Jan. 14, 1936. A. V. BEDFORD 2,027,666

POWER TRANSMISSION DEVICE

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Fig. 1.

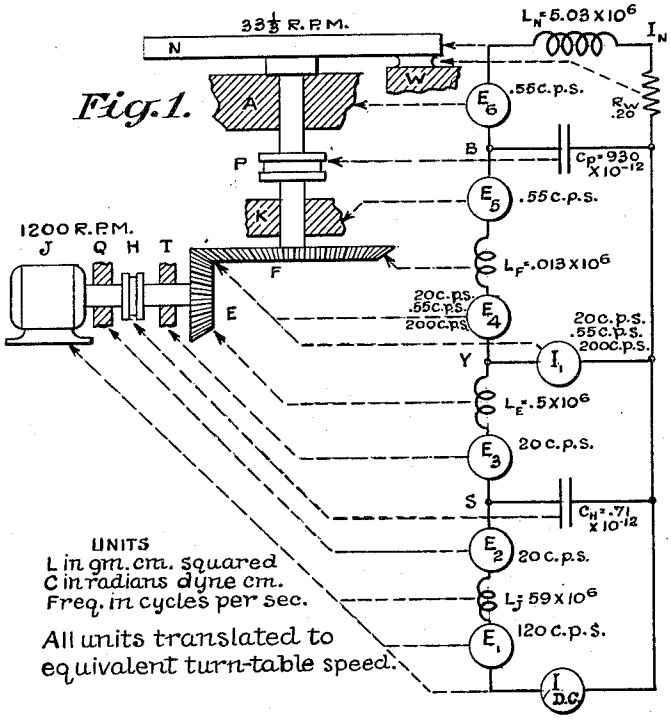


Fig. 2.

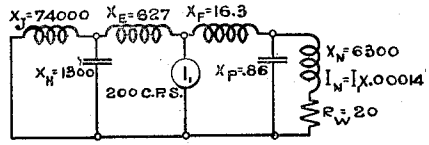


Fig. 3.

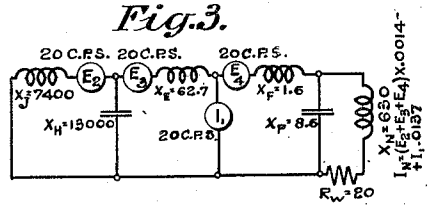


Fig. 4.

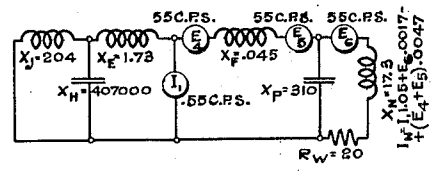


Fig. 5.

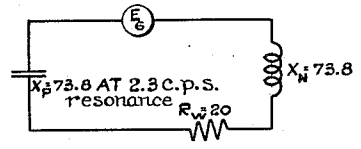


Fig. 6.

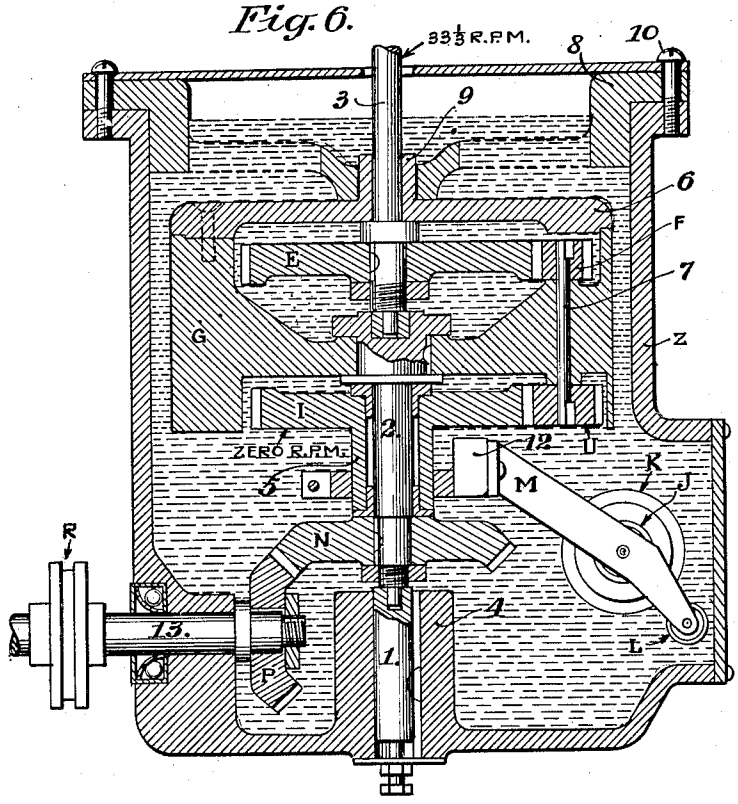
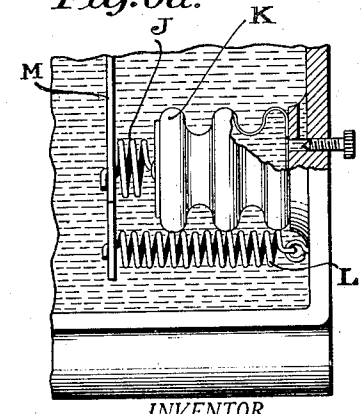


Fig. 6a.



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Fig. 7.

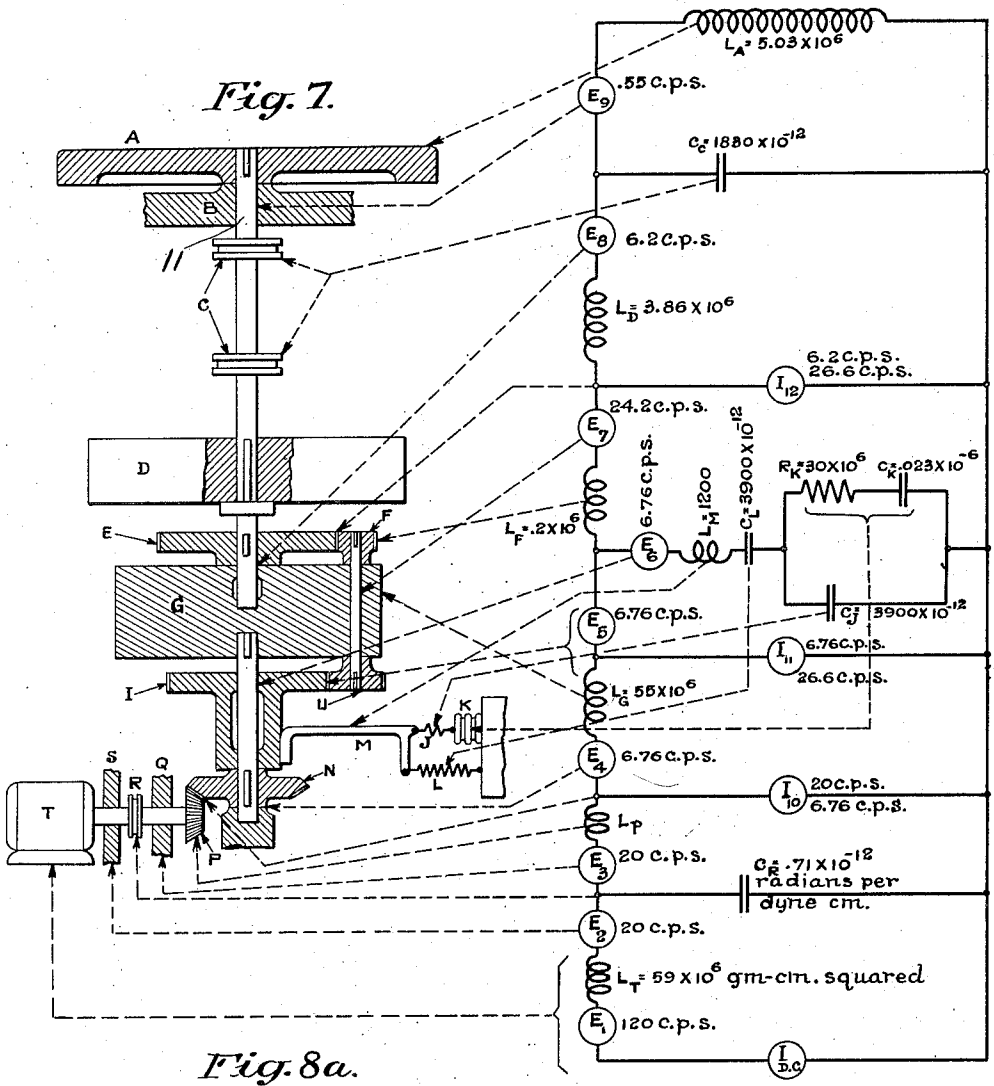


Fig. 8a.

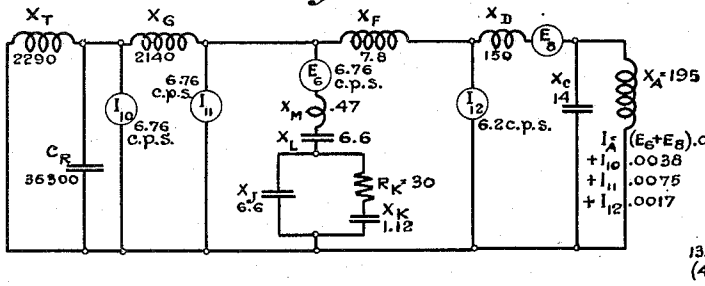


Fig. 8c.

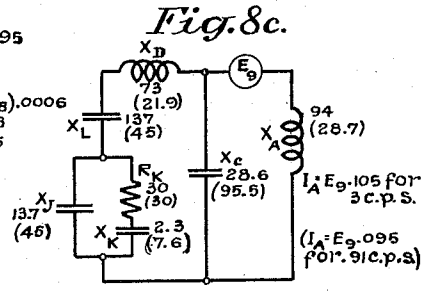
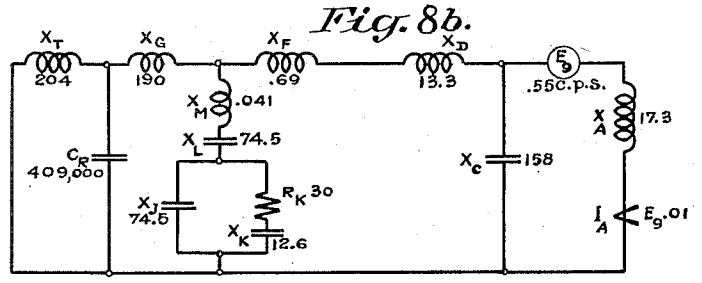


Fig. 8b.



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UNITED STATES PATENT OFFICE

2,027,666

POWER TRANSMISSION DEVICE

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Application January 28, 1932, Serial No. 589,353

3 Claims. (Cl. 74—306)

My invention relates to power transmission devices and more particularly to a turntable drive system for sound recording and reproducing apparatus and/or synchronized sound and motion picture apparatus.

In synchronized sound and motion picture apparatus, for example, the sound accompaniment for a particular film is usually, although not necessarily, in the form of a disc or tablet having a spiral sound groove therein corresponding to the sound which is desired to be reproduced in synchronism with the film. For reasons well known in the art, such a record tablet is rotated by turntable mechanism which is connected to a film projection apparatus in a manner adapted to cause rotation of the turntable at a predetermined speed, for example 33⅓ R. P. M., when the projection apparatus is in operation.

Considerable difficulty, however, has been experienced in driving a turntable of this character uniformly at the desired angular speed because a reduction gear mechanism is usually employed to reduce a motive source speed of 1200 R. P. M. to exactly 33⅓ R. P. M. at the turntable. In this connection, it has been found that the best gears which can be obtained at reasonable cost are so inaccurate in effective tooth pitch and in similar respects that they cause irregularities to appear in the speed of the turntable which have frequencies of recurrence equal to the turns per second of the turntable or to small multiples thereof.

Various attempts have been made in the past to filter out these disturbances or irregularities by means of elastic drive couplings and/or massive turntables. Experience has shown that when such couplings are made flexible enough to give sufficient filter action at such low frequencies, they are so very flexible that synchronism with a motion picture film is lost more or less. Massive turntables, also, have been found to be ineffective to overcome such irregularities in angular turntable speeds for the purpose set forth.

I have found, however, that by proper preselection of the gear train in the reduction gear mechanism, the frequency of irregularities therein can be increased, and when so increased, it is possible to more easily filter out these irregularities and thus render the filtering apparatus efficient to a degree heretofore unattainable. Thus I am able to realize the primary object of my invention, namely, to provide a driving mechanism of the character set forth which will be free from the defects and disadvantages present in driving mechanisms of the prior art.

A more specific object of my invention is to provide a drive coupling means between a driving element and a driven element whereby a turntable or analogous structures comprising the driven element may be driven at a predetermined angular speed uniformly.

Another object of my invention is to provide a planetary drive coupling between a driving element and a driven element whereby fluctuations generated by the inherent inequalities residing in the planetary drive coupling are absorbed to prevent their transmission to the driven member.

A further object of my invention is to provide a reduction gear drive system whereby the frequency of the disturbance introduced by the inherent inequalities in the gears employed is increased for a given speed, so that a given mass in the rotating parts of the system and a predetermined amount of flexibility in the coupling of the said mass will have a greater filtering effect.

Another object of my invention is to provide a planetary gear drives system for rotatable supports wherein irregularities or fluctuations of speed set up by the elements of the system, when in operation, are increased in frequency to a predetermined degree and simultaneously absorbed.

A further object of my invention is to provide a planetary drive system for rotatable supports which is adapted to generate physical disturbances at a high frequency of occurrence without increasing the magnitude thereof.

Another object of my invention is to provide a planetary gear drive system for rotatable supports wherein the gears smultaneously travel at a relatively high speed with respect to each other and at a slow speed with respect to a fixed point.

A further object of my invention is to provide a planetary drive system for rotatable supports wherein mechanical disturbances are generated at a high frequency of occurrence without increasing the magnitude thereof and are substantially simultaneously absorbed by an elastic medium.

The novel features characteristic of my invention are set forth with particularity in the appended claims. The invention itself, however, both as to its organization and its method of operation, together with additional objects and advantages thereof, will best be understood from the following description of a specific embodiment, when read in connection with the accompanying drawings, in which

Figure 1 is a schematic diagram of a simple turntable drive system and its electrical equivalents for purposes of analogy,

Fig. 2 is an analogous circuit diagram repre-

sentative of the tooth frequency in a simple gear drive system,

Fig. 3 is a similar diagram representative of the motor speed frequency in a simple gear drive system,

Fig. 4 is a similar diagram representative of the turntable frequency in a simple gear drive system,

Fig. 5 is a circuit diagram illustrating resonance at a frequency of 2.3 cycles per second in a simple gear drive system,

Fig. 6 is a vertical sectional view of a complete assembly of my planetary gear drive system,

Fig. 6a is a fragmentary detail view, in elevation, illustrating the manner in which a sylphon damping element is connected in my planetary drive system,

Fig. 7 is a schematic diagram of my planetary gear drive system and its electrical equivalents, also for purposes of analogy,

Fig. 8a is a circuit diagram analogous to my planetary gear drive system at 6.2 cycles per second impedance in megohms,

Fig. 8b is a similar diagram analogous to my planetary drive system, illustrating a .55 cycle per second impedance therein, and

Fig. 8c is a similar circuit diagram illustrating resonance in my planetary gear drive system of 3 and .91 cycles per second respectively, the impedances for .91 cycle per second being shown in parentheses.

Referring to Fig. 1 of the drawings, an analysis of a simple gear system discloses the fact that the relation of mass or moment of inertia to an inductance, and of a mechanical compliance to capacitance in which force or torque corresponds to voltage, and velocity to current, are analogous factors. Following along this line of reasoning, an inductance has $2\pi FL$ ohms positive reactance and a capacitance has $1/(2\pi FC)$ ohms negative reactance. Certain forms or kinds of friction are likewise analogous to electrical resistance. Friction such as is found in a common bearing is not a true resistance since it does not follow Ohm's law.

In such a bearing, the force E is approximately independent of the velocity I, whereas Ohm's law specifies that it should be proportional to velocity, I, or $E=IR$. This law makes the resistance R an independent constant since the velocity and the force referred to is relative only; i. e. between the parts affected.

Force due to eddy currents in a conductor moved across a magnetic field follows the true resistance law. Viscous liquids and gases also exhibit a true mechanical resistance in respect to the force required to move one liquid surface over another in shear. This action is used in the automobile shock absorber and similar damping devices in which a fluid is forced through a small passage or orifice.

Since questions frequently arise concerning the manner in which the elements of a complicated mechanical system should be connected in the analogous electrical circuit, the procedure in developing such a system is herein described for a clear understanding of my invention. As an example, let it be assumed that a synchronous motor is used to drive a record support or turntable through the agency of a pair of flexible couplings of conventional design, and a pair of bevel gears or worm and worm gear, with a viscous braking load applied to one face of the turntable as indicated at W in Fig. 1. The system illustrated in this figure will now be described in detail since the planetary gear system hereinafter described in accordance with my invention will be compared to the first mentioned system.

Considering the electrical synchronous motor previously referred to as a generator of force and velocity and the electrical power supplied to the motor as of a constant frequency and assuming that the motor does not fall out of synchronism, the said motor will then rotate at a constant average R. P. M., and may, therefore, be said to be analogous to a constant current, CI, direct current generator. This generator is indicated at I_{DC} in the analogous circuit in Fig. 1 and forces constant direct current through the inductance L_J which corresponds to the moment of inertia of the rotor.

Now the generated torque of a motor varies throughout the alternating power cycle and is shown by the alternating current generator E_1 in series with I_{DC} and L_J. It is to be understood that the generator I_{DC} will freely pass an alternating current and that E_1 will offer no obstruction to the direct current from I_{DC}. The generator E_1 is of the constant voltage, C. E., type since it represents torque variation rather than the rate of actual displacement of angular position. The lowest frequency of this torque variation is generally twice the electrical power supply frequency since the torque impulses occur on each half cycle. The harmonics of this torque might also be such as to produce any multiple of the driving frequency.

The drag of the motor bearing Q (Fig. 1) can be represented by a constant voltage direct current generator E_2 in series with L_J. However, experience has shown that high spots, non-uniform lubrication, and uneven loads commonly cause the frictional torque to vary throughout each revolution so that E_2 should be shown as a generator of direct voltage with superposed alternating voltage of frequencies corresponding to the bearing speed and multiples thereof. Since the direct component will only load the motor and will not effect uniformity of speed, any bearing may henceforth be considered as only a constant voltage alternating current generator. In this connection, it should be noted that E_2, corresponding to friction, is opposed to E_1, corresponding to torque, and hence is of opposite sign.

Further, with reference to Fig. 1, the coupling H is a torsionally flexible one and is analogous to a capacitance C_H. Since the rate of flexing of this coupling plus the velocity of the gear E instantaneously equals the velocity of the rotor of the motor, the condenser C_H must be connected in shunt rather than in series with the inductance L_E. The truth of this statement can be proved by applying Kirchoff's law to the circuit junction point S. This law states that the sum of all currents entering any point in a circuit is equal to the sum of all currents flowing out of the said point.

Similarly, bearing T is represented as an alternator E_3 acting in series with its associated mass L_E of the gear E. The teeth of this gear mesh with and drive gear F and if the teeth of both gears were perfect in spacing and shape, there would be a constant relation between their angular speeds. However, due to manufacturing errors and wear, no gear is ever perfect in either respect. If it is assumed that the gear teeth are so stiff that they do not flex appreciably and that the direct current load on the entire system is

enough to insure that the teeth are always in driving contact, in spite of the presence of backlash, these gear teeth imperfections will cause a definite change in their relative velocity and the forces acting on the teeth will be whatever is required to maintain their definite instantaneous angular position, so that the analogous alternating generator I_1, which produces the disturbance, is connected so as to force its current into the point Y, which connects the inductances L_E and L_F. Applying Kirchoff's law to the point Y, it will be readily apparent that the angular velocity of gear F is equal to that of gear E plus that introduced by the angular tooth errors in the said gears.

It is well known that gears of various types are very likely to bind when rough or "high" teeth thereof are in mesh. This condition produces an intermittent braking force on the gears which may be likened to the voltage of an alternator E_4 connected in series with the inductances L_E and L_F corresponding to the moments of inertia of the gears. The non-uniform component of drag of the bearing K is shown as a constant voltage alterator E_5. A similar procedure shows why the compliance C_P of the coupling P is shown in shunt and why the alternator E_6 is in series with the large turntable mass L_N.

The friction pad W has a film of viscous fluid between it and its bearing surface so that one component of the drag increases in proportion to the speed and hence is exemplified as a resistance R_W. This force opposes directly the rotation of the turntable N and hence is connected immediately in series with the inductance L_N.

Another law of Kirchoff states that the sum of all voltages acting in series around any closed circuit is equal to zero. As a matter of accuracy, this law can be applied to various parts of Fig. 1. Applying this to the circuit E_6 L_N R_W and C_P, it is found that the total force acting on the inertia of N and the resistance load of the turntable is that applied through the coupling P and that arising in the bearing A. It is possible that any slight errors in tooth shape or lack of tangency of the gear pitch circles would generate frequencies corresponding to the rate at which the teeth pass the center line through the gear centers and at harmonics thereof. In any event, the lowest frequency which we have to consider corresponds to the lowest gear speed, and, as hereinafter pointed out, is an important factor.

An analysis of the effect of gear eccentricity or inaccurate tooth spacing shows a definite variation in the relative angular velocities of the gears, which is independent of the torque on the gears. In such analysis, it is reasonably assumed that the gears are not appreciably elastic for the forces involved. It further shows the important fact that such gear variations are analogous to generators of the constant current and not of the constant voltage type. By constant current generator, I mean one whose current output is independent of loading or of the addition of series voltages. Such a generator is sometimes termed "a high internal resistance" generator.

The binding of two gears may be due to high spots or rough spots on or between the teeth of either one or both gears, as aforesaid. Now if there is only a high spot that binds on one gear, it will effectively offer a cycle of irregular braking on the gears which repeats itself during each revolution of the gear having the high spot. If both gears have high spots, for example, which bind independently of each other, the disturbing frequencies will be those corresponding to the gear speeds and their individual multiples. However, if each gear has a high spot which causes excessive friction when the two high spots are meshed together, it is evident that the lowest frequency disturbance generated will correspond to the teeth per second of the larger gear divided by the least common multiple of the number of teeth of the two gears. This may be a very low frequency compared to the gear speeds so that it is difficult to filter out. In the electrical analogy given, the effect of binding is represented as a constant voltage, C. E., generator in series with the inductances corresponding to the inertia of each gear.

In the design of turntable drive systems in accordance with my invention, the practice is to allow sufficient clearance between the gear teeth so that binding therebetween is impossible. Obviously, this does not require close limits of tolerance in gear manufacture, and the use of such gears will be assumed in the following analysis:

Numerical solution of simple gear system

A numerical analysis of the simple gear system illustrated in Fig. 1 will now be made to serve as an example of procedure and also as a standard to which may be compared the behavior of the more complicated planetary system hereinafter described.

The constants herein given have been assigned values in accordance with what they might be for reasonably good design, and are given in gram—cm. squared and radians per dyne—cm. units. Since a gear ratio which steps up angular speed is analogous to a transformer that steps up current, all the constants herein given have been changed to the basis of turntable speed by dividing all inductances and all capacitances by the square of the ratio of the turntable speed to the speed of the part being considered. The generators indicated at E_1, E_2, etc., in Fig. 1, are of the constant voltage (or low impedance) type and represent sources of torque or friction, while those indicated at I_1, I_2, etc., are of the constant current type and represent sources of angular velocity. The frequencies indicated, are the fundamental frequencies of 200 cycles per second tooth frequency and 20 and .55 cycles per second gear frequencies.

Referring specifically to Figures 2, 3, 4, 5 and 6 in Fig. 2, the impedances in megohms for 200 cycles per second as calculated, shows that I_N equals only about .00014 of I_1, and that the filtering or damping effect for this frequency is tremendous. A similar computation shows that the impedances at 120 cycles per second are such that the effect of the alternator E_1 in Fig. 1 is insignificant in forcing current thru the inductance L_N. These considerations show, further, that due to the high moment of inertia of the turntable N and the moderately flexible couplings H and P, there can be no appreciable high frequency rotational oscillations in the turntable N.

In Fig. 3, the impedances for the 20 cycles per second disturbances are given and calculations show that the 20 cycles per second "current" in the turntable inductance I_N is only .0014 times $(E_2+E_3+E_4)$ plus .013 times I_1 or relatively small values of disturbance.

In Fig. 4, the values for a frequency of .55 cycle per second are given, but in this instance, the value for I_N is 1.05 times I_1 plus .0017 times E_6 plus .0047 times (E_4+E_5). This shows that, for this frequency, the disturbance I_N intro-

duced by the larger gear is slightly increased instead of being decreased by the action of the filter system. This is the distinctly weak point of this drive system since it means that the error in the angular turntable position at any moment is about as great as the fundamental error in the angular tooth pitch in the lowest speed gear.

Considering the circuit in Fig. 4 at its resonance will give a clue to its response to transient disturbances, such as starting of the system, for example. An inspection of Figs. 3 and 4 shows that, for any frequency at which X_N and X_P might be equal, the remainder of the circuit has so much higher impedance that it will not appreciably affect the resonance and hence may be neglected. This gives us the circuit in Fig. 5 which, although not "critically damped", has quite a high "time" decrement, since the resonant frequency is 2.3 cycles per second. If, by chance, it should have a disturbance E_6 of this frequency, the current would be opposed only by the resistance Rw.

Description of the planetary drive system

Referring now, in detail to Figs. 6 and 7 wherein a vertical cross-sectional view of the planetary gear drive system constructed in accordance with my invention is shown, a motor T of the type rated, for example, at 1200 R. P. M. is arranged to drive a shaft 13 through a flexible coupling R of conventional design. The shaft 13 extends through a suitable bearing in the gear housing Z and carries on its inner free end a bevel gear P (of, say, 25 teeth) suitably keyed thereon.

Vertically disposed in axial alignment in the housing Z are a plurality of interconnected shaft sections 1, 2 and 3, of which sections 2 and 3 are rotatable independently of each other, while section 1 is adjustably disposed in an inwardly extending boss 4 formed with the housing Z, and is adapted to act as a pivot support for the intermediate shaft 2. A bevel gear N (of, say, 74 teeth) is suitably secured to the shaft section 2 adjacent its lower end and is adapted to mesh with the driving gear P. Hereinafter, the shaft 3 will be designated as the driving shaft.

Loosely mounted on the shaft section 2 immediately above the gear N is a spur gear I having a depending hub 5 which spaces the gear I from the bevel gear N. Keyed to the upper end of the shaft section 2 is a flywheel G having a recess in both faces thereof. The recess in the bottom face thereof is adapted to provide a space sufficient for the reception of the spur gear I having 118 teeth. The recess formed in the upper face of the flywheel G has a diameter sufficient to provide a similar clearance space for the reception of a spur gear E having, for example, 120 teeth, the gear E being secured to the lower end of the shaft section 3 carried by an annular bearing plate 6 suitably secured to the upper face of the flywheel G.

A vertically disposed rotatable shaft 7 extends through the flywheel G at a predetermined point to one side of the axial center thereof and adjacent its periphery, the free ends of said shaft extending beyond the upper and lower face of the recesses formed in said flywheel for the fixed reception thereon of a pair of planetary pinions U and F having, for example, 30 and 28 teeth respectively. The said pinions are adapted to mesh, respectively, with the spur gears I and E, as clearly shown in the drawings.

An annular cap bearing plate 8 provides the upper bearing for the joined shaft sections 1, 2 and 3, the bearing plate 8 having a concentric bore adapted to receive an upstanding bearing hub 9 formed on the plate 6. The cap bearing 8 is attached to the upper flanged periphery of the housing Z by means of suitable screws or bolts 10.

Secured to the shaft section 3 at a point above the cap bearing 8 is a second flywheel D (Fig. 7), and finally, a flexible coupling C from which a continuing shaft section 11 extends thru a bearing B which is supported independently of the housing Z, and a turntable A secured to the uppermost end of the shaft extension 11. It should here be noted that the driving relation of the motor with the gear housing Z and the turntable A with the said housing is such that the motor, gear housing and turntable are supported independently of each other for the purpose of minimizing vibration.

Secured to the hub of the spur gear I is a split collar 12 which in turn carries a radially extending arm M whose function is to prevent substantial rotation of the gear I. To prevent such rotation, the arm M is connected by means of a spring J to an oil-filled, vented, damping, metal sylphon K, whose other end is held in abutment with the side wall of the gear housing Z. The arm M is further restrained by a spring L which has one of its ends anchored to the end of the arm M, and the other end to the side wall of the gear housing Z. Suitable needle valve means is provided in conjunction with the sylphon K for regulating the flow of oil into and out of the said sylphon as clearly shown in Fig. 6a. In this connection, it is to be noted that the elements within the housing Z are submerged in a fluid, preferably oil, whose level is indicated.

In operation, the flywheel G rotates at about 405 R. P. M. when gears having the number of teeth indicated heretofore are employed, but, since the pinion F is only slightly smaller than the pinion U, the gear E rotates at a speed of 33⅓ R. P. M. as illustrated by the formula

$$\frac{\text{Speed of } E}{\text{Speed of } N} = \left(1 - \frac{IF}{UE}\right)$$

where I, F, U and E represent, respectively, the number of teeth on each gear and pinion.

At this point, it should be noted that no gear in the system runs at as low a speed as 33⅓ R. P. M. with respect to its meshed mate. Even the gear E, which has an angular speed of 33⅓ R. P. M. about its own axis, contacts each of its teeth with gear F about 375 times per minute. This means that any irregularity in its angular tooth spacing will make a disturbance which occurs 11.3 times as often as it turns in space on its own axis (33⅓ R. P. M.). As will be seen later, this gives a frequency which is very much easier to filter than is the frequency of 33⅓ cycles per minute. In this fact lies the chief advantage of my invention.

It should be noted, further, that the support of all the moving parts is such that no bearing in the gear box runs at a speed lower than 375 R. P. M. This, again, is an advantage in filtering since in this manner, only high frequencies can be generated by irregularities in bearing friction. It has even been found desirable to support the major portion of the weight of the heavy turntable on the bearings in the gear box thru a longitudinal thrust in the flexible couplings above the gear box in order to further decrease the probable low frequency irregularity of torque in the stationary turntable bearing. It should be observed that the static friction of the bearing of

gear I has been removed by the rotation of its journal. This increases the sensitivity of the filtering and damping action of the arm M.

Analysis of planetary system

Fig. 7 gives a schematic view of the turntable and drive system, including the flexible couplings and the damping device. The equivalent electrical circuit is shown and dotted arrows indicate the constants represented. The frequencies shown are the fundamental frequencies, and it is to be understood that integral multiples of these frequencies may exist equally well. The coil springs L and J and sylphon K are in reality so located as to directly oppose the rotation of arm M, but are so shown schematically that their connections can be seen. All constants are given in c. g. s. (centimeter gram seconds) units translated to turntable speed for convenience in calculation.

The impedances of the various elements of the circuit have been worked out for the fundamental frequency of the gear E (6.2 cycles per second) and are given in mechanical megohms in Fig. 8*a*. All generators of disturbances of about this frequency are shown, and the current I_A in microamperes or microcentimeters per second in the branch corresponding to the moment of inertia of the turntable is calculated approximately in terms of the several initial sources.

$$I_A \text{ for } 6.2 \text{ c.p.s.} = (E_6+E_8).0006+ I_{10}.0038+I_{11}.0075+I_{12}.0017$$

(E_6 and E_8 are in volts or dyne cm. torque)

The factor .0006 for E_6 and E_8 is smaller than .0017 and .0047 which were the similar factors for the simple gear system previously analyzed. This means that, to obtain the same degree of freedom from bearing troubles, the bearings which carry the turntable weight will not have to be made as uniform for the planetary system as for the simpler gear system. The .0075 factor for I_{11} and the .0017 factor for I_{12} indicates that only those fractions of the actual angular error of tooth displacement of the gears I and E reach the turntable. However, it should be acknowledged here that although the planetary system has not increased the total unfiltered angular error of the gears I and E (which corresponds to charge and not current), it has increased the frequency of the displacement (or charge) about 11.2 fold, which means a corresponding increase in rate of angular displacement (or current). For comparison purposes, the factors can be changed to .0842 for I_{11} and .0191 for I_{12} to give a true picture of the decrease in the effect of gear error. This still shows a minimum of about 12.5 fold gain over the factor 1.05 of the simple gear system. Fortunately, the speeding up of bearings does not increase their probable variable torque so that E_6 and E_8 of the planetary system are not increased.

Fig. 8*b* shows that the factor for the table bearing, E_9, in producing current in X_A, is less than .01 and that the value of E_9 can be reduced by perhaps 10 fold by supporting most of the table weight on bearings in the gear box (as mentioned hereinbefore). This will give a combined factor of less than .001 which is better than the .0017 obtained for the simple gear system.

It should be readily apparent, from a consideration of the foregoing description of a preferred embodiment of my invention, that it marks an important forward step in the art of sound reproduction. Through its use, the angular velocity of rotating parts, such as disc record supports or driving sprockets for films, may be held substantially constant even though the gears involved are not manufactured by precision methods. Accordingly, my invention actually enables a reduction in the cost of such devices while, at the same time, it provides vastly improved operation.

My invention is not to be restricted to the specific structural details and/or arrangement of parts hereinbefore set forth, as various modifications thereof may be effected without departing from the spirit and scope thereof. I desire, therefore, that only such limitations shall be imposed as are indicated in the appended claims.

I claim as my invention:

1. In combination, a driving shaft subject to slight speed fluctuations, a driven shaft, an intermediate shaft co-axial therewith, means for imparting to said driven shaft a uniform speed free from the speed fluctuations of said driving shaft, said means comprising an inertia member carried by said intermediate shaft, a planetary gear mechanism connecting said intermediate shaft and said driven shaft and including a substantially stationary gear adapted to respond to a change in the speed of said driving shaft, and means connected to said gear for damping out the speed fluctuations transmitted by said driving shaft.

2. In combination, a driving shaft subject to slight speed fluctuations, a driven shaft, an intermediate shaft coupled to said driven shaft for rotation relative thereto, and means for imparting a uniform speed to said driven shaft free from the speed fluctuations of said driving shaft, said means comprising a flywheel carried by said intermediate shaft, a planetary gear mechanism positively connecting said intermediate and driven shaft and including a reaction gear carried by said intermediate shaft, said reaction gear having a yielding movement in response to said speed fluctuations, a dash-pot adapted for damping said yielding movement, and means connecting said reaction gear and dash-pot.

3. Driving mechanism for phonographic apparatus comprising a housing, a driving shaft subject to slight speed fluctuations journaled in said housing, a driven shaft journaled in said housing and comprising an intermediate section and a final driven section, said final driven section being rotatable relative to said intermediate section, means for imparting a uniform speed to said final driven section free from the speed fluctuations of said driving shaft, said means comprising a flywheel carried by said intermediate section, a planetary gear mechanism associated with said flywheel and coupling said intermediate and final driven shaft sections, said planetary gear mechanism including a reaction gear, a dash-pot secured in said housing, and means connecting said reaction gear and said dash-pot for damping out the speed fluctuations transmitted by said driving shaft.

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