar boundary limitations to figure 8A, but with viscons damping;

FIGURE 9 illustrates a further embodiment of the invention, in which reciprocating masses generate the coupling torque; and,

FIGURES 10A and 10B illustrate a flow diagram of a spread sheet model used to calculate various variables for the embodiment of the invention shown in figure 6.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

Referring firstly to Figure 1, the general principles related to net gyroscopic torque generation are described as follows. A rotor R spinning about the Z-axis has an angular momentum M. If this rotor R is rotated about the X-axis with angular velocity N then a gyroscopic (reaction) torque T about the Y-axis is generated. The torque T is given by the relationship:

$$T = M \times N$$

Rotor R1 represents rotor R once it has rotated through

180° about the X-axis. its angular momentum M1 when spinning about the axis Z is in the direction shown. If the rotor R1 rotates about the axis X with the same angular velocity N as rotor R then gyroscopic torque T1 is in the direction indicated and equal to M1 x N. The resultant torque for both rotors is thus:

$$T_R = N(M - M_1).$$

Accordingly, two rotors of constant moment of inertia rotating with constant speed about the X axis and having angular momentum as shown in Figure 1 produce gyroscopic reaction torques which cancel. Gyroscopic torques are at a maximum value in the position shown in Figure 1. If the axis about which rotors R and R1 spin was inclined towards the X or Y axes there would be a decrease in the gyroscopic torque generated in the direction Y, until, when the axis becomes co-axial with either axis X or axis Y, there would be no gyroscopic torque in the direction Y.

It can also be seen that if a single rotor were spinning about axis Z and rotating about axis X in the direction indicated by the arrow the gyroscopic torques in fact cancel through a complete cycle. Accordingly, no net gyroscopic torque would be generated over a complete cycle. Only an oscillating torque would be produced. If, however, the angular momentum of a rotor when in the position of rotor R is varied over a complete revolution about axis X net gyroscopic torque will be generated about axis Y. This will introduce an acceleration/decleration torque and, as will be seen later, for transmission of energy from the input to the output using such net gyroscopic torque transfer of coupling energy between the fluctuation means and another device is necessary.

Referring to Figure 2, a simplified embodiment of the coupling device of the invention will now be described. A rotor 4 is rotatably mounted about an axis 9 on a sub-frame 5. The sub-frame 5 is rotatably mounted on a shaft 3. A gear wheel 6 is rigidly attached to the sub-frame 5 and shafts 1 and 3 are rigidly attached to the main frame 2. Shaft 8 is free to rotate with respect to the main-frame 2 and a gear wheel 7 is rigidly fitted to the shaft 8. Gear wheels 6 and 7 are connected through to gear train G1-G3 rotatably mounted on the main frame 2. Rotation of the shafts 1 and/or 8 thus causes rotation of the sub-frame 5. The output shaft 1 rotates at angular velocity N_1. Thus the system provides a double precession (i.e. two axes of rotation other than spin) to the rotor 4 due to the rotations N_1 and N_6.

Let the angular velocities N_1, N_6 and N_8 be constant. i, j and k are mutually perpendicular vectors with respect to the sub-frame 5, so that j is always coincident with the axis 9 of the rotor 4 and k is always coincident with the axis of the shaft 3 or the sub-frame 5.

From vector analysis with a rotating frame of reference the reaction torque vector on the rotor 4 due to inertia of the rotor 4 is given by the relationship:

$$T_r = I \times [i(N_4N_6) + j(\underline{-d}(N4)+N_1N_6\text{Cos }\mu) + k((-N_1N_4 \text{ Cos }\mu$$
$$(\overline{dt}$$

$$+ (N_1)^2 \frac{\text{Sin }\mu \text{ Cos }\mu}{2})]$$

$$- \frac{dI}{dt} [i \frac{(N_1 \text{ Cos }\mu\ )}{2} + j\ (N_4 - N_1 \text{ Sin }\mu\ ) + k\ (\frac{N6}{2})]$$

where,

I is the polar moment of inertia of the rotor.
N_1, N_4 and N_6 are the angular velocities shown.
is the angle turned by the sub-frame 5 from the position shown in the direction N_6.
$\frac{d}{dt}$ is the time derivative.

Accordingly the following are apparent from consideration of the above equation and accompanying discussion:

a) by maximising and/or minimising the angular momentum of the rotor 4 once every complete revolution of the sub-frame 5 appropriate net reaction torque (gyroscopic) will be generated at the output, shaft 8, due to the rotation of the main frame 2, provided the means effecting the fluctuation is not coupled to rotation of the sub-frame 5, and the net transfer of coupling energy will occur between the transmission and the surroundings, another device, or recirculated back to the transmission input, shaft 1. Net gyroscopic torque on the output shaft 8 is proportional to the energy recirculation (assuming no dissipation of energy by the medium) and the energy recirculation is not directly related to the power transmitted.

b) the net torque generated at shafts 1 or 8 is proportional to the speed of the other shaft when coupling energy recirculation occurs.

c) Fluctuating reaction torques generated with net zero value can be balanced at the shafts 1 and 8 with respect to time by employing a minimum four rotors disposed around the sub-frame 5 at 90 degree intervals appropriately.

d) The energy recirculation, being dependent on the output torque and slip, will be reduced as the output torque is reduced and the output speed increased. This is the reverse of conventional transmissions, in which the working medium always transmit output power.

Figures 3A & 3B show two of many possible configurations which are similar to the configuration in Figure 2 whereby each rotor 4 is subjected to a double precession.

The reaction torque about the axis 9 of each rotor 4, if stationary, due to the rotations of the sub-frame 5 and the main frame 2, is as follows:

$j\ IN_1N_6\ Cos\mu$

and, as noted, is synchronous with rotation of the sub-frame 5.

In the most preferred embodiment, which is illustrated in part in figure 4, the oscillatory nature of the reaction torque is used, in conjunction with a spring arrangement, to force the rotor 4 to vibrate. Net torque on the sub-frame 5, and hence on the shafts 1 and 8, is achieved by using suitable damping techniques to phase shift the vibrations. Thus the damping energy provides the coupling energy in this case.

As has been noted above, coupling energy is required to achieve a net power output. Preferably the transfer of energy to achieve coupling is always from the sub-frame 5, since it has been considered that the coupling energy is most easily provided using the oscillatory reaction torque to pump hydraulic fluid from the sub-frame 5. When a double acting positive displacement pump is used coulomb type damping is pro-

-vided to the rotor 4. Referring also to figures 8A & 8B, with coulomb type damping the operating mode of the rotor assembly should be kept within the non-stop motion region, and this can be achieved by use of a frequency controller, such as is illustrated in figures 6 and 7, and/or by controlling the fluid pressure differential between the two sides of the pump.

A small amount of viscous damping is inevitable and preferable, especially to optomise operation at or near resonance, however, too great of amount of viscous damping will result in significant energy losses.

Referring generally to figure 6 and specifically to figures 4 and 5, the preferred transmission system, as generally indicated at 50 comprises a transmission 51, a speed controller 52 and a coupling energy recirculation device 53.

The transmission 51 is of the type outlined above, having four gyroscopic rotor assemblies disposed around the sub-frame axis 3 at 90 degree intervals. Figure 4 illustrates in schematic detail the various components of each rotor assembly.

Each rotor assembly is mounted in the sub-frame 5 and comprises a rotor 4 mounted on a shaft 9. The rotor 4 and shaft 9 can rotate, or oscillate. Any rotation of the shaft 9 is transmitted, through a reducing gear train, to a further shaft 15. The reducing gear train comprises a first gear 10 fixed on the shaft 9, gears 11 and 12 fixed to an intermediate shaft 14, and a gear 13 fixed on the shaft 15, and may provide a reduction of up to 15:1, ie for 10 revolutions of the rotor 4 the shaft 15 rotates through approximately 240 degrees.

The end of the shaft 15 is fixed to, and oscillates the cylinder housing 16 of a unique double acting positive displacement type pump. The cylinder chamber 17 is formed as an annulus, with the annulus being semi-circular in cross-section. The base of the cylinder 17 comprises a circular flat plate 18 and the cylinder housing 16 oscillates relative to the plate 18. A seal 26 around the rim of the connection between the housing 16 and plate 18 maintains pressure within the pump and provides a fluid tight joint.

Upstanding from the plate 18 is a divider 19 shaped to substantially correspond with the cross-sectional profile of the cylinder 17. A similar divider 20 extends downwardly to the plate 18 from the housing 16. In static orientation the dividers 19 and 20 are spaced approximately 180 degrees apart around the annular cylinder 17, however, under operating conditions the divider 20 oscillates back and forth around the cylinder 17 carried by the housing 16, which in turn is driven by the rotor 4.

In the plate 18, on either side of the divider 19 is an inlet port 21 and and outlet port 22, each of which incorporates a one-way flap valve to ensure that fluid may only pass through in the desired direction.

As the divider 20 sweeps around the annular cylinder 17 fluid ahead of the divider 20 is squeezed between it and the static divider 19, and is forced, under pressure, to exit the cylinder 17 via the outlet port 22. Similarly, fluid is sucked into the cylinder 17 via the inlet port 21 behind the divider 20.

Fluid is supplied from and pumped into a fluid accumulator 23. The fluid accumulator 23, which is in the form of a drum surrounding the shaft 3 on which the sub-frame 5 is carried, feeds fluid to and receives fluid from all four rotor assemblies. Internally the accumulator 23 is divided into two chambers, a low pressure chamber 24 which supplies fluid to the inlet ports 21 of each of the rotor assemb-ies, and a high pressure chamber 25 which receives the fluid pumped out of the outlet ports 22.

The shaft 3 incorporates bores 27,28 extending from either end respectively for a distance along its length. A port 29 extends from the exterior of the shaft 3 to the bore 27 allowing fluid from the high pressure chamber 25 of the fluid accumulator 23 to pass therethrough and out along the bore 27.

A port 30 is also provided to the bore 28 allowing communication of fluid between the bore 28 and the low pressure chamber 24 of the accumulator 23.

A lug 31 is attached to the cylinder housing 16 and a corresponding lug 32 is attached to the sub-frame 5. A pair of semi-circular coil springs 33 extend between the lugs 31,32. A semi-circular guide rod (not shown) may be fitted within each spring 33, attached at one end to either of the two lugs 31,32, while moving freely with respect to the other lug 32,31. The springs 33 assist in providing reversal of the spin direction of the rotor 4.

In use rotation of the shaft 1 causes the mainframe 2 to rotate. If the output shaft 8 is not rotating at the same speed as the input shaft 1 the subframe 5 will rotate due to the gear train G1-G3 connecting the output shaft 8 to the subframe 5.

Rotation of the main and sub-frames 2 and 5 respectively, causes the rotor 4 to oscillate due to the reaction torque generated. This oscillation is geared down through the gear train 10,11, 12 and 13 to rotate cylinder housing 16. This oscillation is used to pump fluid from the low pressure chamber 24 of the accumulator 28 into the high pressure chamber 25.

Oscillation of the rotor 4 and housing 16 is in sychronous with the rotation of the sub-frame 5, however, the amplitude of the oscillations, ie the number of rotations of the rotor 4 between direction reversals is proportional to the rates of rotation of the main frame 2 and sub-frame 5.

Referring now more particularly to figure 6, the ends of the shaft 3 are rotatably mounted to the main frame 2. A rotatable fluid joint 34,35 is incorporated in each mounting and a fluid line 36,37 respectively extends from each joint 34,35 around the main frame 2 to the input shaft 1. The bore 28 communicates, by way of the joint 34, with the line 36 and the bore 27 communicates, by way of the joint 35, with the line 37.

Fluid under pressure may pass from the high pressure chamber 25 of the fluid accumulator 23, through the bore 27 and along the fluid line 37. From the line 37 the high pressure fluid passes through a rotatable fluid coupling 39 about the shaft 1, and through several take off tubes 40 to the inlet of the energy recirculation device 53 mounted on the inlet shaft 1. After giving up its energy to the energy recirculation device 53 the fluid drains down into a bore 42 in the shaft 1 from where it moves on to the fluid line 36, back through the bore 28 into the low pressure chamber 24 of the fluid accumulator 23.

Mounted in parallel to the transmission 51 is a speed controller 52.

The speed controller 52 enables the forcing function (fluctuating torque together with the effects of the coil springs 33 and coulomb damping provided by the pump) to be controlled in a predetermined manner by controlling the speed of the sub-frame 5 dependently or independently of the speed of the output. The basic principle of operation of the speed controller 52 is to vary the speed of the shaft 1 and the speed of the shaft 3 by predeterined values as the output shaft 8 changes speed.

The speed controller 52 consists of an epicyclic gear arrangement where two epicyclic gear units 41,42 have a common spider arm 43 geared to the transmission system output shaft 54. The annulus 44 for the output shaft side of the speed controller 52 is fixed. The corresponding sun gear 45 is coaxial with the transmission 51 and is geared to the output shaft 8 via right angled gear trains 46,47.

The annulus 48 for the input shaft side is free and is geared to the transmission system input 55. The corresponding sun gear 49 is also coaxial with the transmission 51 and is geared to the shaft 1, via right angled gear trains 56,57.

Analysis of the above described arrangement shows that the speed difference between the sun gears 45 and 49 is a constant and is proportional to the speed of the input 55. If the gear ratios are not equal the speed difference will vary with the output speed proportional to the product of the output speed 54 and the difference between the gear ratios. Since the speed of the subframe 5 will be proportional to the speed difference between the sun gears 45,49 the former is controlled in a predetermined manner as the speed of the output 54 varies.

Since the frequency of the forcing function is exactly equal to or proportional to the speed of the sub-frame 5 the speed controller 52 can significantly influence the output torque-speed characteristics.

Many variations to the arrangement described are possible. Figure 7 illustrates one such variation. The basic principle involved is to cancel or reduce the effect of the change in output speed on the sub-frame speed by equalising appropriately the effects on the main frame speed and on the speed of the coaxial shaft rotatable with respect to the main frame 2.

The following is typical performance data for the above described embodiment of the invention not optimised for any particular application:

Output speeds (rad/s)	30	60	90	120	150	180
Gyroscopic Rotor Radius (m)	0.15	0.15	0.15	0.15	0.15	0.15
Rotor thickness (mm)	10	10	10	10	10	10
Input speed (rad/s)	300	300	300	300	300	300
Fluid pressure (bar)	9.4	11.1	13.1	15.0	17.7	22.9
Nozzle diameter (mm)	12.5	13.5	14.8	16.7	17.5	14.3
Amplitude of rotor Oscillation (rad)	11.0	14.9	20.7	30.0	38.6	28.6
Reynolds No.	3030	3955	5265	7306	9000	6350
Power transmitted (kw)	16	44	104	224	404	376
Power loss (kw)	5.2	9.2	17.2	36.18	62.4	33.6

The above data is derived from a computer generated spread sheet model. A flow diagram for the spread sheet model is provided in figures 10A and 10B.

The flow diagram of 10A and 10B incorporate the following codings for the variables.

ELECTABLE, OR KNOWN INPUT VARIABLES

VARIABLE	CODE
Gear Radio	GR
Input Speed	IS
Output Speed (design)	OS
Rotor Polar Moment of Inertia	RI
Coloumb Damping Ratio	CD
Radius of Centre of Pressure	RP
Frequency Ratio (design)	FR
Viscous Damping Ratio	VD
Hydraulic Line Dimensions	LD
Fluid Properties	FP
Piston Dimensions	PD

CALCULATED VARIABLES

Variables	Code
Main Frame Speed	MS(n)
Sub-frame Speed	SS(n)
Turbine Speed	TS(n)
Forcing Function	FF(n)
Pressure Torque	PT(n)
Fluid Pressure	HP(n)
Natural Frequency of Rotor	NR(n)
Spring Constant	SC(n)
Vibration Amplitude	VA(n)
Fluid Flow Rate	FR(n)
Viscous Losses	VL(n)
Line Velocity	LV(n)
Reynolds Number	RN(n)
Flow Losses	FL(n)
Discharge Velocity	DV(n)
Nozzle Losses	NL(n)
Nozzle Dimensions	ND(n)
Turbine Dimensions	TD(n)
Turbine Losses	TL(n)
Turbine Power	TP(n)
Turbine Torque	TT(n)
Coupling Torque	CT(n)
Input Torque	IT(n)
Output Torque	OT(n)
Input Power	IP(n)
Output Power	OP(n)
Drive Efficiency	DE(n)
Total Losses	LT(n)

Where n = the calculation number at which the variable is derived

Sun gears 107 and 108 are attached to the transmission shafts 1 and 8 respectively and the internal gear 109 is fixed. If the ratio of the pitch circle radii of the gears 107 to 104 is equal to the ratio of the pitch circle radii of the gears 108 to 109 then variation in the speed of the frame 102 will not affect the subframe 5 speed and the sub-frames speed will remain constant irrespective of the speed of the output shaft 110.

However, if the ratios are not equal then the sub-frames speed will vary with the speed of the output shaft 110. The features described above can be very useful in designing the torque speed characteristics and energy recirculation of the transmission device.

Referring now to Figure 10, the shaft 1 is fixed to the main frame 2. The shaft 3 is secured at either end to the main frame 2. The sub-frame 5 is rotatably mounted on the shaft 3. The shaft 8 is coaxial with the shaft 1 and is free to rotate with respect to the main frame 2. Gear wheel 7 is fixed to the shaft 8 and connected to a gear wheel 28 through idler gears 24 through to 27. The gear wheel 28 is fixed to the sub-frame 5 so that rotation of either shafts 1 or 8 will cause the sub-frame 5 to rotate on the shaft 3.

A gear wheel 30 is fixed to the shaft 3. Gear wheels 31 are fixed to shafts 35. The shafts 35 are rotatably mounted on the sub-frame 5 and drive the twin fly-ball governor type rotors 4. One end of each of the twin rotors 32 and 36 are fixed to the shaft 35 while the sleeve ends 33 and 42 are free to slide along the shaft 35 on splines provided on the shaft. In order to approach a true rotor under operating conditions the mass distribution 43,44,45,46 and 47,48,49,50 (Figure 12) are so arranged that when projected on a plane normal to the shaft 35 they will form a complete ring at their mean oscillating position.

In the alternative embodiment of the invention a reciprocating mass arrangement is used instead of a gyroscopic rotor arrangement. Such a device is shown in Figure 9.

In this embodiment masses 201 reciprocate radially with respect to the axis 3 of the sub-frame 5 within cylinders 202. Eight cylinders 202 are provided soaced at 45 degree angle steps radially around the axis 3. The axis 3 comprises a hollow shaft in much the same manner as the shaft 3 of figure 4, having a low pressure side 203 and a high pressure side 204. As the masses 201, which are spring loaded using springs 204,205, reciprocate fluid is successively sucked into one end 206 cylinder 202 through port 207 as it is expelled through port 208 at the other end 208, and then expelled through port 208 at the end 206 as it is sucked into the end 208 through its port 207. Fluid returns via 210,211a to the cylinders 202. From the ports 208 the fluid passes to the high pressure collection chamber 209, from where it enters the high pressure side 204 of the shaft 3 through an aperture 210. The fluid then passes along a fluid line 211 around the main frame 2 to the shaft 1, from where it may be used to power a turbine or the like. Having transferred its energy to the turbine the fluid is returned to the sub-frame 5 through a fluid line 202, into the low pressure side 203 of the shaft 3, through an aperture 213 and into a low pressure collection chamber 214. From the low pressure collection chamber 214 the fluid can again enter the chamber 202 through the port 207 to begin a repeat of the cycle. Seals 214 are provided at the point of connection of the sub-frame 5 to the shaft 3 to allow rotation whilst maintaining a fluidtight seal.

The inertial reaction force on the masses 201 due to the rotations of the main frame and the sub-frame is given by the following expression:

$$F = i\ [4a\ xN_6{}^2 \times Sin2\mu + r \times N_6{}^2 + r \times N_1{}^2 \times Sin^2\mu] +$$

$$j\ [-4a \times N_6 \times Cos2\mu + r \times N_1{}^2 \times Sin\mu \times Cos\mu]$$
$$k\ [-2r \times N_1 \times N_6 \times Cos\mu + 4a \times N_6 \times Cos2\mu \times Sin\mu]$$

Where r is the radial distance of the centre of gravity of masses 201 from the sub-frame axis

$r = r_0 + a\ Sin2\mu$

A, r_0 are constants

is the rotation of the sub-frame from the position shown.

N_1, N_6 are the rotational speeds shown

From the above equation it is apparent that:

a) maximum and minimum output torque occurs twice every rotation of the sub-frames about its axis 3;

b) by arranging the reciprocating masses 201 at 90 degree phase intervals (45 degree intervals around sub-frame axis) net torque significantly free from fluctuation can be generated; and

c) previous comments applicable to gyroscopic rotor systems are equally generally application to the reciprocating mass system.

The present invention may find particular applications in automobile transmission and power generation from fluctuating energy sources such as wind and wave. However, the invention is not limited to these applications and several other possible uses are envisaged.

In automobile applications the invention can either be used directly, possibly with a parallel combination of two units to give the desired characteristics or in conjunction with conventional device such as a gear box to enhance performance. The coupling energy imparted to the hydraulic fluid can in fact be used to prevent overspeeding occurring. Regenerative braking at infinitely variable speed ratio enables energy to be stored in a device such as fly wheel.

The transmission is readily decoupled without special decouplings devices such as a friction clutch. This feature provides an unparalleled advantage in the application of power generation from wave energy; the back flow of electricity during reversal of the input motion can be prevented. Another important feature relevant to power generation is that the constant output speed can be maintained with a varying input speed. By diverting the hydraulic energy to another source, which is easily accomplished, excess energy can be eliminated.

The invention may provide particular advantage in the application to aero industry where weight reductions are of value.

The transmission of the invention may find application in power generation due to its unique feature whereby a constant output speed can be maintained with a varying input speed, thus absorbing energy continuously from an energy source with varying intensity. Further, since the output shaft coupled to the generator is already rotating the rate of coupling of energy can be a small fraction of the maximum power transmitted.

Since the torque-speed characteristics of the invention can be designed with great flexibility, application of the invention can eliminate the need for special type prime movers. For example, if high starting torque is required the system parameters can be so chosen that the fluctuation of the angular momentum of the rotor is high when the output speed is zero and the fluctuation rapidly diminishes with output speed. Another possible application is for easy torque measurement. The output torque is proportional to the input speed and the input torque is proportinal to the output speed. Knowing the system parameters and the shaft speeds the required torque can be easily calculated, since the energy loss within the device can be estimated. The transmission may also find application where a specified torque must be repeatedly applied such as for fastening bolts in engineering work, without damaging the driving source. This is made possible by the fact that the driving source will not stall when the output shaft is locked, and in fact the torque on the driving source will be theoretically zero when the output shaft is locked.

It is thus seen that the present invention provides a continuously variable, automatic, low loss, reversible compact universal transmission which may find application in a wide variety of uses. The transmission of the present invention provides flexibility in the design of torque profiles and operating shaft speeds for a variety of applications.

Where in the aforegoing description reference has been made to integers or components having known equivalents then such equivalents are herein incorporated as if individually set forth. It is especially to be appreciated that reciprocating masses can replace rotating discs in the embodiments shown.

Further, although there are signficant advantages in transferring the coupling energy from the transmission, it is to be understood that coupling energy can equally well be transferred into the transmission. In this connection, the fluctuation means may consist of push-pull electromagnetic solenoids, triggered by the sub-frame rotation, which oscillates the mass distribution.

Although this invention has been described by way of example and with reference to possible embodiments it is to be appreciated that improvements and/or modifications may be made thereto without departing from the scope of spirit of the invention as defined in the appended claims.

I CLAIM:

1. A continuously variable transmission comprising an input shaft, an output shaft and a rotatably mounted main frame which have aligned rotational axes, a sub-frame rotatably mounted within the main frame, having an axis of rotation substantially perpendicular to the rotational axis of the main frame, and a mass distribution mounted within the sub-frame movable about an axis lying in a plane substantially perpendicular to the rotational axis of the sub-frame, characterised in that the main frame is fixed to the input shaft and is rotatable therewith, and the output shaft is connected, by way of a right angled gear train, to the sub-frame at the sub-frame rotational axis, such that when the input shaft and the output shaft rotate at different speeds the sub-frame is caused to rotate about its rotational axis, the transmission further comprising a fluctuation means independant of the output shaft to vary the angular momentum of the mass distribution to be a maximum and/or minimum once every predetermined number of rotations of the subframe so that any rotation of the main frame results in the generation of a net torque at the output shaft.

2. A continuously variable transmission according to claim 1 wherein the fluctuation means is driven or activated by the rotation of the main frame and the sub-frame and causes the mass distribution to oscillate about its mounting axis so that the angular velocity of the mass distribution is a maximum and/or minimum once every rotation of the sub-frame.

3. A continuously variable transmission according to claim 1 wherein the fluctuation means is driven or activated by the rotation of the main frame and the sub-frame, and causes the mass distribution to oscillate about its mounting axis so that the moment

of inertia of the mass distribution is a maximum and/or minimum once every rotation of the sub-frame.

4. A continuously variable transmission according to claim 1 wherein the mass distribution is a gyroscopic rotor.

5. A continuously variable transmission according to claim 4 wherein the fluctuation means consists of a double acting positive displacement pump driven by oscillating interial torque acting on the rotor in the vector direction parallel to its axis of oscillation.

6. A continuously variable transmission according to any one of claims 1,2 or 3 wherein the fluctuation means consists of a first device on the input shaft attached to the main frame to either energy derived from fluctuation of the angular momentum of the mass distribution, a second device mounted on the sub-frame to control the oscillation of the angular momentum of the mass distribution, and a means of transferring energy between the first and second devices.

7. A continuously variable transmission according to claim 1 wherein the mass distribution comprises four rotors disposed around the sub-frame at 90 degree phase intervals so that each rotor is dynamically identical when occupying any given position with respect to the plane of the main frame.

8. A continuously variable transmission according to claim 6 wherein the fluctuation means takes energy from the mass distribution to be substantially recirculated to the input shaft of the transmission.

9. A continuously variable transmission according to claim 8 wherein the energy is transferred using hydraulic fluid as the energy transfer medium.

10. A continuously variable transmission according to claim 9 wherein the energy is recirculated back to the input shaft by way of a pelton wheel or turbine.

11. A continuously variable transmission according to claim 10 further comprising a speed controller connected between the input shaft and the output shaft in parallel to the main frame to control the speed of rotation of the sub-frame.

12. A continuously variable transmission comprising an input shaft, an output shaft and a rotatably mounted main frame which have aligned rotational axes, and a sub-frame rotatably mounted within the main frame, having an axis of rotation substantially perpendicular to the rotational axis of the main frame, characterised in that a mass is movably mounted on the sub-frame to move substantially radially towards and away from or back and forth parallel to the rotational axis of the sub-frame, further, the main frame is fixed to the input shaft and is rotatable therewith, and the output shaft is connected, by way of a right angled gear train, to the sub-frame at the sub-frame rotational axis, such that when the input shaft and the output shaft rotate at different speeds the sub-frame is caused to rotate about its rotational axis, the transmission also including a fluctuation means independent of the output shaft to vary the radial distance from the rotational axis of the sub-frame, of the mass, to be a maximum and/or minimum once every half revolution of the sub-frame so that rotation of the main frame generates a net torque at the output shaft.

13. A continuously variable transmission according to claim 12 wherein the fluctuation means and the mass distribution consists of a spring loaded double acting piston pump, the piston being the reciprocating mass or being connected to the effort provided by the oscillating inertial force on the mass due to the sub-frame and/or main frame rotation.

14. A continuously variable transmission according to any one of claims 12 or 13 wherein the mass reciprocates substantially radially towards and away from the rotational axis of the sub-frame.

15. A continuously variable transmission according to claim 12 wherein the fluctuation means consists of a first device on the input shaft attached to the main frame to receive energy derived from fluctuation of the momentum of the mass, a second device mounted on the sub-frame to control the oscillation of the momentum of the mass, and a means of transferring energy between the first and second devices.

16. A continuously variable transmission according to claim 15 wherein the fluctuation means takes energy from the mass distribution to be substantially recirculated to the input shaft of the transmission.

17. A continuously variable transmission according to claim 16 wherein the energy is transferred using hydraulic fluid as the energy transfer medium.

18. A continuously variable transmission according to claim 17 wherein the energy is recirculated back to the input shaft by way of a pelton wheel or turbine.

19. A continuously variable transmission according to claim 18 further comprising a speed controller connected between the input shaft and the output shaft in parallel to the main frame to control the speed of rotation of the sub-frame.

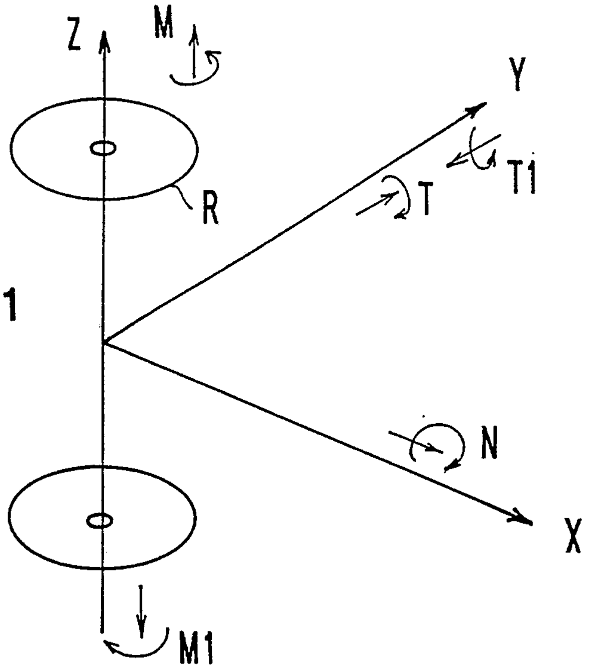

FIG.1

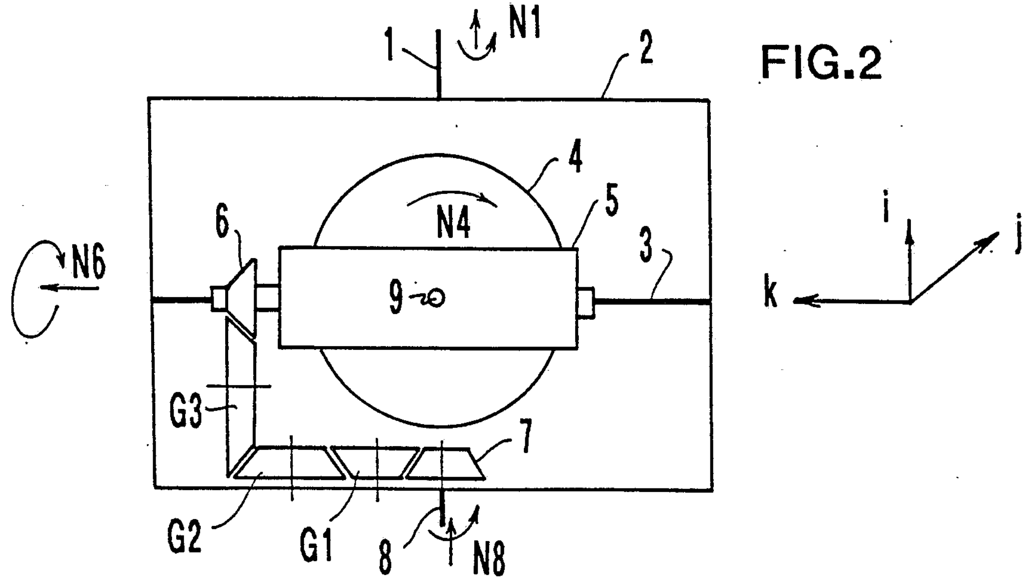

FIG.2

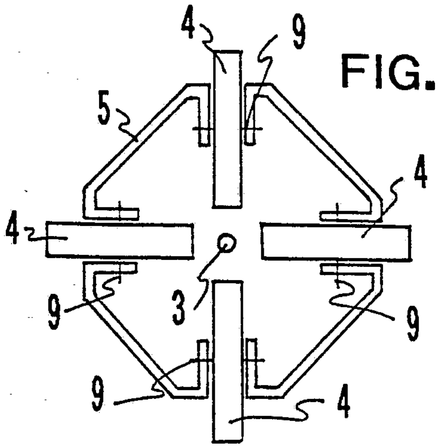

FIG.3A

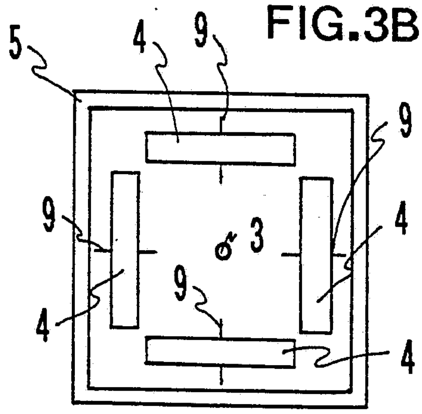

FIG.3B

FIG.4

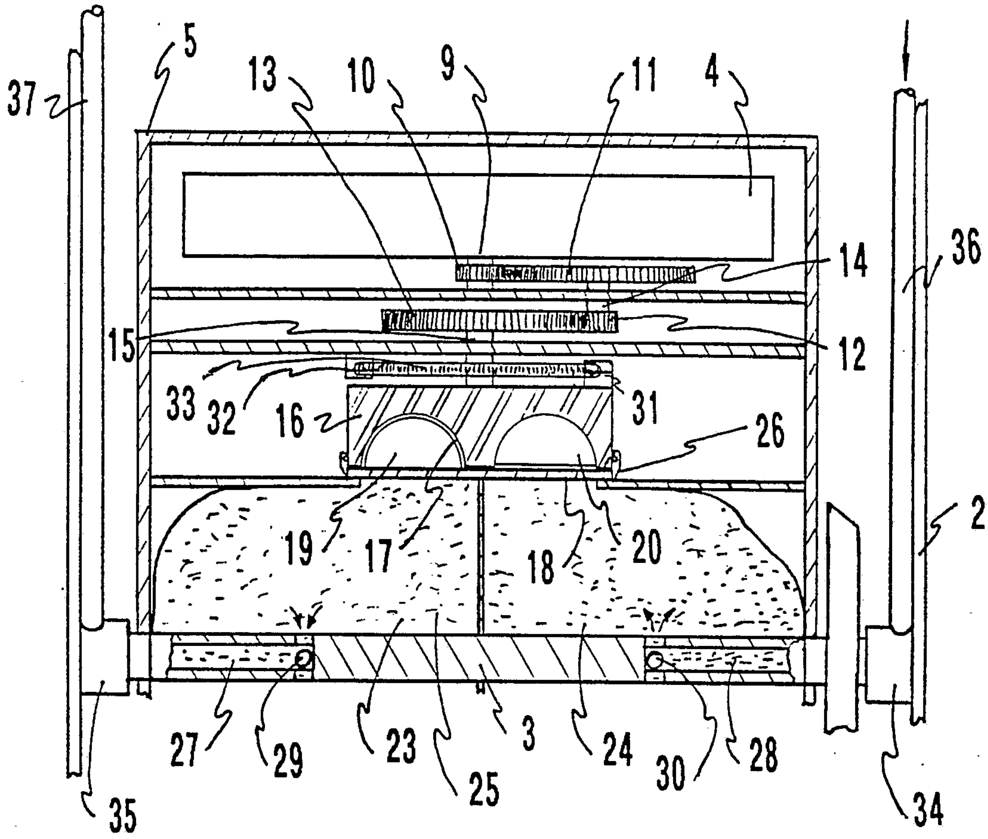
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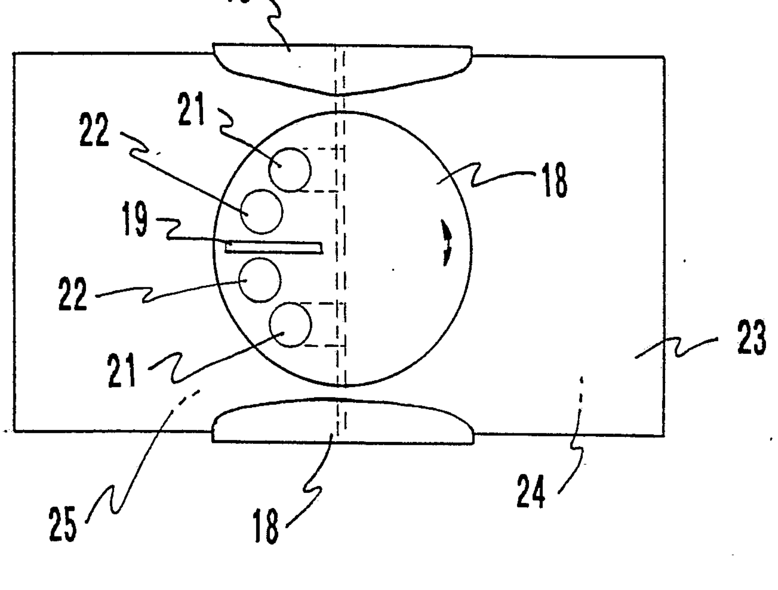
FIG.5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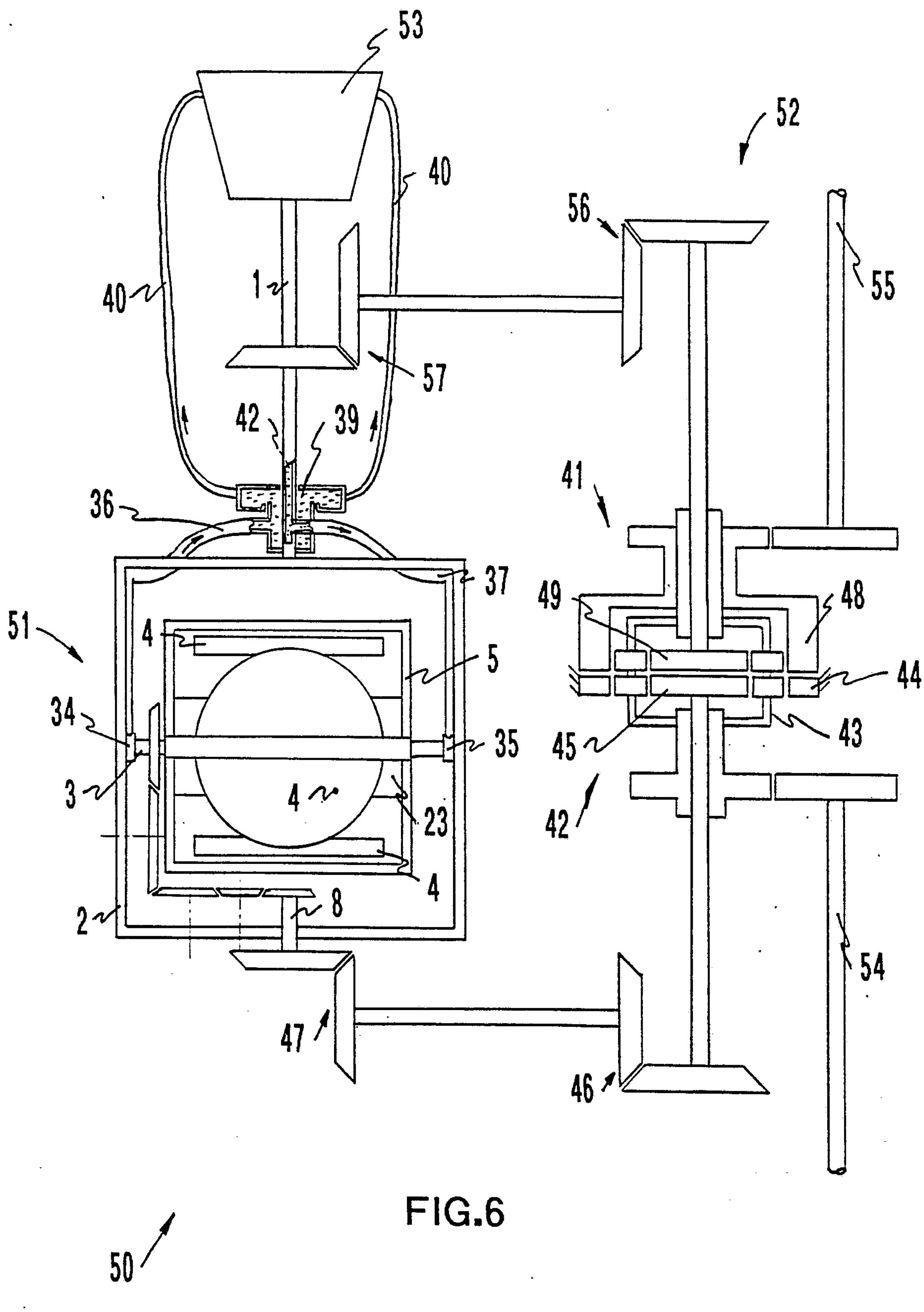
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FIG.6

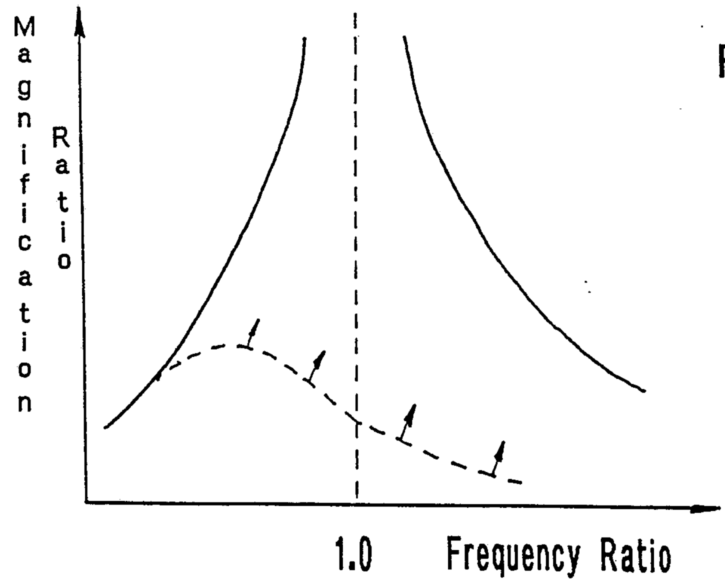
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FIG.8A

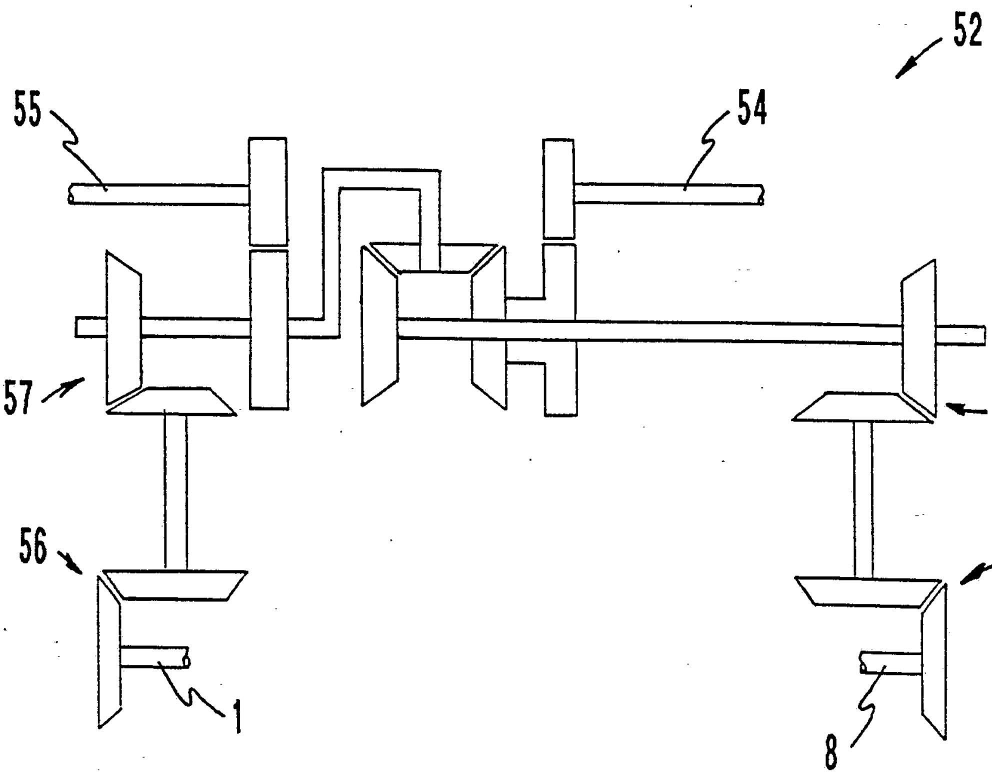
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FIG.7

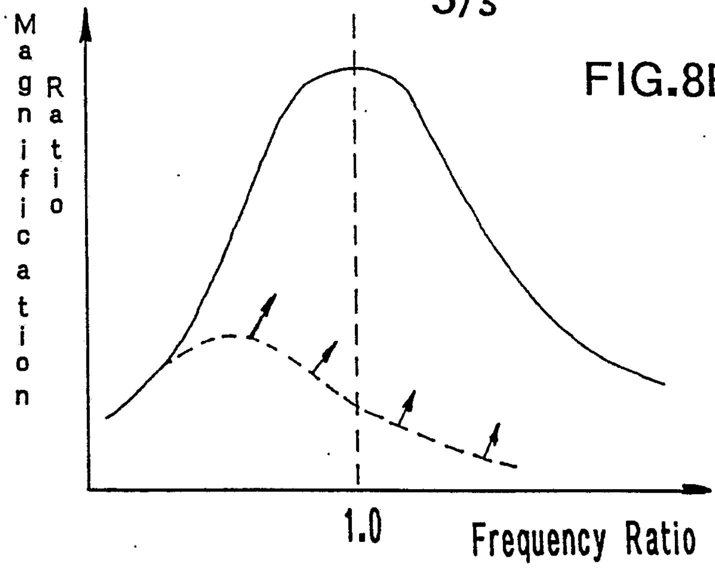
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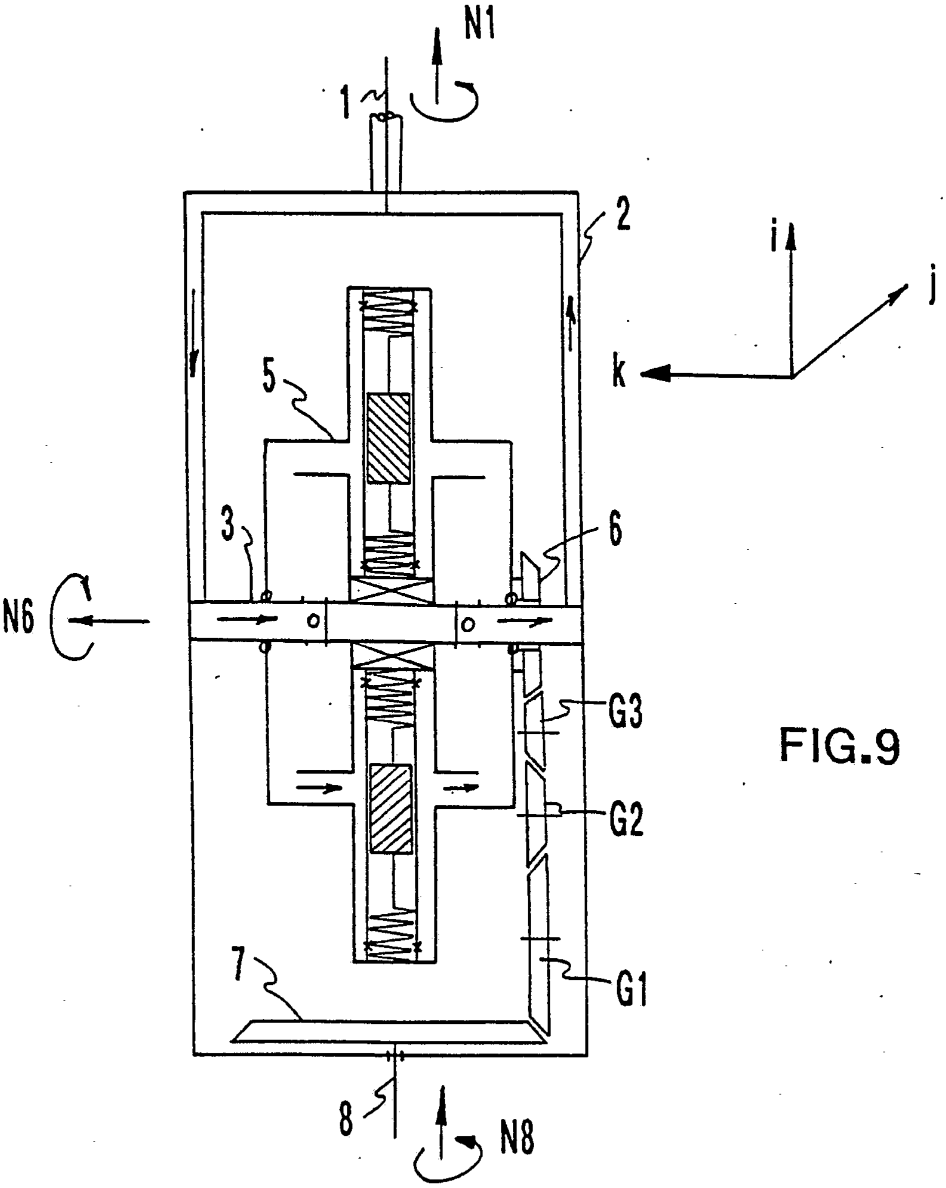
FIG.8B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N6
G3
G2
G1
7
8
N8

FIG.9

INTERNATIONAL SEARCH REPORT

International application No.
PCT/NZ92/00004

A. CLASSIFICATION OF SUBJECT MATTER

Int. Cl.[5] F16H 33/10

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
IPC F16H 33/02, 33/08, 33/10

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched
AU:IPC as above

Electronic data base consulted during the international search (name of data base, and where practicable, search terms used)
DERWENT

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to Claim No.
A	US, A, 4169391 (SCHONBERGER) 2 October 1979 (02.10.79) whole document	
A	US, A, 3851545 (GUMLICH) 3 December 1974 (03.12.74) whole document	
A	US, A, 2639631 (TAYLOR) 26 May 1953 (26.05.53) whole document	

☐ Further documents are listed in the continuation of Box C.

☒ See patent family annex.

* Special categories of cited documents :

"A" document defining the general state of the art which is not considered to be of particular relevance
"E" earlier document but published on or after the international filing date
"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)
"O" document referring to an oral disclosure, use, exhibition or other means
"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention
"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone
"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art
"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
5 March 1993 (05.03.93)	1 APR 1993 (1·4·93)

Name and mailing address of the ISA/AU	Authorized officer
AUSTRALIAN PATENT OFFICE PO BOX 200 WODEN ACT 2606 AUSTRALIA Facsimile No. 06 2853929	C.M. Wyatt Telephone No. (06) 2832538

This Annex lists the known "A" publication level patent family members relating to the patent documents cited in the above-mentioned international search report. The Australian Patent Office is in no way liable for these particulars which are merely given for the purpose of information.

Patent Document Cited in Search Report		Patent Family Member	
US	3851545	DE	144213
			END OF ANNEX