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(56) Documents Cited:

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US 5827146 A	**US 3257867 A**
US 20080242468 A1	**US 20060276295 A1**
US 20050227809 A1	

(58) Field of Search:
INT CL **F16H**
Other: **ONLINE: EPODOC, WPI**

(54) Title of the Invention: **Transmission system**
Abstract Title: **Transmission comprising an infinitely variable drive and a differential**

(57) A transmission system having a single input 12 and a differential 14 having a first shaft 16, a second shaft 18 and a third shaft 20. The first, second and third shafts 16, 18, 20 are coupled together via the differential 14 and the single input 12 drives the first shaft 16 and the second shaft 18. The single input shaft 12 may be connected to the first shaft 16 via an infinitely variable drive 22 which is configured to infinitely vary the drive ratio between the single input 12 and the first shaft 16 so as to provide an infinitely variable overall drive ratio between the single input 12 and the third shaft 20. In another embodiment the infinitely variable drive 22 connects the single input shaft 12 to the first shaft 16 and the second shaft 18. In a further development seven differentials may be provided wherein the seventh differential has the third shaft 20.

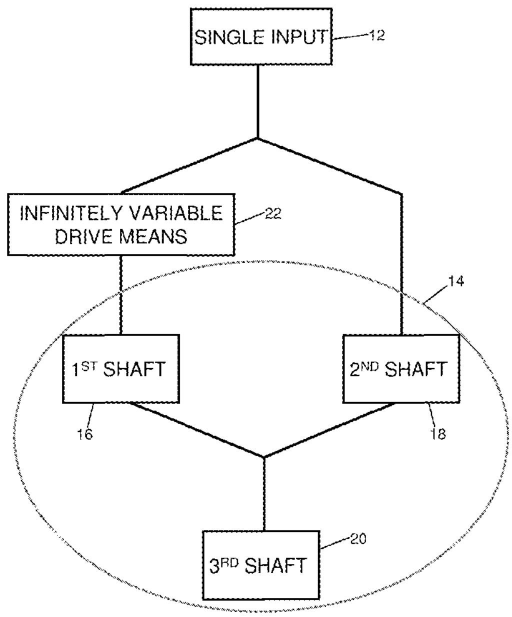


FIG 1

GB 2471859 A

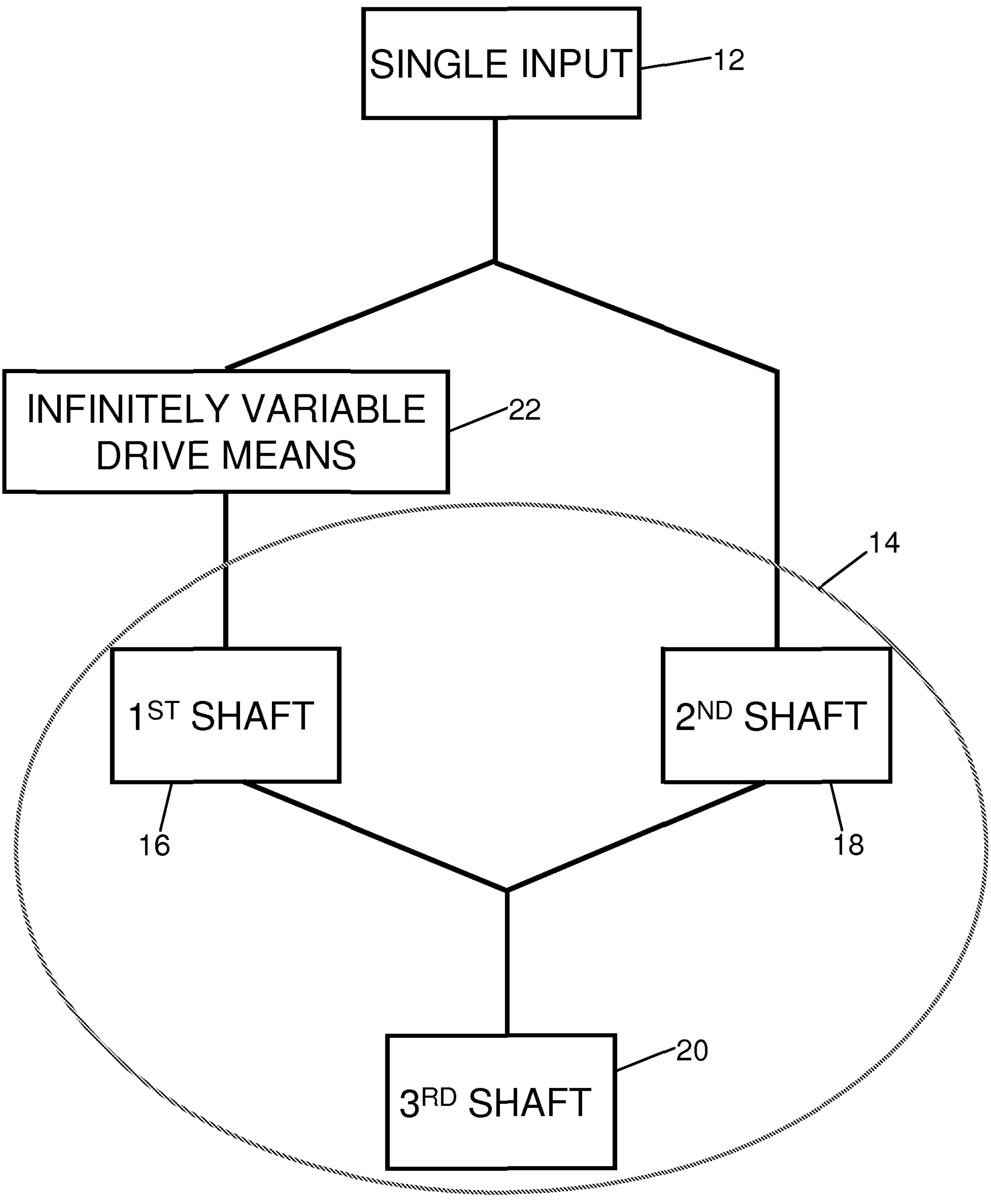

SINGLE INPUT
12
INFINITELY VARIABLE DRIVE MEANS
22
14
1ST SHAFT
2ND SHAFT
16
18
3RD SHAFT
20

FIG 1

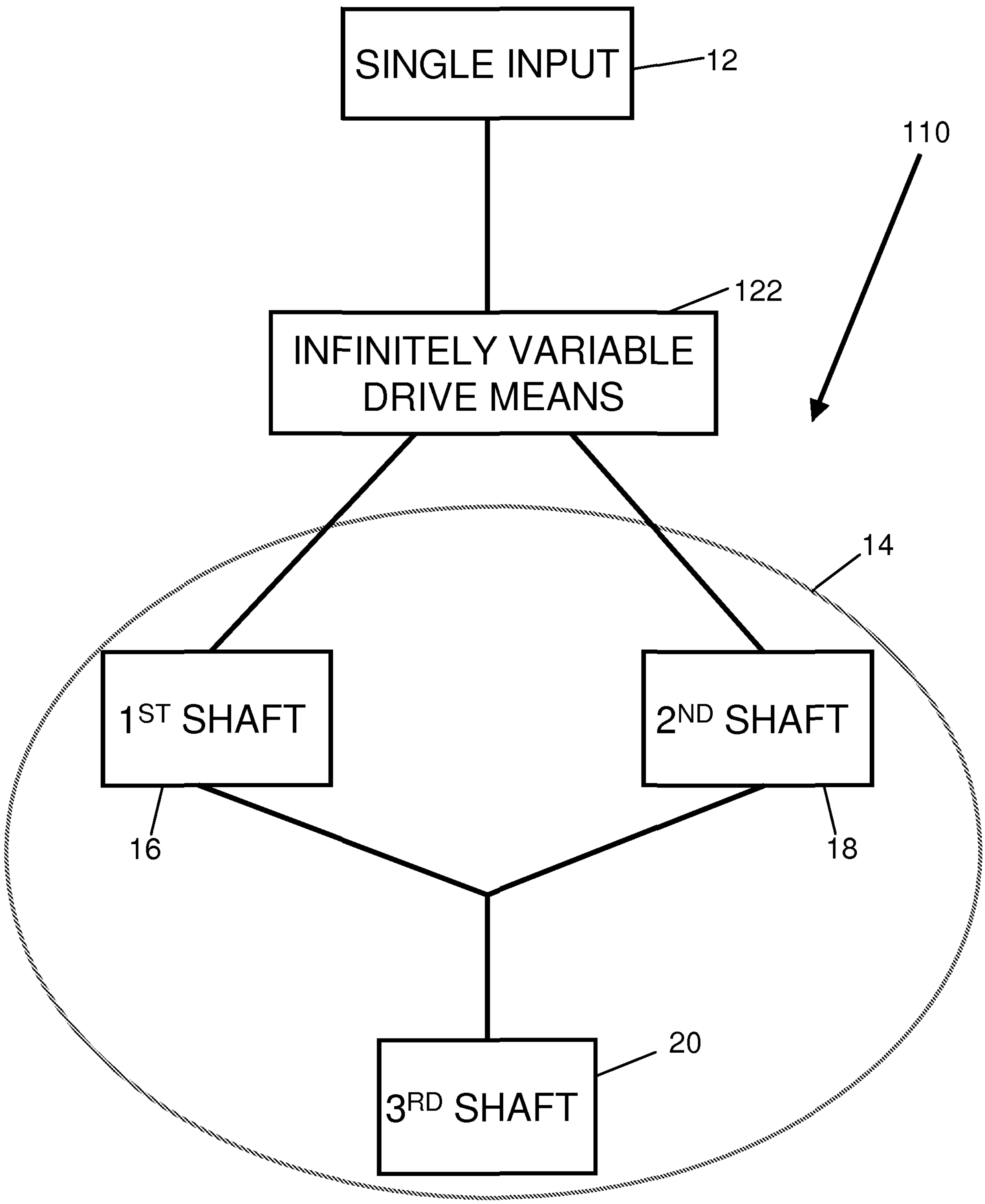

SINGLE INPUT
12
110
122
INFINITELY VARIABLE DRIVE MEANS
14
1ST SHAFT
2ND SHAFT
16
18
20
3RD SHAFT

FIG 2

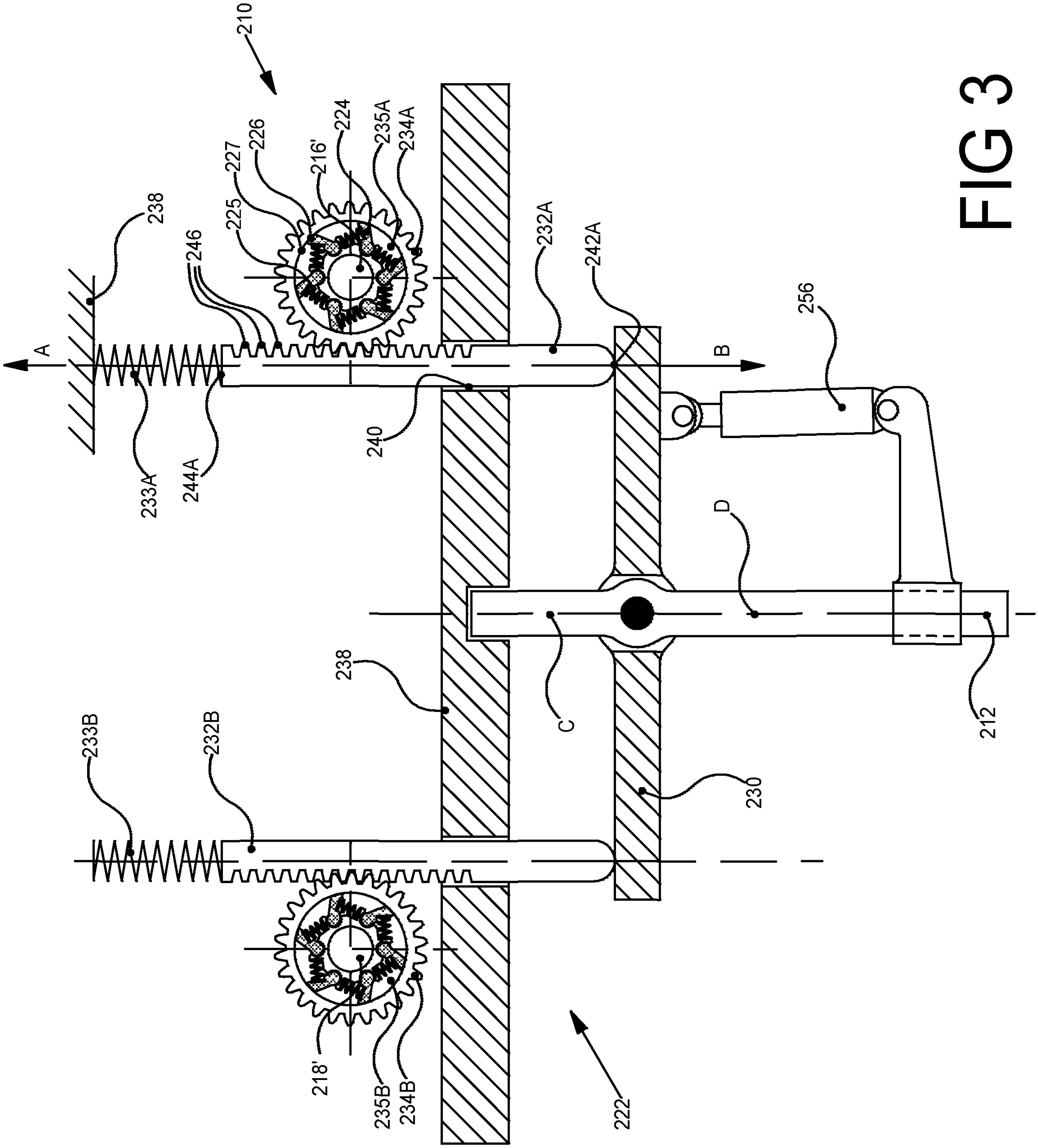

210
238
246
227
226
225
216'
224
235A
234A
232A
242A
256
A
B
233A
244A
240
D
C
212
238
230
233B
232B
218'
235B
234B
222

FIG 3

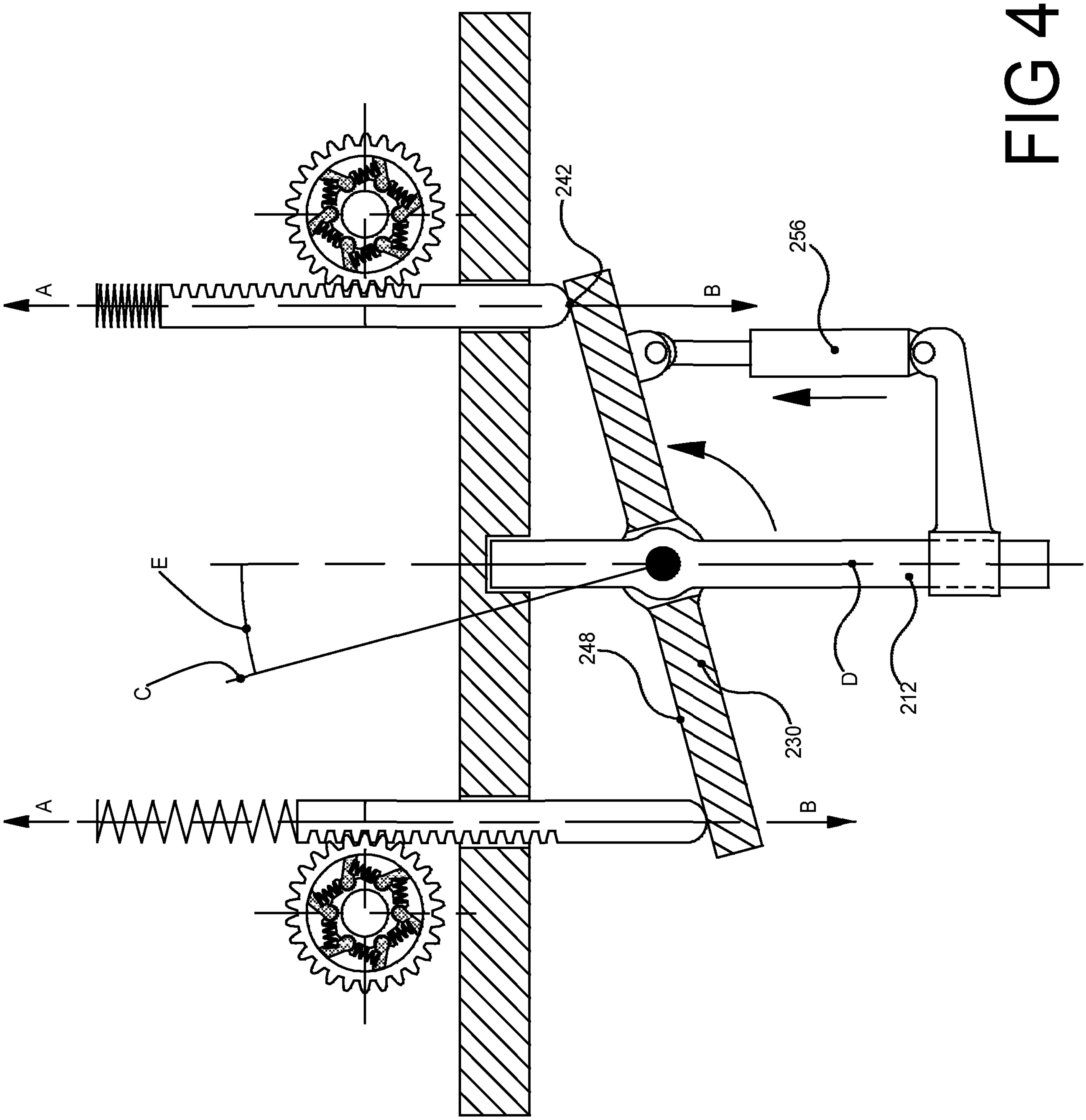


FIG 4

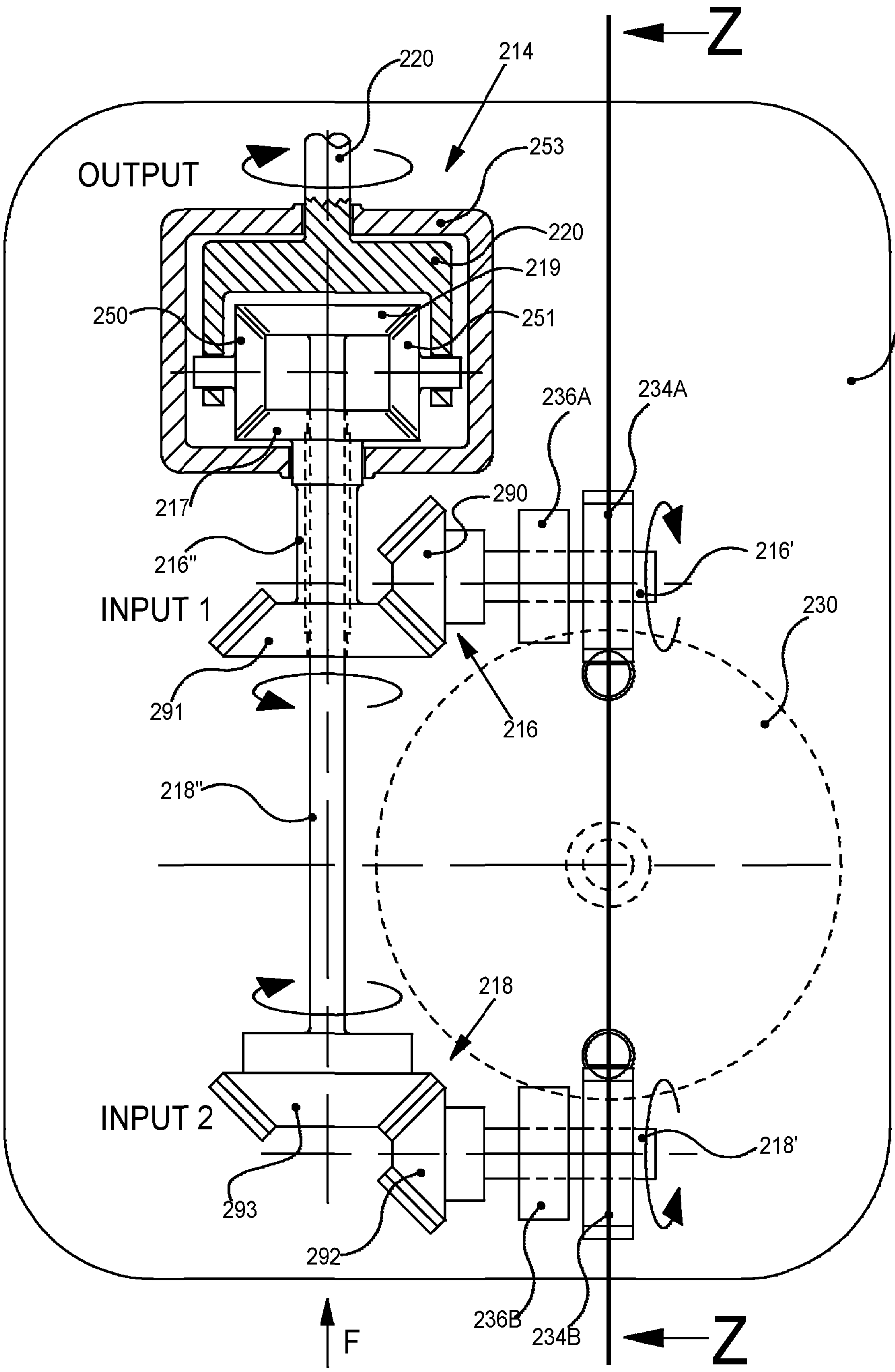

Z
220
214
OUTPUT
253
220
219
250
251
238
236A
234A
290
217
216''
216'
INPUT 1
230
291
216
218''
218
INPUT 2
218'
293
292
236B
234B
F
Z

FIG 5

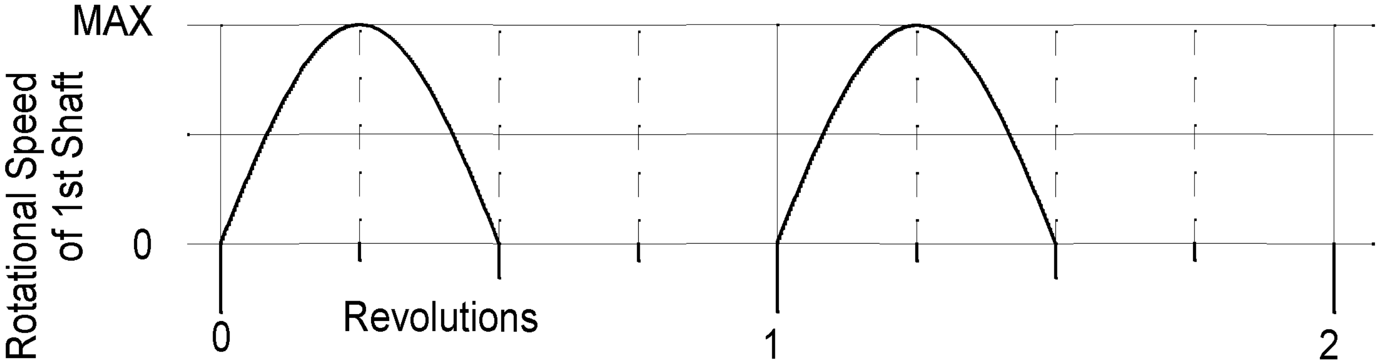


FIG 6

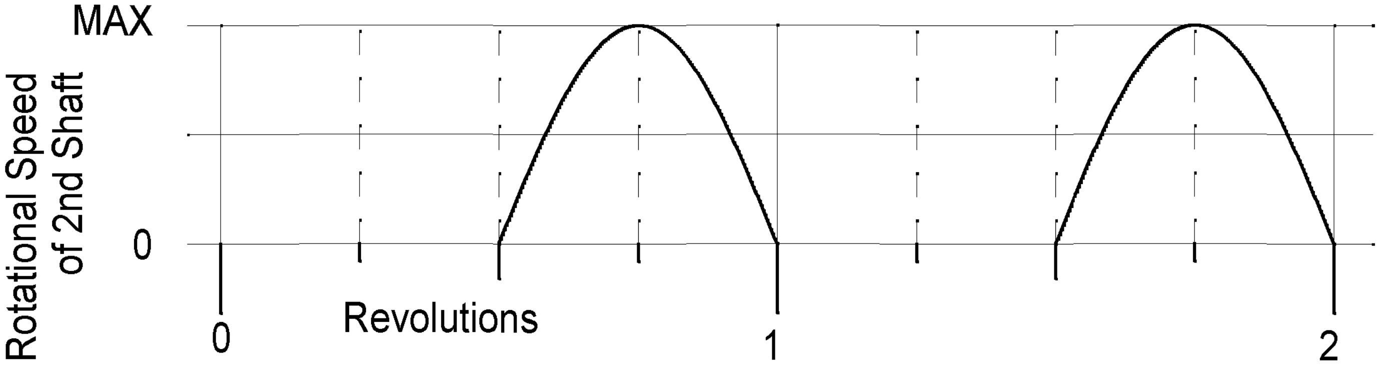


FIG 7

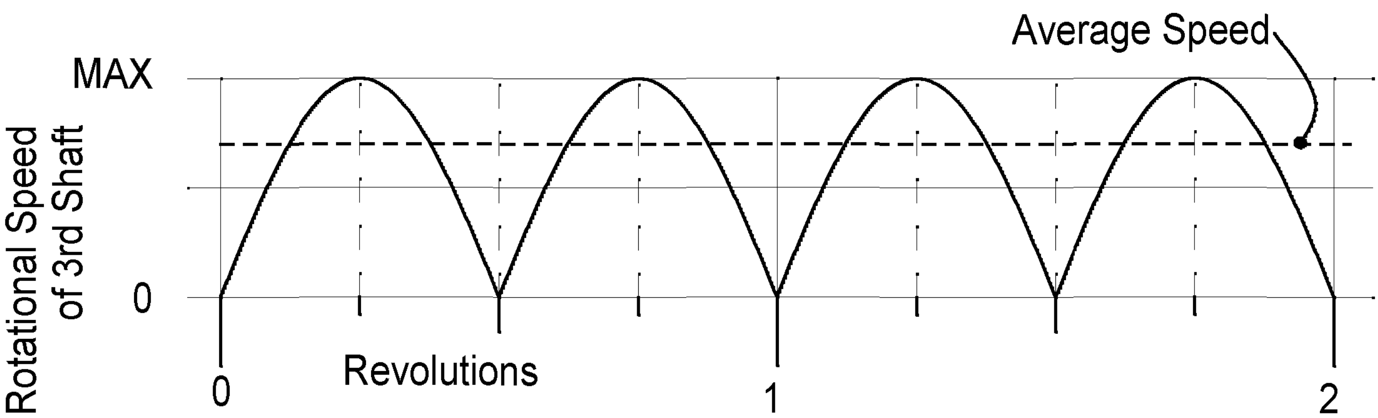
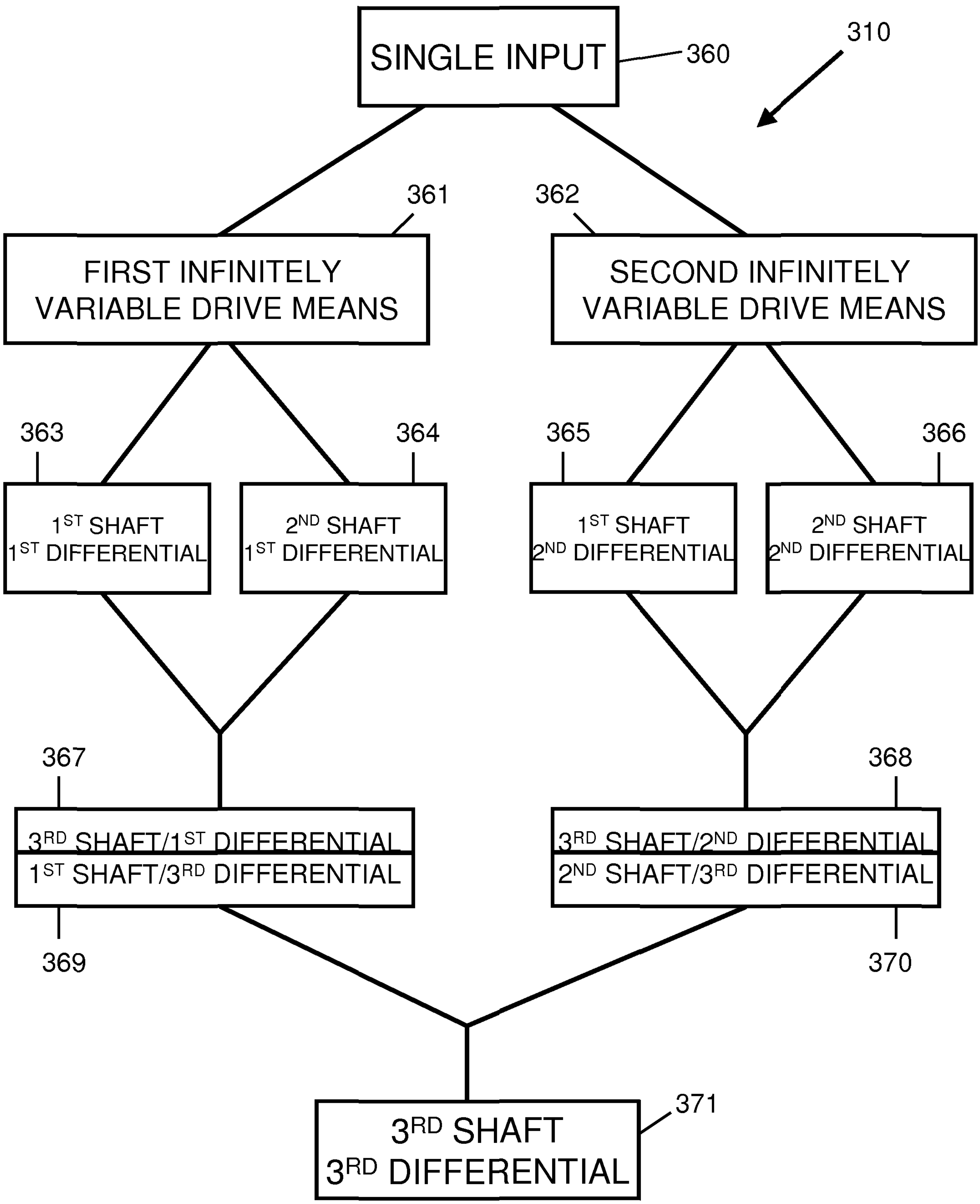


FIG 8



SINGLE INPUT
360
310
361
362
FIRST INFINITELY VARIABLE DRIVE MEANS
SECOND INFINITELY VARIABLE DRIVE MEANS
363
364
365
366
1ST SHAFT 1ST DIFFERENTIAL
2ND SHAFT 1ST DIFFERENTIAL
1ST SHAFT 2ND DIFFERENTIAL
2ND SHAFT 2ND DIFFERENTIAL
367
368
3RD SHAFT/1ST DIFFERENTIAL
1ST SHAFT/3RD DIFFERENTIAL
3RD SHAFT/2ND DIFFERENTIAL
2ND SHAFT/3RD DIFFERENTIAL
369
370
3RD SHAFT 3RD DIFFERENTIAL
371

FIG 9

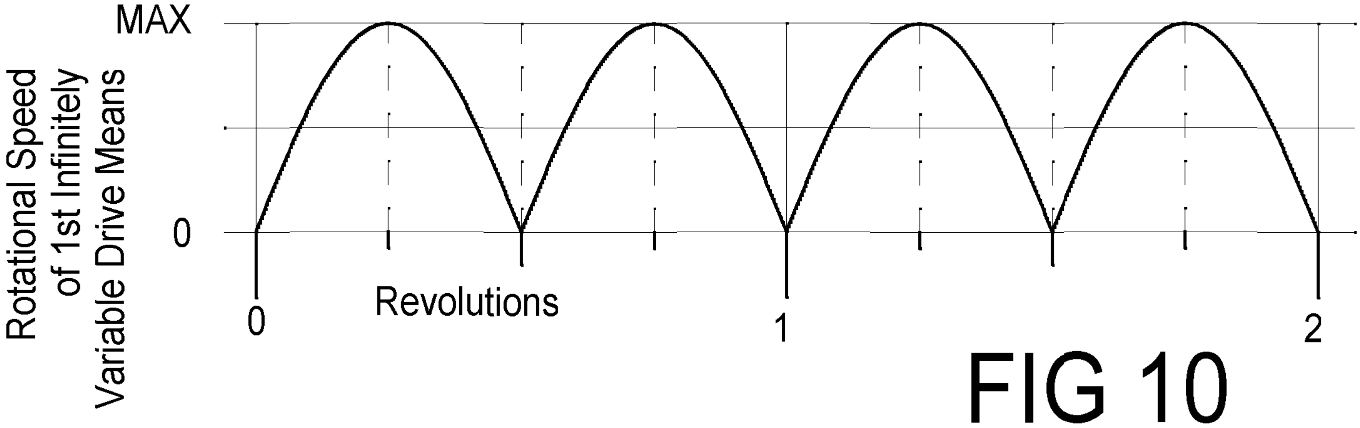


FIG 10

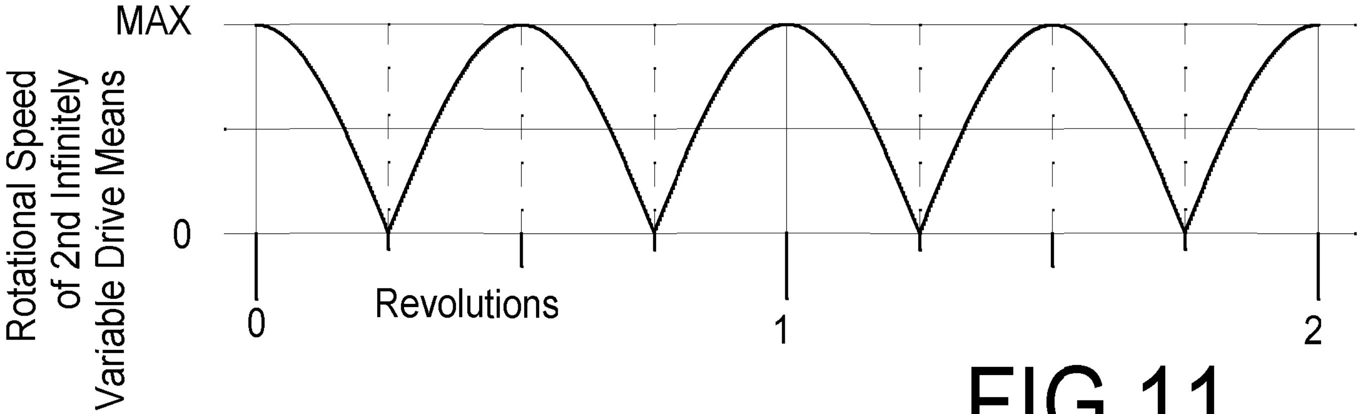


FIG 11

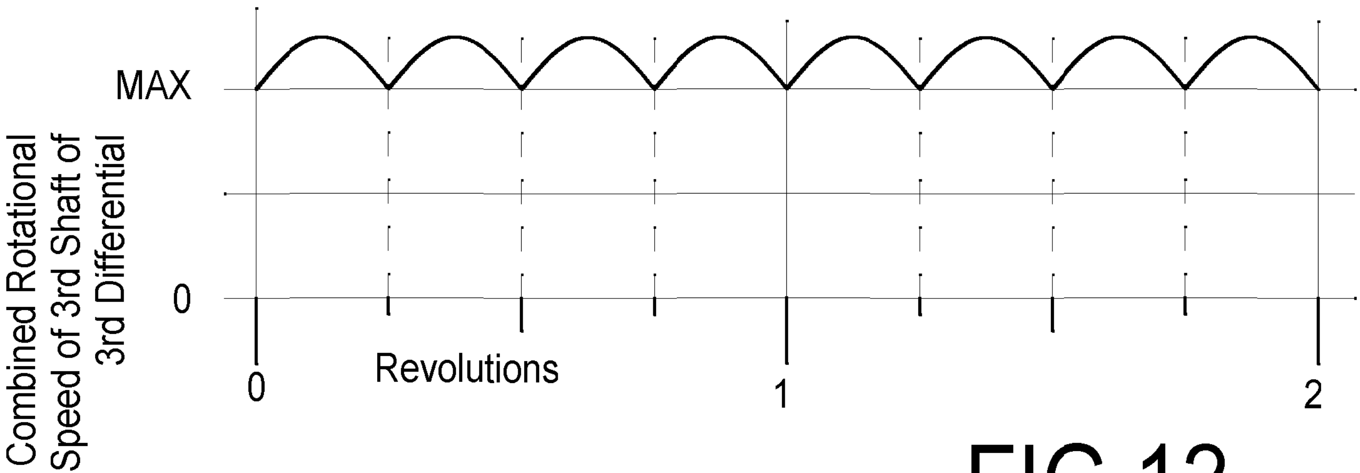


FIG 12

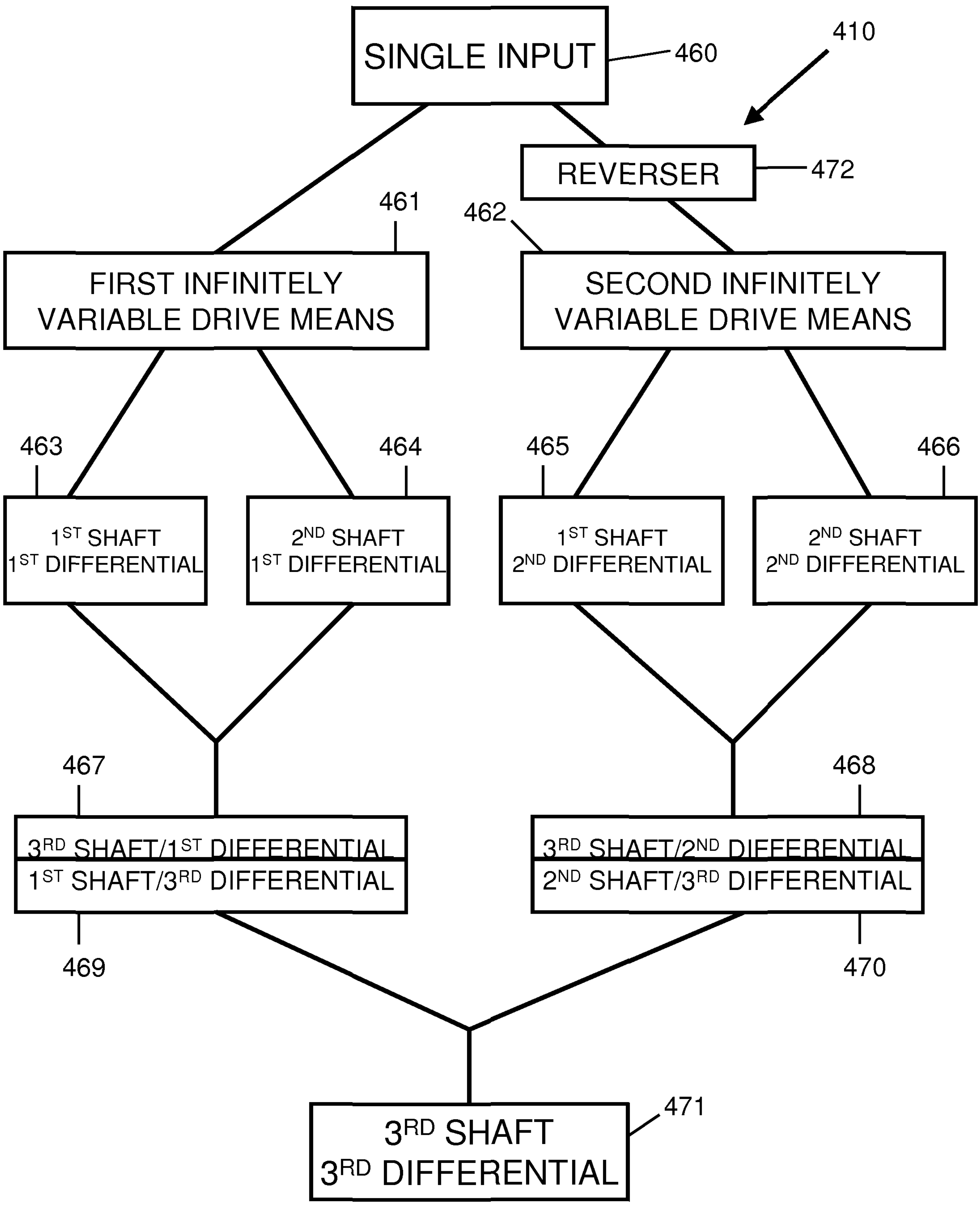

SINGLE INPUT
460
410
REVERSER
472
461
462
FIRST INFINITELY VARIABLE DRIVE MEANS
SECOND INFINITELY VARIABLE DRIVE MEANS
463
464
465
466
1ST SHAFT 1ST DIFFERENTIAL
2ND SHAFT 1ST DIFFERENTIAL
1ST SHAFT 2ND DIFFERENTIAL
2ND SHAFT 2ND DIFFERENTIAL
467
468
3RD SHAFT/1ST DIFFERENTIAL
1ST SHAFT/3RD DIFFERENTIAL
3RD SHAFT/2ND DIFFERENTIAL
2ND SHAFT/3RD DIFFERENTIAL
469
470
3RD SHAFT 3RD DIFFERENTIAL
471

FIG 13

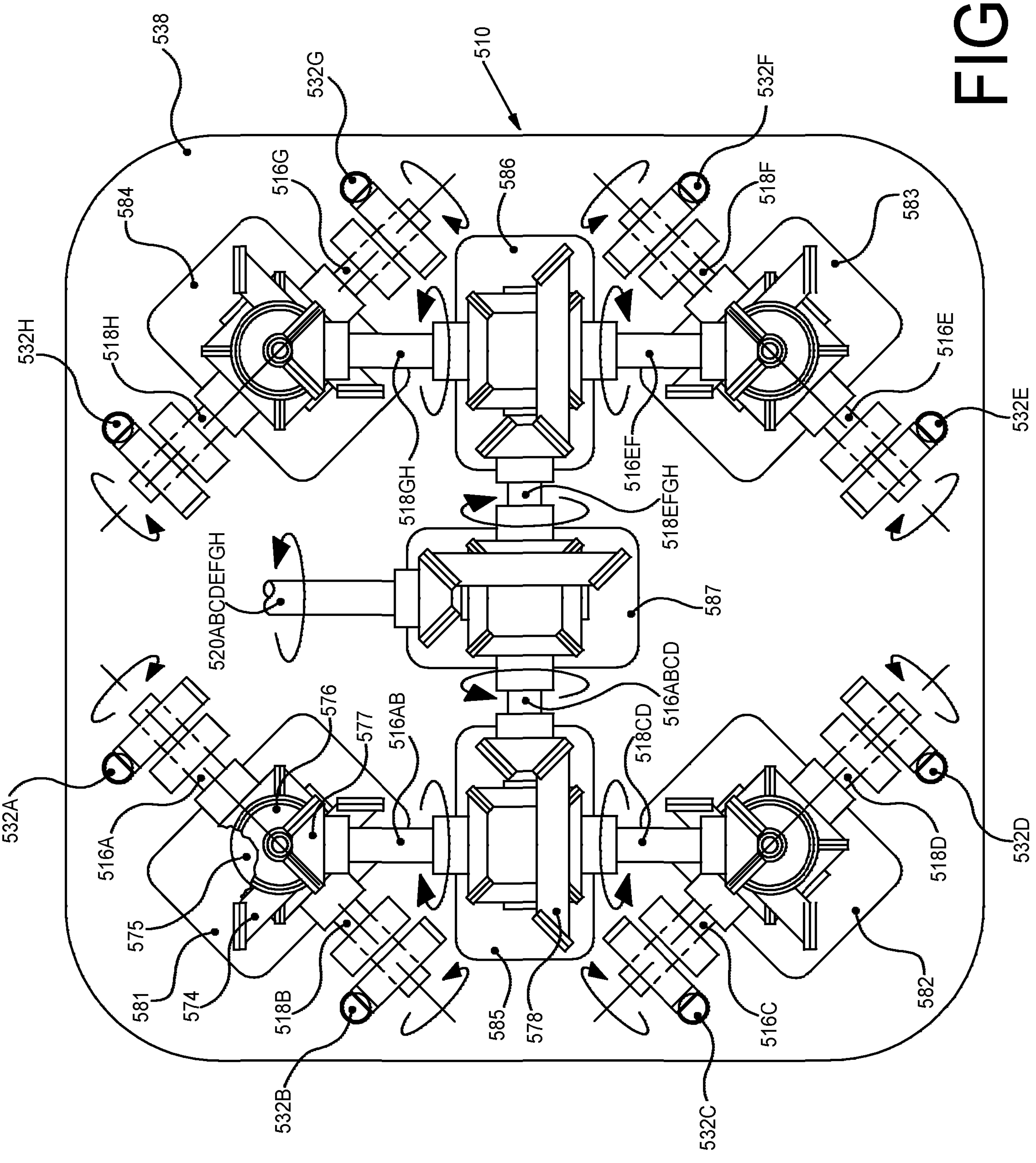

538
510
532G
532F
532H
532E
584
583
586
516G
518F
518H
516E
518GH
516EF
518EFGH
520ABCDEFGH
587
516ABCD
532A
576
577
516AB
518CD
532D
516A
518D
575
581
574
518B
585
578
516C
582
532B
532C

FIG 14

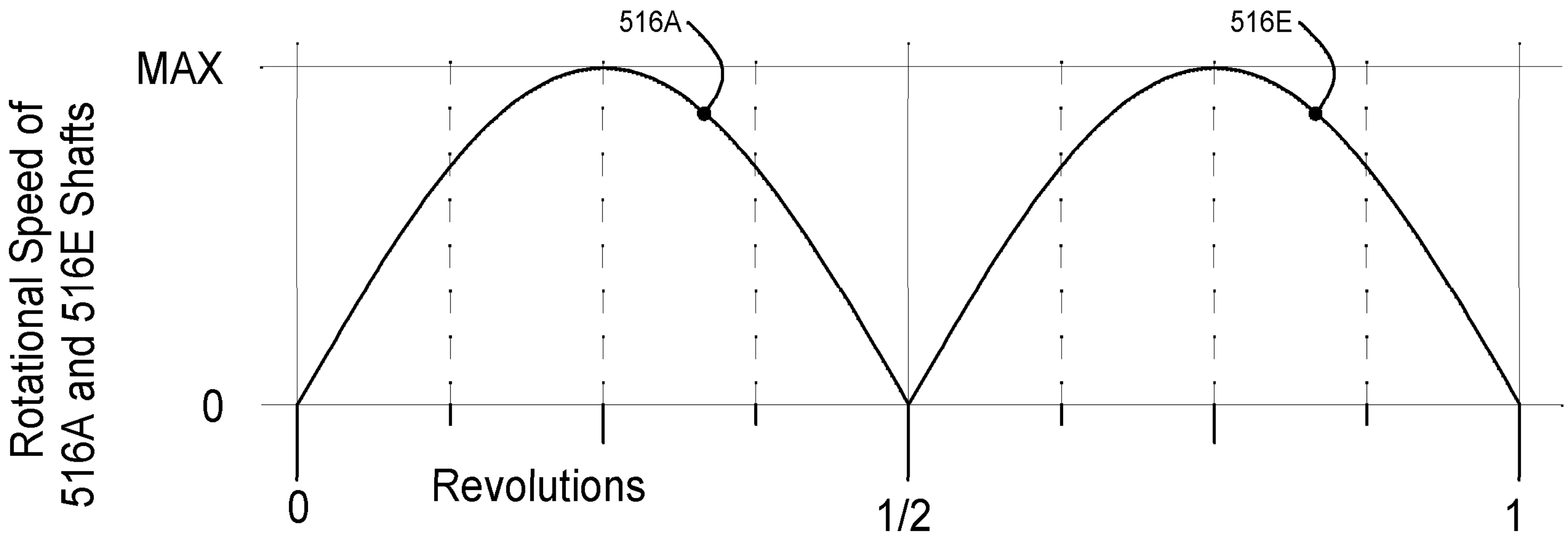

Rotational Speed of
516A and 516E Shafts
MAX
0
516A
516E
Revolutions
0
1/2
1

FIG 15

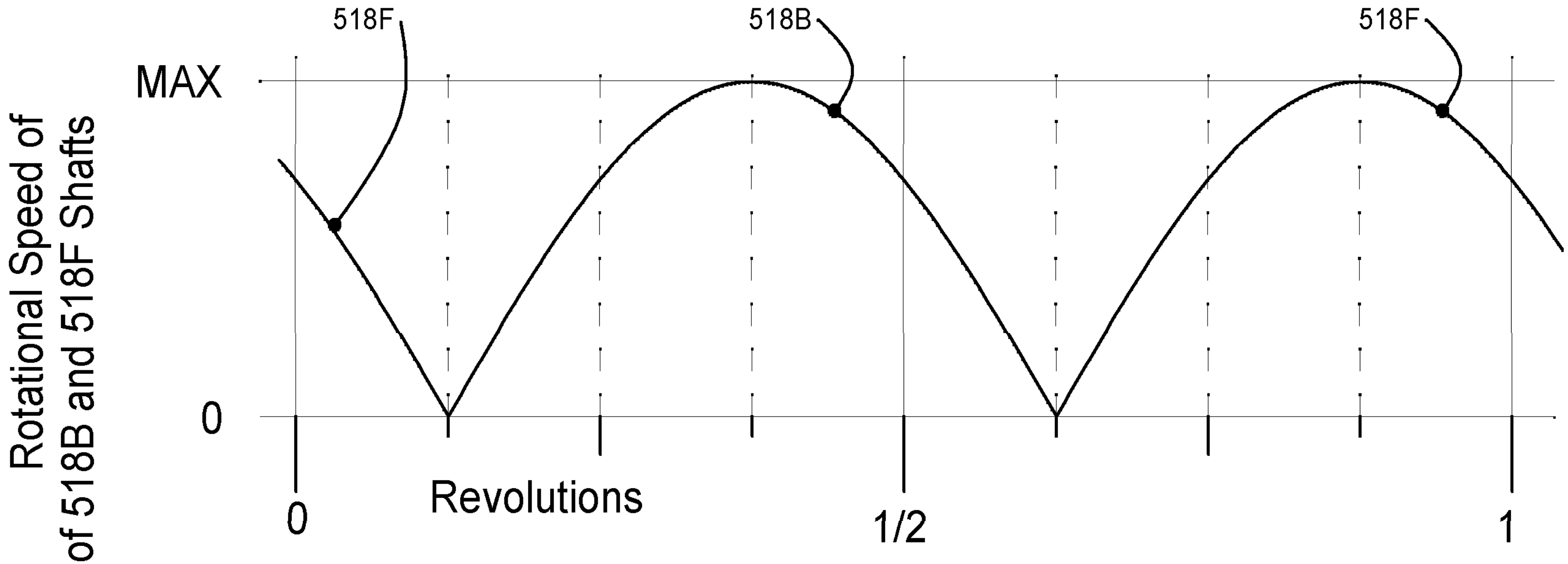

Rotational Speed of
of 518B and 518F Shafts
MAX
0
518F
518B
518F
Revolutions
0
1/2
1

FIG 16

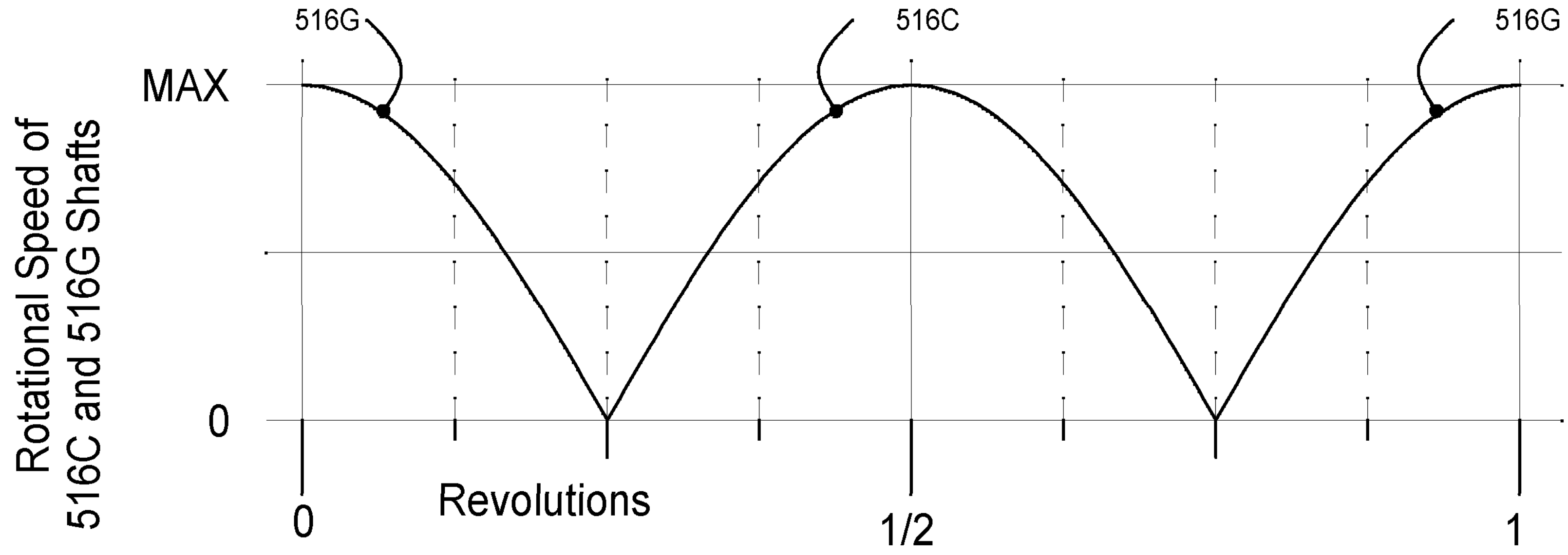

516G
516C
516G
MAX
0
Rotational Speed of
516C and 516G Shafts
Revolutions
0
1/2
1

FIG 17

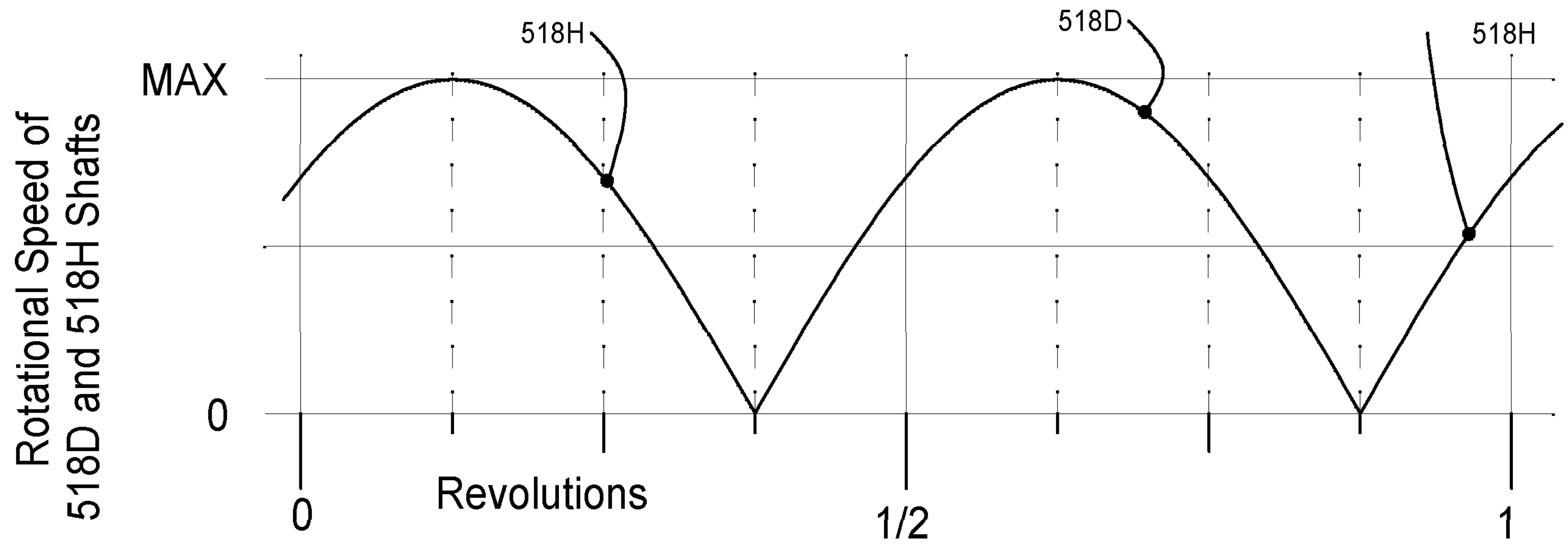

518H
518D
518H
MAX
0
Rotational Speed of
518D and 518H Shafts
Revolutions
0
1/2
1

FIG 18

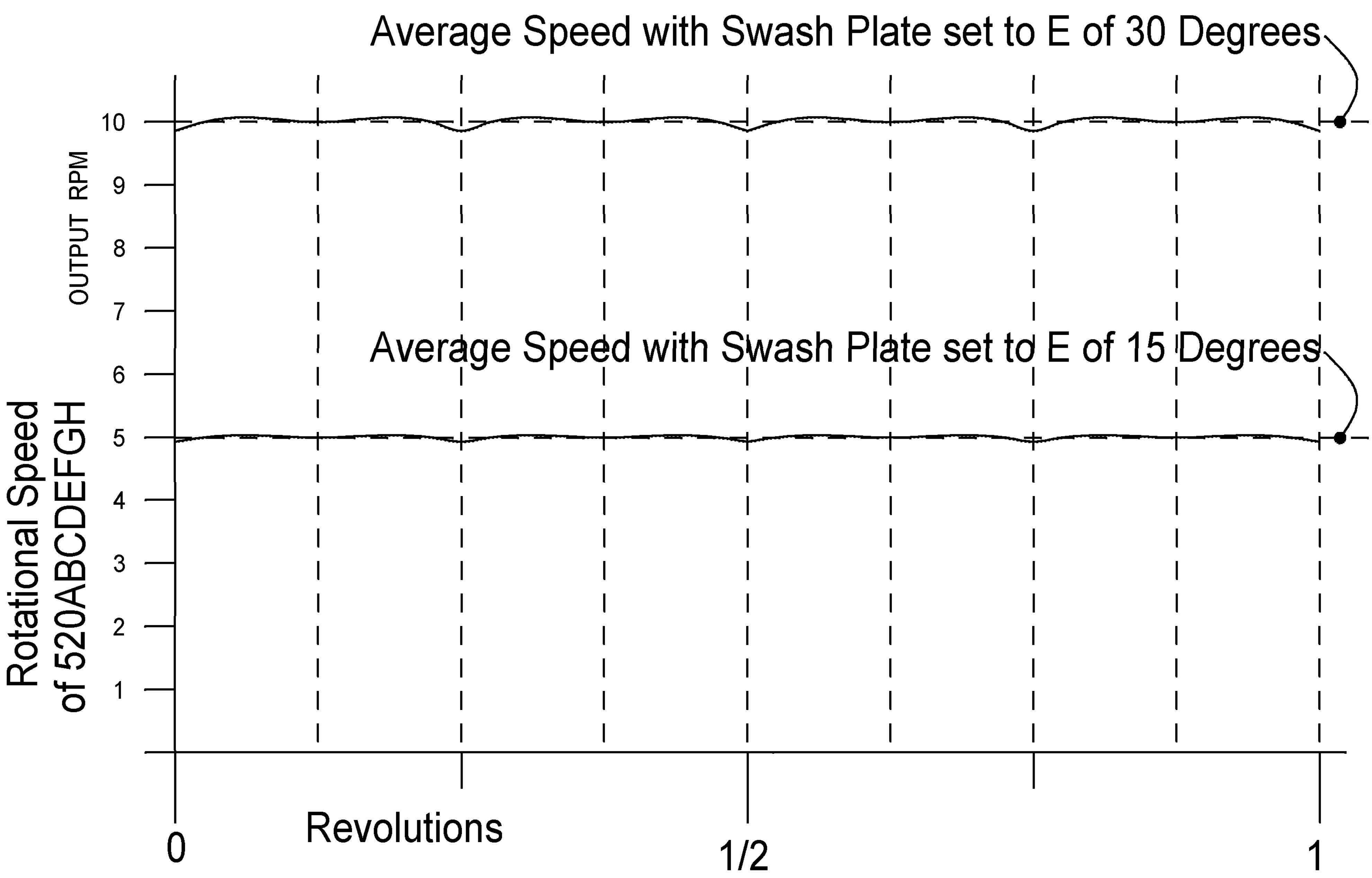

Average Speed with Swash Plate set to E of 30 Degrees
Average Speed with Swash Plate set to E of 15 Degrees
Rotational Speed of 520ABCDEFGH
OUTPUT RPM
10
9
8
7
6
5
4
3
2
1
0
1/2
1
Revolutions

FIG 19

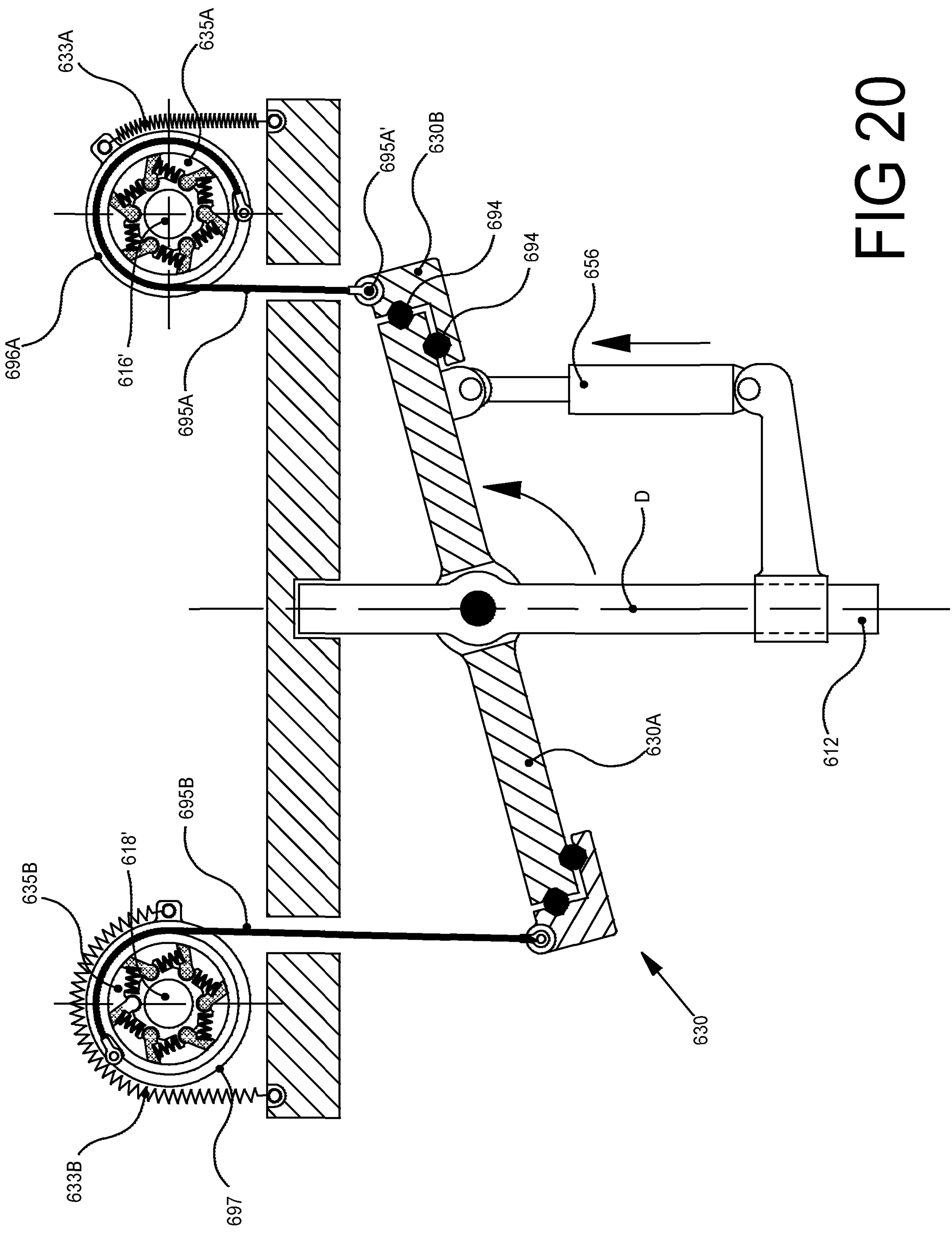

633A
635A
695A'
630B
694
694
656
696A
616'
695A
D
630A
612
635B
618'
695B
630
633B
697

FIG 20

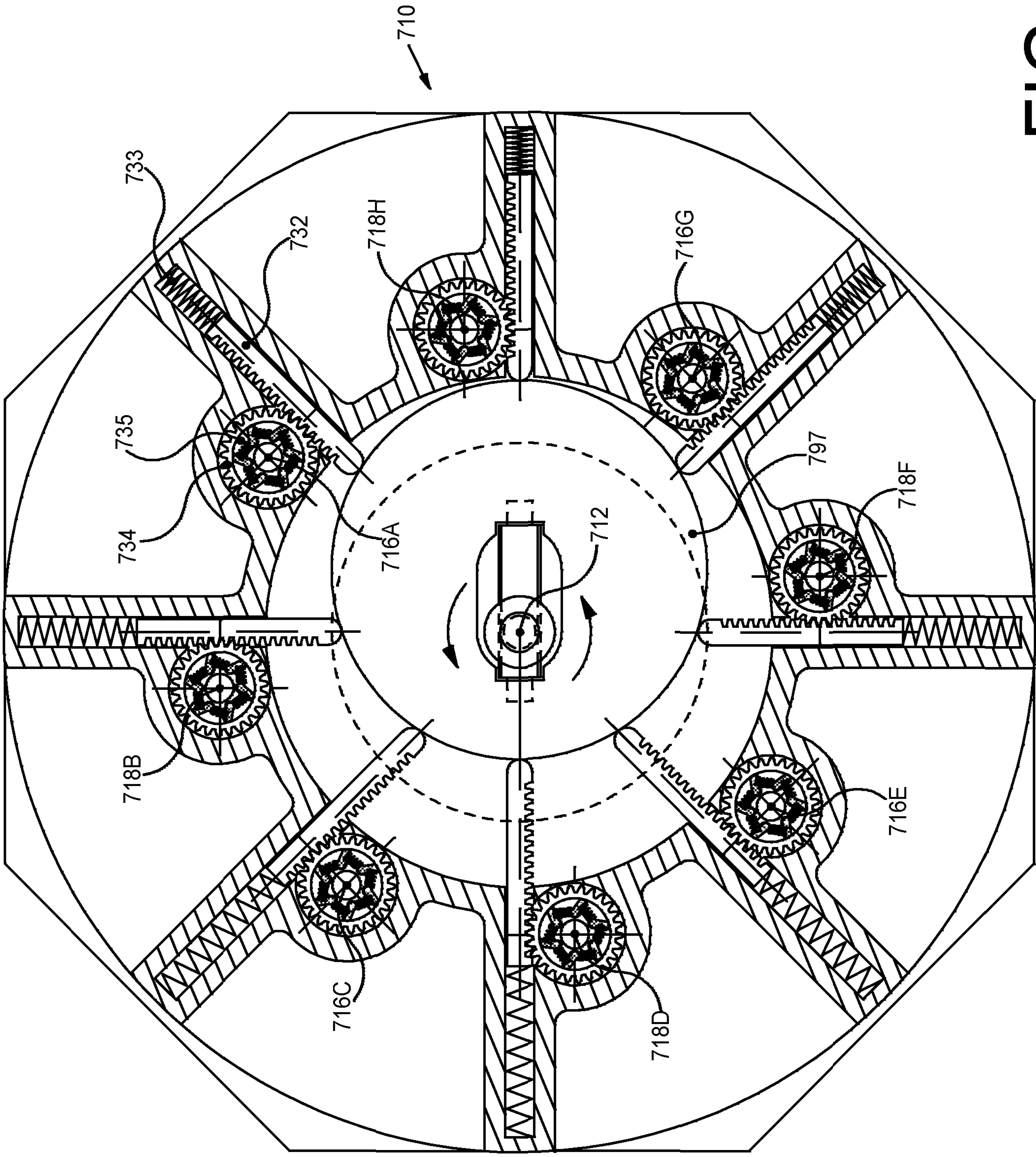
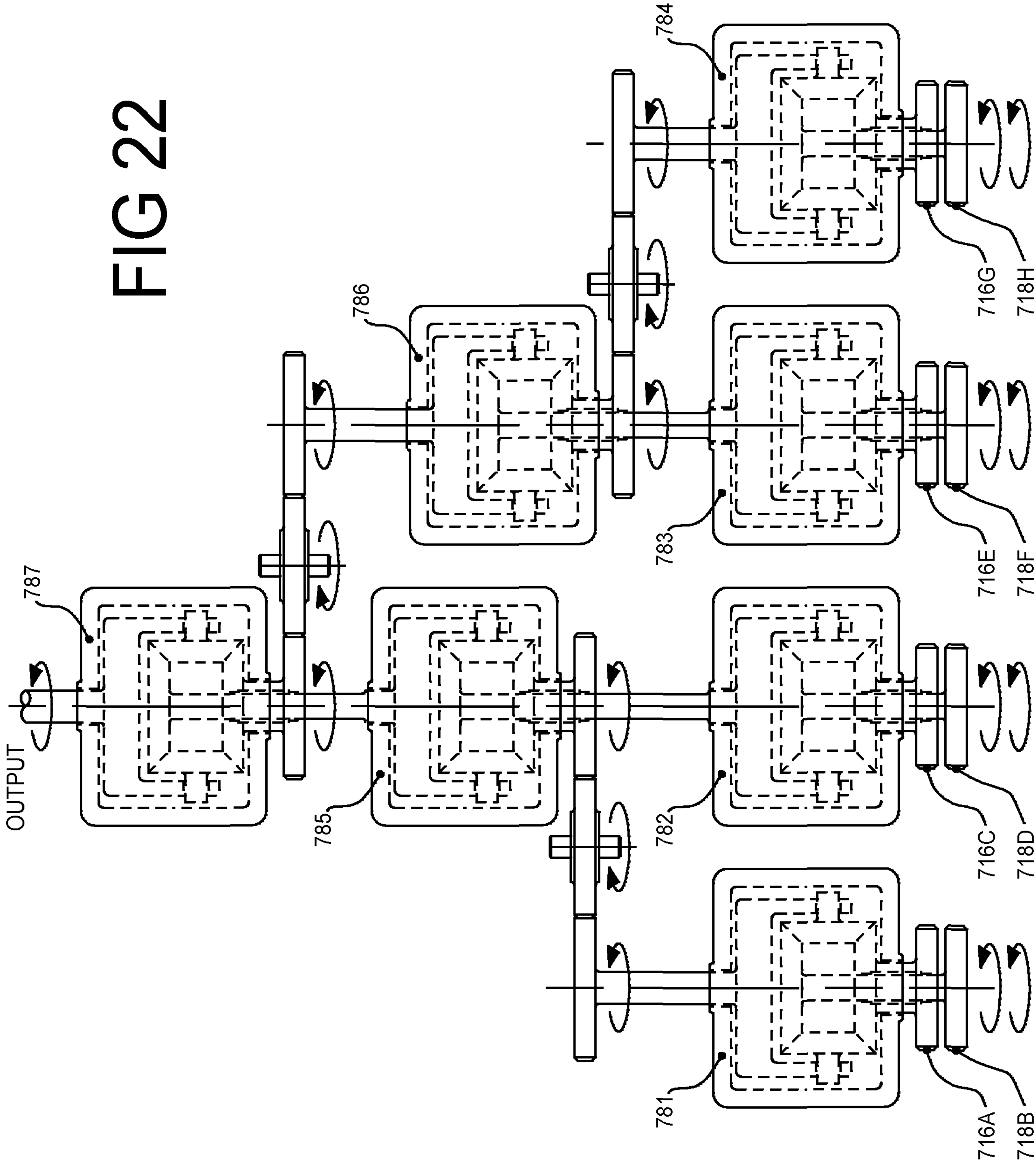
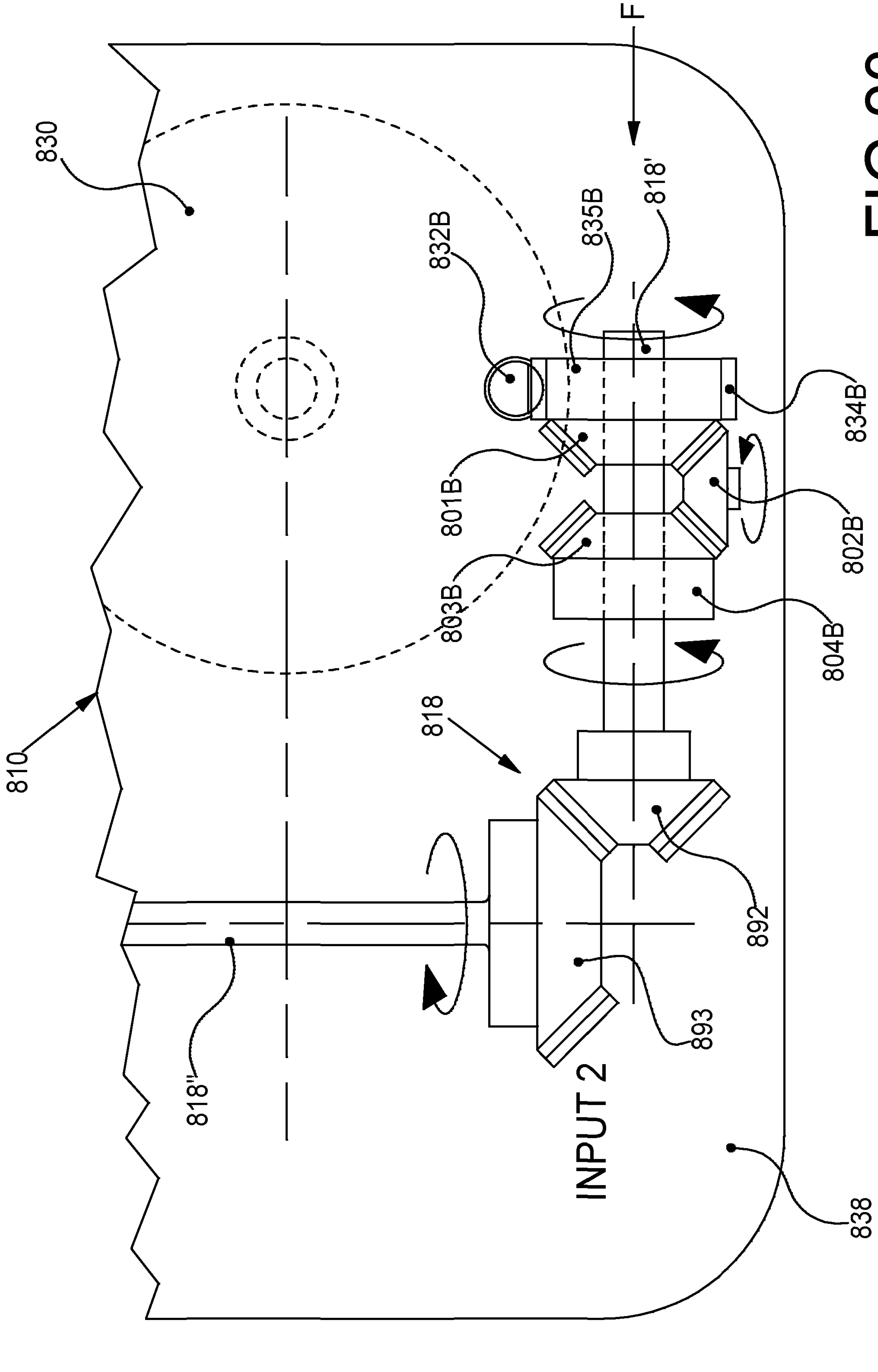

710
733
732
718H
716G
735
734
716A
712
797
718F
718B
716C
718D
716E

FIG 21

FIG 22



OUTPUT
787
785
786
784
783
782
781
716A
718B
716C
718D
716E
718F
716G
718H



F
830
832B
835B
818'
834B
801B
802B
803B
804B
810
818
892
893
818''
INPUT 2
838

FIG 23

Transmission System

The present invention relates to a transmission system, in particular an infinitely variable transmission system. The infinitely variable transmission system may be used as part of a vehicle transmission system, in particular for coupling a prime mover, such as an engine to driven wheels of the vehicle.

Automatic vehicles typically use an internal combustion engine, either a spark ignition (petrol) engine or a compression ignition (diesel) engine. The internal combustion engine can only operate within a certain speed range, the minimum speed being defined by an idle speed and the maximum speed being defined at a speed above which significant engine damage may occur. Within the operating speed range the engine will produce a varying power output, and will also produce a varying torque output. Typically the engines produce maximum power towards the top of the operating speed range and produce maximum torque at a speed slightly less than the maximum power speed.

In order to drive the vehicle at various speeds a transmission system is used to couple the engine to the driven wheels. The speed requirements may be anything between stationary and a maximum forwards speed of the vehicle. Some vehicles (such as automobiles) will preferably additionally include the ability to drive the vehicle in reverse, whereas other vehicles (such as motorbikes) do not require this ability.

Various types of transmission system have been employed:-

One type of transmission system is the use of a friction clutch in conjunction with a gearbox having predefined gear ratios, typically one of which may be a reverse gear ratio. Because the gear ratios are fixed then in order to drive the vehicle at a particular speed it is necessary to vary the speed of the engine. Typically, internal combustion engines operate most efficiently in a narrow band of engine speed and the fixed gear ratio requires the engine to operate outside this optimum band of speed.

A second type of transmission system is the use of a “fluid” clutch coupled to a gearbox having fixed ratios. The fluid clutch, such as a torque converter, allows a degree of slip between the clutch input and the clutch output. Such a system allows the vehicle to creep forward at very low speeds. Typically the changing of transmission speeds will be automated, i.e. such a system is known as an “automatic” gearbox. Clearly, the degree of slip between the input to the clutch and the output from the clutch represents a loss of efficiency.

A further type of transmission is an infinitely variable transmission. One such arrangement is shown in US patent application US2006/0063638. This system uses a V-belt to transfer power from one pulley to another pulley. The drive ratios are varied by the flanges of one pulley being moved closer together with a corresponding separation of the flanges of the other pulley. Such a system relies on friction between the belt and the pulleys and hence is only capable of transmitting relatively low power. In the case of US2006/0063638, the transmission system is used on a relatively low powered vehicle in this case a scooter.

The above mentioned transmission systems have been employed to couple a single prime mover, typically a single internal combustion engine, to driven wheels. Hybrid vehicles include two prime movers, usually an internal combustion engine and an electric motor. It is known to couple the prime movers to the driven wheels via a differential. Thus, the first prime mover is connected to one shaft of a differential, the second prime mover is connected to a second shaft of the differential, and the drive shaft to the driven wheels is connected to third shaft of the differential.

One object of the present invention is to provide an improved variable transmission system.

Another object of the present invention is to provide a variable transmission system capable of transmitting relatively high power.

Thus, according to the present invention there is provided a transmission system having a single input and a differential having a first shaft, a second shaft and a third shaft, the first, second and third shafts being operably coupled together
the single input being configured to drive the first shaft and the second shaft, and being configured to drive the first shaft via a drive means,
the drive means being configured to infinitely vary the drive ratio between said single input and said first shaft so as to provide an infinitely variable overall drive ratio between the single input and the third shaft.

Advantageously, the present invention can be used in a vehicle having a single prime mover such as an internal combustion engine. Power can be transferred to the single input where it is split between the first and second shafts of the differential. The third shaft then recombines the power from the first and second shafts to provide a single output.

The invention will now be described, by way of example only, with reference to the accompanying drawings in which:

Figure 1 is a schematic view of a first embodiment of the present invention,

Figure 2 is a schematic view of a second embodiment of a transmission system according to the present invention,

Figure 3 is a partial view (taken along the line ZZ of figure 5) of the transmission system of figure 2 set at a first drive ratio,

Figure 4 shows the transmission system of figure 3 set at a different drive ratio,

Figure 5 shows a plan view of figure 4,

Figures 6, 7 and 8 show graphs of output speed with time of the first, second and third shafts of figure 2,

Figure 9 shows a further embodiment of a transmission system according to the present invention,

Figures 10, 11 and 12 show graphs of output speed with time of various components of the embodiments shown in figure 9,

Figure 13 shows a further embodiment of a transmission system according to the present invention incorporating forward and reverse ratios,

Figure 14 shows a further embodiment of a transmission system according to the present invention,

Figures 15 to 19 show graphs of output speed with time of various shafts of the embodiment shown in figure 14,

Figure 20 shows a further embodiment of a transmission system according to the present invention,

Figure 21 shows a further embodiment of a transmission system according to the present invention,

Figure 22 shows a schematic view of the differentials used in association with the embodiment shown in figure 21, and

Figure 23 shows a variant of the transmission system of figure 3.

The principle of operation of the present invention is best described with reference to figure 1. The transmission system 10 has a single input 12. Typically the single input will be in the form of a rotating input shaft.

Transmission system 10 also includes a differential 14 having a first shaft 16, a second shaft 18 and a third shaft 20.

The differential can be any type of differential including any type of known differential. A differential is a device, usually but not necessarily employing gears, capable of transmitting torque and rotation through three shafts. The three shafts are connected in such a manner that the speed of the first shaft is a function of the speed of the second and third shafts. Similarly the speed of the second shaft is a function of the speed of the first and third shafts, similarly the speed of the third shaft is a function of the first and second shafts.

Known automobile rear axle differentials include a differential carrier which is rotatable about the axle axis. The differential carrier is typically driven by a crown wheel, which in turn is driven by a pinion mounted on the axle and in turn driven by a drive shaft from the engine. Rotatably mounted on the differential carrier are typically three or more bevel gears having an axis of rotation orthogonal to the axis of rotation of the differential carrier. These bevel gears engage with a gear on a right hand drive shaft and also engage with a gear on a left hand drive shaft. The three shafts are therefore the right hand drive shaft, the left hand drive shaft and the differential carrier. In this case the axis of rotation of all three shafts is the same.

The speed of rotation of the differential carrier is half the sum of the speed of rotation of the right hand drive shaft and the left hand drive shaft, i.e.:-

Differential carrier speed = ½(right hand drive shaft speed + left hand drive shaft speed)

Other types of differential are known including epicyclic differentials wherein the three shafts are defined by a sun gear, an axis of rotation of one or more planet gears, and a ring gear (also known as an annulus gear).

Another type of differential is a spur gear differential as used in an Oldsmobile Tornado. Such a device consists of spur gears only.

As seen in figure 1, the single input 12 drives the first shaft 16 via a variable drive means, in this case an infinitely variable drive means 22. The drive means 22 allows the drive ratio between the single input and the first shaft to be varied infinitely, and in particular to be varied infinitely within a particular range.

The single input also drives the second shaft 18, in this case without an infinitely variable drive means. Thus the speed ratio between the single input and the second shaft 18 is always fixed and cannot be varied.

For the purposes of explanation it is assumed that the single input rotates at a constant speed of 10 rpm. The drive means 22 has a variable ratio of 1:0 and 1:1. The ratio 1:0 means one rotation of the single input shaft 12 causes zero rotations of the first input shaft 16. The ratio 1:1 means one rotation of the single input shaft 12 causes one rotation of the first input shaft 16. It is assumed that the relationship between the rotational speeds of the first shaft 16, second shaft 18 and third shaft 20 is:-

3rd shaft speed = $\frac{1}{2}$*(*1^{st} *shaft speed +* 2^{nd} *shaft speed)*

The transmission system 10 can produce an infinitely variable overall gear ratio between the single input and the third shaft as follows:-

With the drive means 22 set at a ratio of 1:0, with the single input rotating at 10 rpm the first shaft will be stationary and the second shaft will be rotated at 10 rpm. It follows that the third shaft will be rotating at 5 rpm.

However, with the drive means set at a ratio of 1:1, with the single input rotating at 10 rpm, the first shaft will rotate at 10 rpm and the second shaft will rotate at 10 rpm. It therefore follows that the third shaft will rotate at 10 rpm.

Because the drive means is infinitely variable between the ratio of 1:0 to 1:1, the overall ratio between the single input and a third shaft is also infinitely variable, in this case between 10:5 and 10:10.

Figure 2 shows an alternative transmission system 110 which includes the single input 12 and the differential 14 of the first embodiment 10. However, in this case the single input 12 drives the first shaft 16 via drive means 122 and additionally drives the second shaft 18 via drive means 122. The drive means 122 is an infinitely variable drive means which can be set at a ratio of 1:0 or 1:1 or anywhere between these values. In this case when the drive means 122 is set at a ratio of 1:0, with the single input rotating at a constant speed of 10 rpm the first shaft will be stationary and the second shaft will be stationary and therefore the third shaft will also be stationary. However, when the drive means is set at a ratio of 1:1, then with the single input rotating at 10 rpm the first shaft 16 will rotate at 10 rpm and the second shaft 18 will rotate at 10 rpm resulting in the third shaft 20 rotating at 10 rpm. In this example the overall gear ratio can be varied anywhere between 10:0 and 10:10.

With reference to figures 3 to 5 there is shown a third embodiment of a transmission system 210. The principle of operation of transmission system 210 is similar to that of transmission system 110, i.e. a single input drives an infinitely variable drive means which in turn drives both the first shaft and second shaft of the differential.

In this case the single input is in the form of an input shaft 212. The major components of the infinitely variable drive means 222 are swash plate 230, first rack 232A, second rack 232B, first pinion 234A, second pinion 234B, first pinion sprag 235A, second pinion sprag 235B, first rack return spring 233A, second rack return spring 233B, first shaft sprag 236A, second shaft sprag 236B.

A first shaft 216 comprises shaft 216' and shaft 216". Shaft 216' is coupled to shaft 216" by bevel gears 290 and 291.

A second shaft 218 comprises shaft 218' and 218". Shaft 218' is coupled to shaft 218" by bevel gears 292 and 293.

The differential 214 comprises first shaft 216 (in particular shaft 216"), second shaft 218 (in particular shaft 218") and third shaft 220 (in the form of a differential carrier housing (only part of which is shown)).

The transmission system includes a chassis 238 (only part of which is shown). The first rack 232A is slideably mounted in a hole 240 of the chassis 238. The first rack 232A can therefore move in a reciprocal motion, upwardly as shown by arrow A and downwardly as shown by arrow B. An abutment 242A on the bottom of the rack engages the swash plate 230. A further abutment 244A at the top of the rack engages the rack return spring 233A. Another end of the rack return spring 233A engages a part of the chassis 238.

The first rack 232A includes an array of teeth 246.

As best seen in figure 3 first pinion 234A is mounted on the first shaft 216 (in particular shaft 216') of the differential 214. The first pinion sprag 235A allows the pinion to rotate anticlockwise relative to the shaft 216' but prevents the pinion rotating clockwise relative to the shaft 216'.

Thus, as the first rack 232A moves upwardly in the direction of arrow A the teeth 246 drive the pinion 234A in a clockwise direction, thereby driving the shaft 216' in a clockwise direction via the first pinion sprag 235A.

The first shaft sprag 236A (as shown in figure 5) allows the shaft 216' to rotate clockwise relative to the chassis 238 when viewing figure 3 but prevents the first shaft rotating anticlockwise relative to the chassis 238 when viewing figure 3. Thus, when the first rack 232A is descending and moving in the direction of arrow B of figure 3, the first pinion 234A will be rotating in an anticlockwise direction and the shaft 216' will be stationary. It will be appreciated that the pinion sprag 235A will allow the pinion 234A to rotate anticlockwise relative to the stationary shaft 216' and the first shaft sprag 236A will prevent the first shaft from rotating relative to the chassis.

When the first rack 232A is moving upwardly in the direction of arrow A the first pinion 234A will rotate clockwise and the first pinion sprag 235A will ensure that the shaft 216' also rotates clockwise when viewing figure 3. Under these circumstances the first shaft sprag 236A will not prevent this clockwise rotation of the shaft 216'.

As shown in figure 3 the line C drawn at the centre of the swash plate and perpendicular to the swash plate is coincident with the axis of rotation D of the input shaft 212. As the swash plate 230 rotates a circular line on the swash plate rotates under the abutment 242A. Because this circular line is rotating in the horizontal plane, then no vertical movement of the line occurs, and hence no vertical movement of the rack 232A occurs.

As shown in figure 4 the swash plate has been rotated (by actuator 256) such that the line C is now at an angle to the axis of rotation of the input shaft 212 (in this case 15°). Note that the swash plate 230 is nevertheless rotates about axis D. Under these circumstances the point of contact between abutment 242A and the top surface 248 of the swash plate 230 rises and falls with each rotation of the input shaft 212. As shown in figure 4 the rack is in its most raised position for this particular angle of the swash plate. Once the input shaft 212 has rotated a further 180 degrees, the first rack 232A will be at its most lowered position for this angle of swash plate.

As the input shaft 212 rotates the first rack 232A will reciprocate upwardly and downwardly, i.e. the first rack will be driven upwardly by the swash plate and then be driven downwardly via the first spring 233A. This reciprocal motion of the first rack will be sinusoidal.

Because the first pinion 234A is in permanent engagement with the array of teeth on the first rack, then the first pinion will alternately by driven in a clockwise and an anticlockwise direction.

However, because of the first pinion sprag 235 and the first shaft sprag 236 the shaft 216' will not rotate in alternate directions, rather it will be driven in a clockwise direction whilst the rack 232A is moving in the direction of arrow A but will remain stationary when the rack 232A is moving in the direction of arrow B. The system therefore causes an intermittent rotation of the first shaft 216. Thus, shaft 216' only ever rotates in a clockwise direction when viewing figure 3 and shaft 216" only ever rotates in an anticlockwise direction when viewed in a direction of arrow F of figure 5.

Movement of the second rack 232B, the second rack return spring 233B, the second pinion 234B, the second pinion sprag 235B, and the second shaft sprag 236B is similar to their corresponding first components as described above. However, it will be appreciated that the movement is 180° out of phase. Thus, as will be appreciated from figure 4, as the first rack 232A is moving upwardly in the direction of arrow A, the second rack 232B is moving downwardly in the direction arrow B and vice versa.

It will be appreciated that the system therefore causes an intermittent rotation of the second shaft 218. Thus, shaft 218' will only ever rotate in an anticlockwise direction when viewing figure 3 and shaft 218" will only ever rotate in an anticlockwise direction when viewed in the direction of arrow F of figure 5.

As will be appreciated, both shafts 216" and 218" only ever rotate in an anticlockwise direction when viewed in the direction of arrow F of figure 5.

Figures 6, 7 and 8 show the rotational speed of the first, second and third shafts with time, with the swash plate set at angle E and being rotated at a constant speed.

As will be appreciated from figure 8, a constant input shaft speed will produce a varying speed of the third shaft. There is shown on figure 8 the average speed of the third shaft. Clearly, by increasing the swash plate angle the average speed of the third shaft will increase and by decreasing the swash plate angle the average speed of the third shaft will decrease. By decreasing the swash plate angle to zero the output speed of the third shaft will become zero.

As best shown in figure 5, the differential 214 is analogous to a conventional automotive rear axle differential. Thus, the differential includes the differential carrier 220 (which equates to the third shaft of the present invention). The first shaft 216 is coupled to the first shaft pinion 217, the second shaft 218 (in particular the shaft 218') is coupled to the second shaft pinion 219. The first shaft pinion and second shaft pinion are rotatably mounted in the differential carrier 220 and engage to bevel gears 250 and 251. The bevel gears 250 and 251 are rotatably mounted in the differential carrier 220. A differential housing 253 is attached to the chassis 238 and includes bearings to allow the first, second and third shafts to rotate. In this manner the first shaft, second shaft and third shaft (differential carrier) are coupled by the first shaft pinion 217, second shaft pinion 219 and pinion gears 250 and 251.

As best seen in figure 8, for a constant rotational speed of the input shaft 212 it is possible to produce an infinitely variable average speed of the third shaft 220. However, the rotational speed of the third shaft is not constant, rather it varies between a minimum and a maximum. In this case the minimum is 0 rpm.

With reference to figure 9 there is shown a further embodiment of a transmission system 310 according to the present invention in which the output is relatively smooth. Thus, the single input 360 drives a first infinitely variable drive means 361 and a second infinitely variable drive means 362. The first infinitely variable drive means 361 drives the first shaft 363 of the first differential and the second shaft 364 of the first differential. The second infinitely variable drive means 362 drives the first shaft 365 of the second differential and also drives the second shaft 366 of the second differential. The first shaft 363 and second shaft 364 drive the third shaft 367 of the third differential. The first shaft 365 and second shaft 366 drive the third shaft 368 of the second differential. The third shaft 367 of the first differential acts as the first shaft 369 of a third differential. The third shaft 368 of the second differential acts as the second shaft 370 of the third differential. The first shaft 369 and second shaft 370 drive the third shaft 371 of the third differential.

The single input 360, first infinitely variable drive means 361, first shaft 363, second shaft 364 and third shaft 367 operate in a manner similar to the equivalent component shown in figure 2. The single input 360, second infinitely variable drive means 362, first shaft 365, second shaft 366 and third shaft 368 operate in a manner similar to the equivalent components shown in figure 2. The first shaft 369, second shaft 370 and third shaft 371 operate in a manner similar to the equivalent components shown in figure 2.

Figures 10, 11 and 12 show the rotational speed of the third shaft 367, third shaft 368 and third shaft 371 for a particular swash plate angle (greater than zero) of the first infinitely variable drive means 361 and second infinitely variable drive means 362. As will be appreciated the output from the third shaft 367 out of phase with the output from the third shaft 368. In this way the third shaft 371 rotates at a relatively constant speed, and in particular for a particular rotational speed of the single input 360, and for a particular swash plate angle greater than zero of the first infinitely variable drive 361 and the second infinitely variable drive means 362, the rotational speed of the third shaft 371 never falls to zero.

As will be appreciated, when the single input is rotated in a particular direction (by way of example in a clockwise direction). The direction of rotation of the third shaft will always be in the same direction. In order to reverse the direction of the third shaft it is necessary to reverse the direction of the input.

Figure 13 shows a variant 410 of the transmission system 310. The only difference between transmission system 410 and transmission system 310 is that transmission system 410 has a reverser 472. The reverser can be any type of mechanism (such as a set of gears) that reverses the input being fed to the second infinitely variable drive means 62 from the single input. Components of transmission system 410 which equate to components of transmission system 310 are labelled 100 greater.

For ease of explanation operation of the device will be described in terms of "forwards" and "reverse" directions. Thus, the single input drives the first infinitely variable drive means 461 in a forwards direction and drives the second infinitely variable drive means 462 in a reverse direction (via reverser 472). In order to drive the third shaft 471 in a forwards direction the swash plate on the first infinitely variable drive means 461 is set to an angle greater than zero whereas the swash plate on the second infinitely variable drive means is set to zero. This will result in third shaft 467 rotating in a forwards direction and third shaft 468 being stationary thereby causing third shaft 471 to rotate in a forwards direction. However, by setting the swash plate of the first infinitely variable drive means 461 to a zero angle and by setting the swash plate of the second infinitely variable drive means 462 to an angle greater than zero causes the third shaft 467 to be stationary and the third shaft 468 to rotate in a reverse direction, thereby causing the third shaft 471 to rotate in a reverse direction.

In this manner the transmission system 410 is infinitely variable between a forwards ratio and a reverse ratio.

Figure 14 shows a further embodiment of a transmission system 510. In this case there are eight racks, 532A, B, C, D, E, F, G and H. All racks are operated by a single swash plate such as is shown in figure 3. Each rack has an associated rack return spring, pinion, pinion sprag arrangement and shaft sprag arrangement, similar to those shown in figure 3.

Rack 532A therefore drives shaft 516A with an intermittent motion. Rack 532B drives shaft 518B with an intermittent motion. The motion of shafts 516A and 518B are combined in differential 581 and output to shaft 516AB via crown wheel 574, lower bevel 575, upper bevel 576 and bevel 577. The motions of shafts 516C and

518D are combined in differential 582 and output to shaft 518CD in a manner similar to that described above for shaft 516AB. Similarly the motions of shafts 516E and 518F are combined by differential 583 and output to shaft 516EF. Similarly the motions of shafts 516G and 518H are combined by differential 584 and output to shaft 518GH.

Motions of shafts 516AB and 518CD are combined in differential 585 and output to shaft 516ABCD via crown wheel 578. Similarly the motions of shafts 516EF and 518GH are combined in differential 586 and output to shaft 518EFGH.

The motions of shafts 516ABCD and 518EFGH are combined in differential 587 and output to shaft 520ABCDEFGH.

Figures 15, 16, 17 and 18 show the output speed of shafts 516A, 518B, 516C, 518D, 516E, 518F, 516G and 518H for a certain swash plate angle. Figure 19 shows the output speed of shaft 520ABCDEFGH at a swash plate angle of 15 and 30 degrees. As will be appreciated, the variation in speed from the average speed for shaft 520ABCDEFGH is far less than shaft 220, i.e. the output speed is smoother.

Figure 20 shows a further embodiment of a transmission system 610 in which components which fulfil the same function as those of transmission system 210 are labelled 400 greater. In this case the swash plate 630 is a two-piece swash plate 630A and 630B. Piece 630B is mounted on bearings 694 such that when part 630A rotates with shaft 612 about axis D part 630B does not rotate about axis D. The racks of transmission system 210 have been replaced by cables 695A and 695B. One end 695A' of cable 695A is attached to part 630B and an opposite end is attached to pulley wheel 696A. As shown in figure 20, pulley wheel 696A has been rotated in a clockwise direction via spring 633A. As part 630A of swash plate 630 rotates, end 695A' of cable 695A will descend thereby rotating pulley wheel 696A in an anticlockwise direction. Continued rotation of part 630A of swash plate 630 will cause the end 695A' to ascend thereby allowing spring 633A to pull pulley wheel 696A in a clockwise direction. Shaft 616' is thereby caused to rotate with an intermittent motion similar to that of shaft 216' (note transmission system 610 includes a shaft sprag arrangement similar to shaft sprag arrangement 236 of transmission system 210).

Similarly shaft 618' is caused to rotate with an intermittent motion similar to that of shaft 218'.

Figure 21 shows a further embodiment of a transmission system 710. In this case there are eight racks 732, eight rack return springs 733, eight pinions 734, eight pinion sprag arrangements 735, eight shaft sprag arrangements (not shown). The eight racks 732 are orientated radially relative to the input shaft 712. A cam 797 is driven by shaft 712 and sequentially drives each rack 732 radially outwardly. Each rack is then returned inwardly via its associated rack return spring 733. Thus, each of shafts 716A, 718B, 716C, 718D, 716E, 718F, 716G, 718H are driven with intermittent motion and are combined to form a single output. Figure 22 shows a schematic view of how the various shafts of figure 21 are combined via differentials 781, 782, 783, 784, 785, 786 and 787 to provide a single output in a manner similar to that of the differentials of figure 14.

The cam can be offset relative to the axis of rotation of the input shaft 712. As shown in figure 21 the cam is shown in its most offset direction. The chain dotted line concentric with shaft 712 shows the cam in a position of least offset. In this position none of the racks move as the cam rotates.

With reference to figure 23 there is shown a variant 810 of the transmission system 210 in which components that perform the same function are labelled 600 greater.

Figure 23 is a part plan view of the transmission system 810, similar to the plan view of the transmission system 210 shown in figure 5.

Swash plates 830, pinion 834B and associated pinion sprag arrangement 835B, bevel gear 892, and bevel gear 893 are all identical to equivalent components in the transmission system 210.

However the shaft sprag arrangement 236B of transmission system 210 has been replaced with bevel gears 801B, 802B, 803B and bevel gear sprag 804B.

Bevel gear 801B is fixed to pinion 834B and hence rotates about the axis of shaft 818'. Bevel gear 802B is mounted in a bearing (not shown) fixed to the chassis 838. The bevel gear 802B engages both bevel gears 801B and 803B.

Bevel gear 803B is rotatably mounted on shaft 818' and is coupled to bevel gear sprag 804B. Thus bevel gear 803B and bevel gear sprag 804 are rotatable about the axis of shaft 881'.

When viewing in the direction of arrow F, the pinion sprag arrangement 835B is capable of driving shaft 818' in an anti-clockwise direction, but will disengage and freewheel when the pinion 834B rotates in a clockwise direction relative to shaft 818'. Similarly, the bevel gear sprag 804 is arranged so that bevel gear 803B can drive shaft 818' in an anti-clockwise direction when viewed in the direction of arrow F but will disengage and freewheel when the bevel gear 803B rotates in a clockwise direction relative to shaft 818'.

It will be appreciated that because gears 801B and 803B are connected via an intermediate gear, namely bevel gear 802B, bevel gears 801B and 803B will always rotate in an opposite direction. Thus, when rack 832B is being lifted by the swash plate 803 (i.e. rack 832B is coming towards the viewer when considering figure 23) the pinion 834B and bevel gear 801B will both be rotating in an anti-clockwise direction when viewed in the direction of arrow F. This anti-clockwise rotation will cause the pinion sprag arrangement 835B to engage and drive shaft 818' in an anti-clockwise direction. Whilst this is occurring the bevel gear 803B will be rotating in an clockwise direction when viewed in the direction of arrow F and the bevel gear sprag 804B will be disengaged and freewheeling.

When the rack 832B is descending (i.e. moving away from the viewer when considering figure 23) the pinion 834B will be rotating in a clockwise direction and hence the pinion sprag arrangement 835B will be disengaged and freewheeling.

Meanwhile, the bevel gear 803B will be rotating in an anti-clockwise direction and bevel gear sprag 804B will be engaged, thereby allowing bevel gear 803B to drive shaft 818' in an anti-clockwise direction when viewed in the direction of arrow F.

Such an arrangement will cause shaft 818' to rotate in an anti-clockwise direction when the rack is both ascending and descending. This can be contrasted with the arrangement of transmission system 210 wherein shaft 218' only rotates when the rack 232B is ascending and shaft 218' is held stationary when the rack 232B is descending.

As will be appreciated by comparing figures 23 and figure 5 the transmission system 810 has a similar arrangement of bevel gears and bevel gear sprags driving the first input. The first and second input of transmission system 810 are combined via a differential identical to differential 214.

As will be appreciated, as the rack 832B is ascending, it will be positively moved by the swash plate. As the rack 832B is descending, it is moved by a rack return spring similar to those shown in Figure 3. However, the rack return spring must be sufficiently powerful to ensure that the rack abutment remains in engagement with the swash plate as the rack descends. Depending upon the power to be transmitted, the rack return spring of transmission system 810 may be required to be a more powerful spring than, for example, the rack return spring 233A shown in Figure 3. In an alternative embodiment, the rack could include a 'hook' feature which hooks around the swash plate such that the rack is driven to ascend in a manner similar to that of rack 232A, but is driven to descend by the hook feature engaging the underside of the swash plate. In these circumstances, a rack return spring can be dispensed with since the rack is positively displaced both upwardly and downwardly by engagement with the swash plate.

The embodiment shown in figure 1 uses a single differential. The embodiment shown in figure 9 uses three differentials. The embodiment shown in figure 14 uses seven differentials. In further embodiments any number of differentials could be used. An example of a system having two differentials would be as per figure 9 wherein the second differential was replaced by a simple drive connection connecting the second infinitely variable drive means to the second shaft of the third differential. As shown in figure 21, eight racks provide intermittent motion to the differentials 781, 782, 783 and 784 of figure 22. In further embodiments any one or more of the racks of figure 21 could be deleted with the corresponding input to the associated differential 781, 782, 783 or 784 being held stationary. For example the rack associated with shaft 718H could be deleted and shaft 718 could be fixed relative to the chassis.

The sprag arrangements described above are a particular form of one way clutch. In this case sprag arrangement 235A consists of six pawls 224 pivotally mounted relative to shaft 216'. Springs 225 bias associated pawls 224 in a clockwise direction such that an outer surface 226 of the pawl 224 frictionally engages an inner surface 227 of the pinion 234. Drive is transmitted by static friction between the pawl and the inner surface 227. In further embodiments any form of one way clutch could be used.

One form of one way clutch could utilise pawls similar to pawls 224 having a serrated surface which engages a serrated inner surface of a pinion similar to pinion 234. In such an arrangement there is a limited amount of back lash between the pawl and the inner surface of the pinion. Because of this the drive system using a swatch plate is not strictly infinitely variable rather it is variable via a large number of discreet ratios. Thus, if there are 140 serrations on the inner surface of the pinion, and at full swash plate travel the rack rotates the pinion through one complete revolution, then the ratio is a variable through 140 discreet steps. If the rack rotates the pinion through two revolutions at full swash plate angle, then there are 280 discreet steps.

As shown in figure 3 an actuator 256 operates to move the swash plate between its various positions, one of which is shown in figure 3 and one of which is shown in figure 4. Any type of actuator can be used.

As shown in figure 1, power is transferred from the single input 12 to the third shaft 20. However, when the transmission system of figure 1 is incorporated into a vehicle and “engine braking” is used to slow the vehicle, then power is transmitted in a reverse direction from the third shaft 20 to the single input 12. It will be appreciated that in any of the above mentioned transmission systems power can be transmitted in either direction.

It will be appreciated that the drive means described above (whether infinitely variable, or whether variable over a discreet number of steps), the ratio can be changed whilst the power is being transmitted. This can be contrasted with the prior art transmission system having a friction clutch used on conjunction with a gearbox having predefined gear ratios, wherein power must be interrupted as the gear ratios are changed.

The profile of cam 797 is circular, and as such the motion of the shafts associated with each pinion is similar to that shown in figure 6. However, by having a non circular cam the speed profile of the shafts 716A, 718B, 716C, 718D, 716E, 718F, 716G, 718H can be varied. In particular the peak rotational speed of the shafts could be limited by putting a "flat" on the otherwise circular profile of the cam. The "flat" would coincide with point of maximum rotational speed of the shaft. Such a system would provide a smoother drive since the maximum peak speed of individual shafts would be reduced.

Just as it is possible to use a non circular cam in place of cam 797, it is also possible to use a non flat swash plate in place of any of the swash plates herein before described. The profile of the swash plate would be such as to reduce the peak rotational speed of the shafts.

As such, using non circular cams or non flat swash plates would provide for non sinusoidal motion of the racks in order to smooth the output from the transmission system.

Claims

1. A transmission system having a single input and a differential having a first shaft, a second shaft and a third shaft, the first, second and third shafts being operably coupled together
the single input being configured to drive the first shaft and the second shaft,
and being configured to drive the first shaft via a drive means,
the drive means being configured to vary the drive ratio between said single input and said first shaft so as to provide a variable overall drive ratio between the single input and the third shaft.

2. A transmission system as defined in claim 1 wherein the single input is configured to drive both the first shaft and the second shaft via a said drive means.

3. A transmission system as defined in claim 2 wherein said drive means is configured to vary the drive ratio between said single input and said first shaft and is configured to vary the drive ratio between said single input and said second shaft.

4. A transmission system as defined in any preceding claim wherein the variable overall drive ratio between the single input and the third shaft includes the ration 1:0.

5. A transmission system as defined in any preceding claim wherein the variable overall drive ratio between the single input and the third shaft includes a forwards ratio and a reverse ratio.

6. A transmission system as defined in any preceding claim in which during operation of the transmission system at a selected overall drive ratio with the single input being driven at a constant speed, the first shaft is driven at a variable

speed having a maximum instantaneous first shaft speed and a minimum instantaneous first shaft speed.

7. A transmission system as defined in claim 6 wherein said maximum instantaneous first shaft speed is 10% or more faster than the said minimum instantaneous first shaft speed, preferably 20% or more faster than said minimum instantaneous first shaft speed.

8. A transmission system as defined in claim 6 or 7 wherein said minimum instantaneous first shaft speed is 0 RPM.

9. A transmission system as defined in any one of claims 6 to 8 wherein the second shaft is driven at a variable speed having a maximum instantaneous second shaft speed and a minimum instantaneous second shaft speed.

10. A transmission system as defined in claim 9 wherein said maximum instantaneous second shaft speed is 10% or more faster than said minimum instantaneous second shaft speed, preferably 20% or more faster than said minimum instantaneous second shaft speed.

11. A transmission system as defined in claim 9 or 10 wherein said minimum instantaneous second shaft speed is 0 rpm.

12. A transmission system as defined in any one of claims 9 to 11 in which said maximum instantaneous first shaft speed occurs at a different time to said maximum instantaneous second shaft speed.

13. A transmission system as defined in any one of claims 9 to 12 wherein said minimum instantaneous first shaft speed occurs at a different time to said minimum instantaneous second shaft speed.

14. A transmission system as defined in any preceding claim wherein said differential is a first differential, the transmission system further including a

second differential having a first shaft, a second shaft and a third shaft, the first, second and third shafts being operably coupled together, the third shaft of the first differential being configured to drive the first shaft of the second differential, the single input being configured to drive the second shaft of the second differential so as to provide a variable overall drive ratio between the single input and the third shaft of the second differential.

15. A transmission system as defined in claim 14 further including a third differential having a first shaft, a second shaft, and a third shaft, the first, second and third shafts of the third differential being operably coupled together, the single input being configured to drive the first shaft of the third differential and the second shaft of the third differential, the third shaft of the third differential being configured to drive the second shaft of the second differential.

16. A transmission system having a single input, a first, second, third, fourth, fifth, sixth and seventh differential, the single input being configured to drive:-
the first differential via a first and second variable drive means,
the second differential via a third and fourth variable drive means,
the third differential via a fifth and sixth variable drive means,
the fourth differential via a seventh and eighth variable drive means;
the first and second differential being configured to drive the fifth differential;
the third and fourth differential being configured to drive the sixth differential; and
the fifth and sixth differentials being configured to drive the seventh differential, the seventh differential having a third shaft, the arrangement being such as to provide a variable overall drive ratio between the single input and the third shaft.

17. A transmission system as defined in any preceding claim wherein the drive means includes a swash plate to vary the stroke of a transmission component.

18. A transmission system as defined in any one of claims 1 to 16 wherein the drive means includes a rotatable cam being selectively offsetable to vary the stroke of a transmission component.

19. A transmission system as defined in claims 17 or 18 wherein a transmission component is a rack.

20. A transmission system as defined in claims 17 or 18 wherein a transmission component is a cable.

21. A transmission system as defined in any preceding claim wherein the first shaft is rotatable in a first direction by a first one way clutch.

22. A transmission system as defined in claim 21 wherein the first shaft is prevented from turning in a second direction by a second one way clutch.

23. A transmission system as defined in any preceding claim wherein the drive means can vary the drive ratio between the single input and the first shaft between any one of at least twenty 20 ratios, preferably by any one of at least 50 ratios, preferably by any one of at least 100 ratios, preferably by at least any one of 140 ratios, preferably by at least any one of 200 ratios.

24. A transmission system as defined in any one of claims 1 to 22 wherein the drive means is configured to infinitely vary the drive ratio between the said single input and said first shaft so as to provide an infinitely variable overall drive ration between the single input and the third shaft.

Application No: GB0912244.1 **Examiner:** Mike McKinney

Claims searched: 1 to 24 **Date of search:** 13 November 2009

Patents Act 1977: Search Report under Section 17

Documents considered to be relevant:

Category	Relevant to claims	Identity of document and passage or figure of particular relevance
X	1 and 2 at least	US3257867 A (DENNICK) see figs and, for example, line 31 col 6 to line 56 col 7.
X	1 and 2 at least	FR2286319 A1 (FLAMANT) see figs and abstract.
X	1 and 2 at least	US2008/242468 A1 (WAFZIG) see whole document and in particular fig 12.
X	1 and 2 at least	US2006/276295 A1 (GITT) see entire specification.
X	1 and 2 at least	US2005/227809 A1 (BITZER et al) see complete document.
X	1 and 2 at least	US5827146 A (YAN et al) see, especially, fig 5 and lines 5 to 17 col 4.
X	1 and 2 at least	EP1065412 A3 (NISSAN MOTOR) see whole document, in particular fig 1.
X	1 and 2 at least	GB2263142 A (TOROTRAK DEV LTD) see complete specification.
X	1 and 2 at least	WO03/025431 A1 (ZAHNRADFABRIK FRIEDRICHSHAFEN et al) see entire specification.

Categories:

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Field of Search:

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F16H

The following online and other databases have been used in the preparation of this search report

EPODOC, WPI

International Classification:

Subclass	Subgroup	Valid From
F16H	0037/08	01/01/2006