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(54) Title: INFINITELY VARIABLE TRANSMISSION WITH UNIFORM INPUT-TO-OUTPUT RATIO THAT IS NON-DEPENDENT ON FRICTION

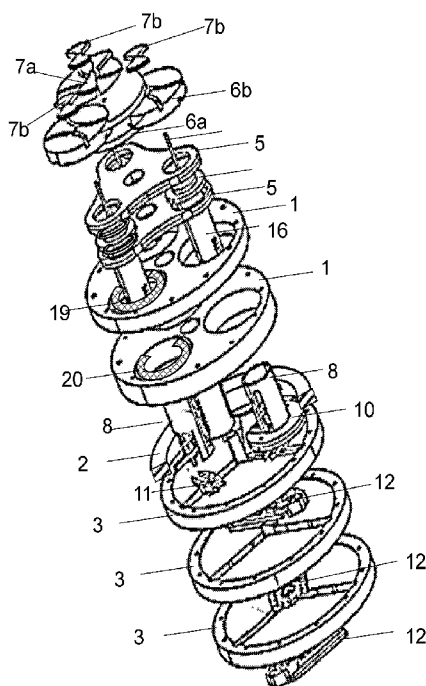


Fig.1

(57) Abstract: The present disclosure is an infinitely variable transmission that is non-dependent on friction. It can be used in high torque applications, offering a steady and uniform output for a steady and uniform input. Since it allows a co-axial input and output, by using a planetary gear system the output can be made continuous from forward to reverse. It uses a "scotch-yoke" mechanism to convert rotational motion to a linear reciprocating motion. The linear distance of this reciprocating motion - "stroke" is changed by altering the crankpin location of the scotch-yoke mechanism. This reciprocating motion is converted to a rocking motion by using a "rack and pinion" and later converted to a unidirectional motion via a One-Way-Bearing. Sets of Geneva wheel system are used to achieve a steady and uniform output, along with gear systems employing a simple mechanism to change the ratio between the input and output of the transmission.



TITLE OF INVENTION:

INFINITELY VARIABLE TRANSMISSION WITH UNIFORM INPUT-TO-OUTPUT RATIO  
THAT IS NON-DEPENDENT ON FRICTION

CROSS REFERENCE TO RELATED APPLICATIONS

BACKGROUND OF THE INVENTION:

FIELD OF THE INVENTION

This invention pertains to transmissions having variable ratios between input and output velocities. Specifically, it relates to all-gear transmissions whose velocity ratios may be changed continuously over a wide range of values ranging from zero to non-zero values, without depending on friction.

## DESCRIPTION OF THE RELATED ART

The patents US 5603240 and US 20100199805 use some of the features used in this design.

The patent US 5603240 does not have a co-axial input to output and therefore cannot be used for applications requiring this configuration. The output travels as the ratio is changed. Therefore, this design cannot be used when stationary output is required. US 20100199805 offers a sinusoidal output and uses several modules just to minimize the "ripple" when a steady and uniform input is provided. Therefore, the design cannot be used when a steady and uniform output is desired.

The patent US 9970520 offers a steady input to output ratio and co-axial input and output shaft in a comparably smaller envelope than that of its prior art. This is achieved with a use of a set of non-circular gears using as few as three modules. The drawback is that it is hard to mass produce the desired non-circular gears and will add significant manufacturing cost. It is also difficult to accurately design the tooth profile to achieve a uniform input to output ratio.

The present invention uses a custom designed Geneva wheel mechanism to achieve uniform rack velocity during functional region and circular/ non-circular gear for non-functional region. The portion of the region used by the Geneva wheel mechanism is also non-functional region which overlap with the non-functional region achieved by the circular/non-circular gears for smooth transition. It is also possible to use Geneva wheel mechanism for functional and non-functional region. However, it will be economical to use a partial circular gear for the non-functional region. The path of the Geneva slot engaging with the Geneva pin determines the shape of the functional or non-functional region. Using a commonly used Geneva wheel mechanism with straight slot will not achieve uniform rack movement in the functional region and these slots has to have a specific shape to achieve uniform rack movement in the functional region. In general, a Geneva wheel mechanism has straight slot and is commonly used in applications needing indexing.

## BRIEF SUMMARY OF THE INVENTION:

The main objective of this invention is to provide a UNIFORM and STEADY output, when the input is uniform and steady, with the ability to transmit high torque without depending on friction or friction factor. Many of the continuously variable transmissions that are in the market today are friction dependent and therefor lack the ability to transmit high torque. Those continuously variable transmissions, which are non-friction dependent do not have a uniform and steady output when the input is uniform and steady. The design that offers all is too complex and hard to mass produce. And This design aids reduction in the overall size and can be economically mass produced. This design can be easily integrated into any system. This design is very versatile and can be used ranging from light duty to heavy duty applications. This design allows replacement of existing regular transmission, requiring very little modification. This design offers stationary and co-axial input and output.

## BRIEF DESCRIPTION OF THE SEVERAL VIEWS OF THE DRAWINGS:

All the gears partial or full in the following figures can be replaced with a sprocket and chain system.

Fig 1- IVT general assembly perspective view - Exploded.

Fig 2- Angular velocity module using Geneva pin and wheel mechanism along with partial circular /non-circular gears

Fig 3a - Crank pin displacement mechanism using link mechanism with sliding collar placed coaxially inside the input shaft and crank pin shaft

Fig 3b - Crank pin displacement mechanism using link mechanism with sliding collar placed coaxially inside the input shaft and input disk

Fig 4A-4B - Scotch yoke module and rectifier module. Rectifier module showing rack and pinion, with pinions placed on a common output shaft on a one-way bearing along with dummy rack, and common output shaft. Showing force acting on the rack is co-planer with longitudinal axis of the pinion.

4A - Perspective view

4B – Perspective view exploded

Fig 5 – Input shaft and input disk assembly perspective view

Fig 6A - 6D Input disk, crank pin shaft and link pivot pin assembly

6A – Top view

6B- Front view

6C- Side view 1

6D- Perspective view

Fig 7A – 7C Different configurations for slotted rack holder

Fig 8A – 8C - Geneva pin wheel

8A- Front view

8B- Side view 1

8C- Perspective view

Fig 9 Input disk

Fig 10A – 10B – Geneva slot wheel double sided with slot and wall

10A – Perspective view showing details of the bottom

10B - Perspective view showing details of the top

10C – Perspective view showing different configuration

Fig 11 - 12 Additional optional configurations for Geneva slot wheel

Fig 13 – Scotch yoke input frame

Fig 14 – Scotch yoke frame

Fig 15 – Scotch yoke rectifier frame

Fig 16 – Ratio modifier frame

Fig 17 - Ratio plate

Fig 18 – Partial driving/driven gear for non-functional region

Fig 19 – Link

Fig 20 – crank pin

Fig 21A – 21D Link Mechanism on crank pin shaft with non-circular input shaft and collar with matching orifice, using offset crank pin mounted on crank pin collar with non-circular orifice sliding on crank pin shaft with matching cross-section

21A- Top view

21B- Front view

21C- Side view

21D-Perspective view

Fig 22A – 22D Link Mechanism on crank pin shaft with non-circular input shaft and collar with matching orifice, using input disk

22A- Top view

22B- Front view

22C- Side view

22D-Perspective view

Fig 23 – Rack velocity profile

Fig 24 -

Fig 25 – 30 – Options for connecting input, output and wheel using planetary gear

Fig 31 – Possible path of crank pin on Geneva wheel mechanism and showing partial gears for non-functional region

Fig 32 – 34 – Alternate configurations for ratio changing mechanism assembly using different shapes for input shaft, crank pin and collar

Fig 35 – 36 – Input shaft with notch for crank pin shaft, link and pivot pin for collar and link

Fig 35 – Front view

Fig 36 – Side view

Fig 37 – Dummy crank pin assembly

Fig 38 – Mechanism to compensate for vibration due to rotational imbalance

Fig 39 – Slotted hollow input shaft

39A-Top view

39B-Front view

39C-Side view

39D-Perspective view

Fig 40 – Collar with thrust bearings

40A-Top view

40B-Front view

40C-Side view

40D-Perspective view

Fig 41 – 2 racks shown 180 degrees apart

40A-Top view

40B-Front view

40C-Side view

40D-Perspective view

Fig 42 – 2 dummy racks shown 180 degrees apart

40A-Top view

40B-Front view

40C-Side view

40D-Perspective view

Fig 43A – 43B – Alternative angular velocity module using stationary sun gear

43A - Perspective view

43B – Section through driven gear

Fig 44A – 44B – Alternative angular velocity module using stationary ring gear

44A - Perspective view

44B – Section through driven gear

Fig 45 - 48 – achieving Reverse/Park/Neutral using bevel gears:

Fig 49 – Rack velocity profile and overlap of functional regions of consecutive modules in X-Y plane

Fig 50 - Rack velocity profile and overlap of functional regions of consecutive modules using polar coordinates

## DETAILED DESCRIPTION OF THE INVENTION:

## SUMMARY OF THE INVENTION

To briefly describe this invention is an Infinitely Variable Transmission (IVT). Unlike existing CVT designs, this particular design does NOT depend on friction to transmit power. Most of the CVTs that exist today depend on friction to transmit power and therefore cannot be used where there is a need to transmit high power at low speed. Due to this advantage, it is possible to use this invention where high torque transmission is required. Co-axial input and output can be achieved with this layout.

## LIST OF COMPONENTS:

All the gears in the following component list can be replaced with a sprocket and chain system. The non-circular gear system can be replaced with a sprocket and chain system where at least one of the sprockets is non-circular.

- 1) Scotch yoke Input frame
- 2) Ratio modifier frame
- 3) Scotch yoke rectifier frame
- 4) Output frame
- 5) Ratio plate
- 6) Geneva slot wheel mechanism
  - a) Pin wheel
  - b) Slot wheel
- 7A-7B) Non-functional region partial driving and driven gear
  - a) driving partial gear
  - b) driven partial gear
- 8) Input shaft
- 9) Crank-Pin
- 10) Input disk
- 11) Slotted Rack holder

- 12) Rack
- 13) Dummy Rack
- 14) Pinion
- 15) Pinion shaft
- 16) Collar
- 17) Link
- 18) Dummy link
- 19) Input shaft bearing
- 20) Input-Disk bearing
- 21) Thrust-Bearing
- 22) One-way bearing/ Computer-Controlled-Clutch / Ratchet-mechanism
- 23) Crank pin shaft
- 24) Dummy Crank-Pin
- 25) Non-functional region driving gear
- 26) Non-functional region driven gear
- 27) Crank pin to link pivot pin
- 28) Collar to link pivot pin
- 29) Power shaft
- 30) Planetary-Gear
- 31) Miter/Bevel-Gear-Differential-input shaft
- 32) Miter/Bevel-Gear-Differential-output shaft
- 33) Miter/Bevel-Gear
- 34) Rack velocity profile
- 35) Clutch-Park/Neutral/Reverse clutch/dog clutch
- 36) Stationary sun gear
- 37) Shaft-Cam
- 38) Cam gear
- 39) Driving circular or non-circular gear
- 40) Driven circular or non-circular gear

- 41) Cam-Input-Shaft
- 42) Planet gear
- 43) Stationary sun gear
- 44) Stationary ring gear
- 45) Carrier shaft

The working of this CVT can be described by the following simple sequential of operations.

- a) A Crank pin 9 (Fig. 3B), revolves around the longitudinal axis of an Input disk 10 (Fig. 9) or a Input shaft 4 (Fig. 39) at an offset distance as shown in Fig. 3A, and this offset distance can be altered. The offset distance ranges from zero to a non-zero value. The concept described in this operation exists in several other patent application US 20100199805, US 9970520 etc.
- b) This offset Crank pin 9 is caged in
  - 1) the Input disk 10 or alternatively slides on a Crank pin shaft 23, and
  - 2) a slot of a Slotted rack holder 11(Fig. 7A-7C).

The input shaft is slotted to allow the crankpin and link to pass thru it allowing the longitudinal axis of the input shaft or the input disk to be co-axial with the longitudinal axis of the crank pin. The Slotted rack holder 11 is restricted such that it can move only in the direction that is normal to its slot. A Rack 12 is fastened to the Slotted rack holder 11, such that the Rack 12 is parallel to the Slotted rack holder's 44 direction of movement. In the alternative construction, the Crank pin shaft 23 is orthogonal to the Input shaft 4. The revolution of the Crank pin 9 about the longitudinal axis 1021 of Input disk 10 is translated to pure linear back and forth movement or reciprocating movement of the Rack 12. This mechanism is commonly known as "Scotch-Yoke-Mechanism" in the industry. The distance of this linear back and forth movement (stroke) is directly proportional to the radial distance of the Crank pin 9 from the longitudinal axis 1021 of the Input disk 10. Since the work done is constant, which is a product of force applied multiplied by the distance traveled ( $F \cdot \text{stroke}$ ), for a smaller stroke, the force applied is greater and for a longer stroke, the force applied is smaller.

- c) The Rack 12 is linked to a Pinion 14 (Fig. 4A) converting this linear movement of the Rack 12 to rocking oscillation of the Pinion 14.

- d) This rocking oscillation is converted to a unidirectional rotation, using a One-way bearing/ Computer-Controlled-Clutch / Ratchet-mechanism 22.

One main purpose of this invention is to achieve a CONSTANT AND UNIFORM output angular velocity when the input angular velocity is constant and uniform. However, using the steps described above, this is NOT achieved, as the output is sinusoidal.

By modifying the rate of change of angular displacement of the Input disk 10, a uniform steady output can be achieved. Patent US9970520 uses a pair of non-circular gears to achieve this. This invention achieves it by using modified Geneva mechanism customized for this.

By using a set of Geneva pin wheel 6a (Fig. 8A-8C) and Geneva slot wheel 6b (Fig. 10A – 10C) the instantaneous rate of change of angular displacement at the Input disk 10 can be altered.

The components are grouped into modules/mechanisms for easier understanding:

Detailed description of Assembly, Sub-assembly of components/ Modules and their functions:

a) Angular-Velocity-Modifier-Module (Fig. 2) The main purpose of this module is to change the uniform power input to a reciprocal of sinusoidal output. This is to reverse the effect of the sinusoidal output in a scotch yoke mechanism. This module comprises of:

- 1) Driving Geneva pin wheel,
- 2) Driven Geneva slot wheel and
- 3) Power shaft

The Driving Geneva pin wheel 6a is mounted on the Input shaft 4. The shape of the Geneva slot wheel 6b is designed to achieve the end result which is the reciprocal of sinusoidal output. Multiple pins and multiple slots are used and with an overlap of more than one pin achieving a portion of the same results simultaneously. More than one set of driving Geneva pin wheel and driven Geneva slot wheel can be used in a single module. The slot or the walls of the slot are terminated where a pin's path forms loop. Also, multiple modules can share a common Geneva pin wheel or a common Geneva slot wheel. In the slot wheel the paths of the slots are cut from a slot wheel or the walls of the path can be raised from a slot wheel or a combination of both. This is to clear the interference of the pin where the pin and slot or slot walls do not produce desired result. Pins of the Geneva pin wheel can be made with different heights so that they do not interfere with the wall of slots of other Geneva pins. A portion of the rotation of the Geneva pin wheel and Geneva slot wheel be achieved using one or more partial circular gears and/or one or more partial non-circular gear in parallel. The partial gears generate the non-functional region of the rack velocity while the Geneva wheel system generates functional portion of the rack velocity. The Geneva wheel slots also have an overlap of the region generated by the partial gear. This is to achieve a 1:X ratio of rotation between Geneva pin wheel and Geneva slot wheel. Here X is an integer or a reciprocal of an integer. Optionally, a one-way bearing can be placed between the circular or non-circular gear linking the Geneva slot wheel to the partial driven gear. Depending on the scenario either the Geneva pin wheel or the Geneva slot wheel can be made driving or driven.

b) Scotch-Yoke-Module (Fig. 4A, 4B): The main purpose of this module is to convert circular motion to a reciprocating motion. The output is sinusoidal for a steady, uniform input. This output is converted to a steady, uniform output using Angular-Velocity-Modifier-Module.

This Scotch-Yoke-Module comprises of:

- 1) Input disk 10,
- 2) Slotted Rack holder 11, and
- 3) Crank pin 9

The Input disk 10 has a radial slot.

The Slotted rack holder 11 has a slot namely "Crank-Pin-Slot" 1013. It also has an extension on either side of the slot at the middle of the slot. This extension is normal to the Crank-Pin-Slot 1013. The Slotted rack holder 44, is placed on the other side of the Input disk 10 sandwiching the Input disk 10 between the Slotted rack holder 11 and a Ratio-Changing- Mechanism, which is described in subsequent paragraphs. The Crank pin 9 passes through the slots of Ratio-Changing-Mechanism, Input disk 10, and Slotted rack holder 11

c) Rectifier-Module: The main purpose of this module is a mechanical equivalent to a diode in an electrical circuit. It allows power transfer to one specific direction.

- 1) Rack 12,
- 2) Pinion 14,
- 3) Shaft-Pinion 48 and
- 4) One-way bearing/ Computer-Controlled-Clutch / Ratchet-mechanism 22

The Rack 12 is attached to the Slotted rack holder 11 normal to the Crank-Pin-Slot 1013 and paired with the Pinion 14. The Pinion 14 is mounted on a Shaft-Pinion 48. The Computer-Controlled-Clutch / One-Way-Bearing / Ratchet-Mechanism 50 is mounted on the Shaft-Pinion 48. The Output-Gear / Output-Sprocket 51 is mounted on the OD of the One-Way-Bearing 50.

Multiple pinions from multiple modules can be mounted on a common shaft-pinion 48. The one-way bearing can be placed between the pinion and the pinion shaft. In this scenario the shaft-pinion 48 will function as the CVT output. The shaft-pinion can be made hollow so that the CVT input shaft can pass thru the shaft-pinion 48 making the input and output of the CVT co-axial.

#### d) Gear-Changing-Mechanisms

##### Link Mechanism:

The Input shaft 66 has a non-circular hole in the middle. This is paired with a Sliding-Collar with a matching exterior contour, which is co-axially placed allowing relative axial movement while restricting rotational angular displacement with respect to each other. Two thrust-Bearings 40 are co-axially placed in contact with one on either end of the Sliding-Collar 67 as shown in Fig. 22D and the Sliding-Collar-Auxiliary-Shaft 67 has a pivot 1028 on the other end. One end of a Link 68 is attached to the pivot 1028 and the other end of the Link 68 is either attached to the Crank pin 9, as shown in (Fig. 3A) or to the Crank pin shaft 23, as shown in (Fig. 3A) as appropriate. An axial displacement of the Sliding-Collar-Auxiliary-Shaft 67 will cause a radial displacement of the Crank pin 9 thru the Link 68. This axial translation is achieved with a Lever-Ratio-Changing-Spiral-Flute-Mechanism 41 that pushes the Thrust-Bearing 40 attached to the Sliding-Collar-Auxiliary-Shaft 67. Optionally this can be sprung back with a Compression-Spring 39 placed between Input disk 10 and the Sliding-Collar-Auxiliary-Input-Shaft 67. Also, when this link mechanism is used the Driven-Geneva-slot wheel 6b can also function as the Input disk 10 when a radial slot is added to the Driven-Geneva-slot wheel 6b, thereby eliminating the need for a separate Input disk 10.

For each scotch yoke module two Racks 64 can be placed on the Slotted rack holder 11 with a phase shift of 180° engaged with their respective pinions placed co-axially on a common pinion shaft via a

one-way bearing/computer-controlled clutch/a ratchet mechanism to allow the pinion shaft to rotate in a specific direction. Many of these scotch yoke modules can be stacked and all the pinions of all the modules can be placed on one common pinion shaft, making the pinion shaft the output of the IVT. Further this common pinion shaft can be made hollow allowing the power shaft which drives the driving Geneva pin wheel, to pass thru. With this arrangement a co-axial input to output can be achieved. This configuration allows to modify the output with a planetary gear system to achieve reverse gear converting the CVT to an IVT. This configuration also allows the force on the rack holder to pass thru the plane of the common pinion shaft axis. In other words, the longitudinal axis of the common pinion and the force of the crankpin acting on the rack holder will be co-planer. This will minimize the moment of the force from the crank pin acting on the rack holder due to the resistance by the pinion and maximize the tangential force on the pinion.

Two Rectifier-Modules 1001 are placed next to the Slotted rack holder 11 as shown in Fig. 83 such that the Rack 12 is placed normal to the Slotted rack holder's 44 Crank-Pin-Slot 1013.

#### MECHANISM TO COMPENSATE VIBRATION (ROTATIONAL IMBALANCE):

1. Dummy-Crank-Pin 43: The Crank pin 9 is placed off-center when the Input disk 10 revolves. This imbalance will result in vibration. To compensate this, a Dummy-Crank-Pin 43 is placed at same distance  $180^\circ$  apart. This movement is identical to the movement of the Crank pin 9. The dummy crank pin is attached to a dummy link that links to the dummy crank pin 43 that is pivoted to the collar placed to move in the opposite direction of the crank-pin. The input shaft is slotted to allow the link and crank pin and dummy link and dummy crank pin to pass thru

2. Dummy-Rack 55 for counter oscillation: As the Input disk 10 rotates the Slotted Rack holder 11 has an oscillatory motion which will result in vibration. It is cancelled by having an appropriate mass oscillating in the opposite direction. This is achieved by pinion as shown in Fig. 4A&4B in contact with the Rack 12, which will spin back and forth. Bringing an appropriate mass in contact with the pinion at  $180^\circ$  apart will compensate for this vibration. A separate wheel can also be used in spite of the pinion or a lever pivoting on the pinion shaft can be used to link the rack and the dummy rack such that they move in opposite direction. A slider connecting the lever and sliding in a slot normal to the rack teeth will guide the lever allowing the rack and dummy rack to only slide in the direction of the longitudinal axis of the rack.

#### REVERSE GEAR MECHANISM:

When the output from the Pinion shaft 15 is coupled with Miter/Bevel-Gear-Differential-input shaft 31. The Miter/Bevel-Gear-Differential-output shaft 32 will rotate in opposite directions via Miter/Bevel-Gear 33. The Miter/Bevel-Gear-Differential-input shaft 31 of this differential-mechanism is placed co-axial with The Miter/Bevel-Gear-Differential-output shaft 32 with clearance so that it is free to spin independently with respect to the Miter/Bevel-Gear-Differential-input shaft 31. Two Clutch-Park/Neutral/Reverse clutch/dog clutch

35 with a clutch are placed on the Miter/Bevel-Gear 33 allowing them to move axially. These can be made to link with

either of the Miter/Bevel-Gear 33, which rotate in opposite direction. When one of the collars is made to link via the Clutch-Park/Neutral/Reverse clutch/dog clutch 35, by means of clutch, with a particular output Clutch-Park/Neutral/Reverse clutch/dog clutch 35 and the Miter/Bevel-Gear-Differential-output shaft 31 will rotate in a particular direction. It will reverse its direction if the link is swapped to the other Miter/Bevel-Gear 33.

#### NEUTRAL GEAR MECHANISM:

When the collars are not in link via the Clutch-Park/Neutral/Reverse clutch/dog clutch 35 with any of the Miter/Bevel-Gear 33, the collar and the Miter/Bevel-Gear-Differential-output shaft 32 are not restricted and, therefore, they are free to spin in any direction and function as a "neutral" gear.

#### PARK MECHANISM:

When the collars are in link via the Clutch-Park/Neutral/Reverse clutch/dog clutch 35 with both the Miter/Bevel-Gear 33, the collar is restricted from spinning and the Miter/Bevel-Gear-Differential-output shaft 32 is totally restricted and, therefore, they are restricted to spin in any direction and function as a "parking" gear.

#### CONVERTING CVT TO AN IVT (INFINITELY-VARIABLE-TRANSMISSION):

Having a co-axial input and output allows the CVT to function as an IVT. This can be achieved by adding a Planetary-Gear-System with a Sun-Gear, Ring-Gear and Planets supported by Carriers, and linking with Input shaft 4, the Co-Axial-Output-Element-With-Internal-Gear/Planetary-Gear 65.

The following are the options to achieve this:

- a) The Input shaft 4 is directly linked to the Sun-Gear of the planetary-Gear-System with following 2 sub-options
  - a. The Co-Axial-Output-Element-With-Internal-Gear/Planetary-Gear 65 is directly linked to the Carrier of the Planetary-Gear-System and Ring-Gear of the Planetary-Gear-System functions as the final output or wheel system 1022
  - b. The Co-Axial-Output-Element-With-Internal-Gear/Planetary-Gear 65 is linked to the Ring-Gear of the Planetary-Gear-System and the Carrier functions as the final output or wheel system 1022.
- b) The Co-Axial-Output-Element-With-Internal-Gear/Planetary-Gear 65 is directly linked to the Sun-Gear of the Planetary-Gear-System with following 2 sub-options.
  - a. The Input shaft 4 is directly linked to the Carrier of the Planetary-Gear-System and the Ring-Gear of the Planetary-Gear-System functions as the final output or wheel system 1022.
  - b. The Input shaft 4 is directly linked to the Ring-Gear of the Planetary-Gear-System and the Carrier functions as the final output or wheel system.

- c) The Input shaft 4 is directly linked to the Ring-Gear of the planetary-Gear-System with following 2 sub-options
  - a. The Co-Axial-Output-Element-With-Internal-Gear/Planetary-Gear 65 is directly linked to the Carrier of the Planetary-Gear-System and Sun-Gear of the Planetary-Gear-System functions as the final output or wheel system 1022.
  - b. The Co-Axial-Output-Element-With-Internal-Gear/Planetary-Gear 65 is linked to the Sun-Gear of the Planetary-Gear-System and the Carrier functions as the final output or wheel system 1022.
- d) The Co-Axial-Output-Element-With-Internal-Gear/Planetary-Gear 65 is directly linked to the Ring-Gear of the Planetary-Gear-System with following 2 sub-options.
  - a. The Input shaft 4 is directly linked to the Carrier of the Planetary-Gear-System and the Carrier of the Planetary-Gear-System and the Sun-Gear of the Planetary-Gear-System functions as the final output or wheel system 1022.
  - b. The Input shaft 4 is directly linked to the Sun-Gear of the Planetary-Gear-System and the Carrier functions as the final output or wheel system 1022.
- e) The Input shaft 4 is directly linked to the Carrier of the planetary-Gear-System with following 2 sub-options
  - a. The Co-Axial-Output-Element-With-Internal-Gear/Planetary-Gear 65 is directly linked to the Ring-Gear of the Planetary-Gear-System and Sun-Gear of the Planetary-Gear-System functions as the final output or wheel system 1022.
  - b. The Co-Axial-Output-Element-With-Internal-Gear/Planetary-Gear 65 is linked to the Sun-Gear of the Planetary-Gear-System and the Sun-Gear functions as the final output or wheel system 1022.
- f) The Co-Axial-Output-Element-With-Internal-Gear/Planetary-Gear 65 is directly linked to the Carrier of the Planetary-Gear-System with following 2 sub-options.
  - a. The Input shaft 4 is directly linked to the Ring-Gear of the Planetary-Gear-System and the Ring-Gear of the Planetary-Gear-System and the Sun-Gear of the Planetary-Gear-System functions as the final output or wheel system 1022.
  - b. The Input shaft 4 is directly linked to the Sun-Gear of the Planetary-Gear-System and the Ring-Gear functions as the final output or wheel system 1022.

In other words, The Co-Axial-Output-Element-With-Internal-Gear/Planetary-Gear 65 is connected to one of the three elements, either a Ring-Gear, a Carrier, or a Sun-Gear of a Planetary-Gear-System. The Input shaft 4 is connected to one of the remaining two elements of the Planetary-Gear-System. The third remaining element of the Planetary-System functions as the final output or wheel system 1022. This converts the CVT to an IVT.

#### COMPENSATING FOR DEVIATION IN RACK MOVEMENT WITH CAMS:

It is beneficial to have smooth and gradual transitions in the rack movement profile to

improve the life of the transmission. As shown in Fig. 23, the ideal rack velocity profile is as follows:

1. gradual increase in acceleration from rest 1025
2. a region of acceleration 1026
3. gradual reduction in acceleration to a constant velocity 1027
4. a region of constant velocity 1028
5. gradual increase in deceleration to a constant deceleration 1029
6. a region of deceleration 1030
7. gradual reduction in deceleration to zero velocity 1031
8. steps 1 through 7 above repeated in the opposite direction

It may not always be possible to generate perfect Geneva wheel mechanism to meet the above desired Rack 12 movement. If the slot curves 1006 of the Geneva slot wheel and the Geneva pin wheel 6a & 6b do not to achieve this desired Rack 12 movement, a planetary system can be used to compensate for any deviations from the desired Rack 12 movement profile. To achieve this, a Stationary Sun Gear 36 with respect to the ratio modifier frame 2 is placed co-axial with a driven circular or non-circular gear 40 which is driven by a driving circular or non-circular gear 39 as appropriate. This can be used in addition to the Geneva wheel system. This is shown in Fig. 43A & 43B. This driving circular or non-circular gear is mounted on the power shaft 29. One or more Shaft-Cam 37 is placed on the driven circular or non-circular gear 40 which acts like a carrier of the planetary gear system. A Cam-Gear 38 is rigidly attached on the Shaft-Cam 37. Each of this Cam-Gear 38 is made to engage another Cam-Input-Shaft 41 each, which is rigidly attached on the Input shaft 8. The cams can be designed to give a desired rack velocity profile. The above configuration will also work when the stationary sun is replaced with a stationary ring gear. This is shown in Fig. 44A & 44B.

#### Mathematical Model

The following formula is used in pitch curve generation of the driving non-circular gear and driven non-circular gear, when expressed using Cartesian coordinates  $(X_1, Y_1)$  and  $(X_2, Y_2)$  respectively, as a function of angle  $\theta$  are

$$X_1 \approx \frac{CTR * \frac{d\Phi(\theta)}{d\theta}}{1 + \frac{d\Phi(\theta)}{d\theta}} * \cos(\theta), Y_1 \approx \frac{CTR * \frac{d\Phi(\theta)}{d\theta}}{1 + \frac{d\Phi(\theta)}{d\theta}} * \sin(\theta);$$

$$X_2 \approx \frac{CTR}{1 + \frac{d\Phi(\theta)}{d\theta}} * \cos(\Phi(\theta)), Y_2 \approx \frac{CTR}{1 + \frac{d\Phi(\theta)}{d\theta}} * \sin(\Phi(\theta));$$

where  $\Phi(\theta)$  is a solution to a piece-wise differential equation which gives rotation ratio of the driving and driven Geneva wheel system as well as the driving and driven non-circular gears.

$$N * \frac{d\Phi}{d\theta} * \sin(N * \Phi) =$$

function of any linear or nonlinear curve connecting points  $\left(\frac{i * 2 * \pi}{N * n}, 0\right)$  to  $(\theta_{1i}, k_i)$

$$\text{if } \frac{i * 2 * \pi}{N * n} < \theta < \theta_{1i},$$

$$k_i \text{ if } \theta_{1i} < \theta < \theta_{2i},$$

function of any linear or nonlinear curve connecting points  $(\theta_{2i}, 0)$  to  $(\theta_{3i}, -k_i)$

$$\text{if } \theta_{2i} < \theta < \theta_{3i},$$

$$-k_i \text{ if } \theta_{3i} < \theta < \theta_{4i},$$

function of any linear or nonlinear curve connecting points  $(\theta_{4i}, 0)$  to

$$\left( \frac{(i+1) * 2 * \pi}{N * n}, -k_i \right) \text{ if } \theta_{4i} < \theta < \frac{(i+1) * 2 * \pi}{N * n};$$

or

$$N * \frac{d\Phi}{d\theta} * \cos(N * \Phi) =$$

$$k_i \text{ if } \frac{i * 2 * \pi}{N * n} < \theta < \theta_{1i},$$

function of any linear or nonlinear curve connecting points  $(\theta_{1i}, k_i)$  to  $(\theta_{2i}, -k_i)$

$$\text{if } \theta_{1i} < \theta < \theta_{2i},$$

$$-k_i \text{ if } \theta_{2i} < \theta < \theta_{3i},$$

function of any linear or nonlinear curve connecting points  $(\theta_{3i}, -k_i)$  to  $(\theta_{4i}, k_i)$

$$\text{if } \theta_{3i} < \theta < \theta_{4i},$$

$$k_i \text{ if } \theta_{4i} < \theta < \frac{(i+1) * 2 * \pi}{N * n},$$

where boundary conditions are

$$\Phi(0) = 0;$$

$$\Phi\left(\frac{(i+1) * 2 * \pi}{N * n}\right) = \frac{(i+1) * 2 * \pi}{N};$$

$$\Phi(\theta_{1i}) = \frac{i * 2 * \pi}{N} + \Phi_1;$$

$$\Phi(\theta_{2i}) = \frac{i * 2 * \pi}{N} + \Phi_2;$$

$$\Phi(\theta_{3i}) = \frac{i * 2 * \pi}{N} + \Phi_3;$$

$$\Phi(\theta_{4i}) = \frac{i * 2 * \pi}{N} + \Phi_4;$$

Where,

CTR is a center-to-center distance of the driving non-circular gear and the driven non-circular gear,

$\theta$  is an angular displacement of the driving non-circular gear;

$\Phi$  is an angular displacement of the driven non-circular gear;

$i$  refers to an  $i$ -th revolution the input disk from 0 to  $N \cdot n - 1$  with a 1st rotation being  $i=0$ ;

$N$  is a number of times the input disk rotates when the driven non-circular gear rotates once;

$n$  is a number of times the driven non-circular gear rotates when the driving non-circular gear rotates once;

regions where the piece-wise function for the rack velocity is constant are functional regions and regions where the piece-wise function for the rack velocity is not constant are non-functional regions which can be linear or non-linear functions of  $\theta$ ;

$\theta_{1i}, \theta_{2i}, \theta_{3i}, \theta_{4i}$  are specific angular positions of the driving non-circular gear, the values of which are solved for using a solution to the piece-wise differential equation;

$\phi_1, \phi_2, \phi_3, \phi_4$  are specific angular positions of the driven non-circular gear corresponding to angular positions  $\theta_{1i}, \theta_{2i}, \theta_{3i}, \theta_{4i}$  of the driving non-circular gear respectively, and are a cutoff between the functional and non-functional regions, values of  $\phi_1, \phi_2, \phi_3, \phi_4$  which can to be solved for by using arbitrary values for  $\theta_{1i}, \theta_{2i}, \theta_{3i}, \theta_{4i}$  ;

and  $k_i$  are constants, which are all equal.

## CLAIMS:

## 1. An infinitely variable transmission comprising:

one or more driving Geneva pin wheels mounted on an input shaft, operably connected to one or more driven Geneva slot wheels each operably connected to rotate an input disk of a scotch yoke mechanism, causing a crank pin of the scotch yoke mechanism, placed at an offset distance to an axis of rotation of the input disk where the offset distance can be altered from 0 to a real value using an external force, to revolve around the axis of rotation of the input disk, which reciprocates one or more racks, which are restricted to only move along the rack's pitch line and each rack rocks a pinion comprising a one way bearing that is mounted on a hollow output shaft that is co-axially placed with the input shaft, wherein the input shaft passes completely through the output shaft.

## 2. An infinitely variable transmission comprising:

## A) at least one scotch yoke module comprising:

- a. a crank pin revolving around
- b. a notched input shaft, at an offset distance between a longitudinal axes of the crank pin and the auxiliary input shaft that remain parallel to each other, and the offset distance that can be altered when the crank pin is co-axial to the auxiliary input shaft from zero to a non-zero real number by displacing the crank pin along a radial slot of
- c. an input disk rigidly mounted on the input shaft, by

## B) a crank pin displacement mechanism comprising:

- a. a sliding collar disposed co-axially with the auxiliary input shaft with a feature preventing relative angular displacement while allowing relative translation,
- b. a link assembly comprising
  - i. a link pivoting the crank pin through the notch
  - ii. a crank pin pivot pin on one end and pivoting the sliding collar about
  - iii. a sliding collar pivot pin on another end of the link,
- c. at least one thrust bearing that is co-axially placed in contact with the sliding collar, such that an external force applied on the thrust bearing causing an axial displacement of the thrust bearing along with the sliding collar with respect to the auxiliary input shaft, alters the offset distance by moving the crank pin along the radial slot of the input disk,
- d. a slotted rack holder comprising one or more racks, which is restricted to only move along a direction of the longitudinal axis of the one or more racks, and a crank pin slot for receiving the crank pin, with a longitudinal axis of the crank pin slot orthogonal to the one or more racks,

## C) at least one angular velocity module comprising:

- a. an input shaft,
  - b. one or more driving Geneva pin wheels mounted on the input shaft and driving
  - c. at least one driven Geneva slot wheel each rotating the input shaft
- D) at least one rectifier module comprising:
- a. a pinion engaged with the rack, and mounted on
  - b. a pinion shaft through
  - c. a computer-controlled clutch, a one-way clutch or a ratchet mechanism

arranged such that a uniform rotation of the driving Geneva pin wheel via the input shaft, causes a non-uniform angular velocity of the input shaft via the driven Geneva slot wheel and the planetary gear system, causing the crank pin to reciprocate the rack substantially along a longitudinal direction of the rack at a substantially constant velocity and slowing down briefly during direction reversal and accelerating to the constant velocity, where a magnitude of the reciprocation is proportional to the offset distance of the crank pin and the auxiliary input shaft, and this reciprocation of the rack causes an alternating rotation of the pinion and this alternating rotation of the pinion is converted to a unidirectional rotation of the pinion shaft via the computer-controlled clutch, the one-way clutch or the ratchet mechanism.

3. The infinitely variable transmission of claim 2, wherein the feature preventing relative angular displacement while allowing relative translation between the sliding collar and the auxiliary input shaft is further defined as one of the co-axially placed sliding collar or the auxiliary input shaft having a non-circular cross section and the other of the sliding collar and the auxiliary input shaft having a non-circular orifice matching the non-circular cross section.

4. Currently Amended) A infinitely variable transmission comprising:

A) at least one scotch yoke module comprising:

- a) a crank pin perpendicularly mounted on
- b) a crank pin collar having a non-circular orifice and sliding on
- c) a co-axial crank pin collar shaft having a matching non-circular cross-section, and the crank pin collar shaft is mounted perpendicularly on
- d) a notched auxiliary input shaft, such that a longitudinal axis of the crank pin is coplanar and parallel and at an offset distance to the longitudinal axes of the auxiliary input shaft, wherein the offset distance can be altered by displacing the crank pin by
- e) a crank pin displacement mechanism comprising:
  - i. a sliding collar disposed co-axially with the auxiliary input shaft, wherein one of the sliding collar and the auxiliary input shaft has a non-circular cross-section and another of the sliding collar and the auxiliary input shaft has a matching non-circular orifice,

such that the sliding collar and the auxiliary input shaft rotate synchronously with each other while having the ability to slide axially relative to each other,

ii. a link assembly comprising

- a) a link pivoting on the sliding collar and the crank pin collar through the notch
- b) a sliding collar pivot pin on one end of the link and pivoting the crank pin collar with
- c) a crank pin collar pivot pin on another end
- f) at least one thrust bearing that is co-axially placed in contact with the sliding collar, such that an external force applied on the thrust bearing causes an axial displacement of the thrust bearing and the sliding collar with respect to the auxiliary input shaft, which alters the offset distance by moving the crank pin collar together with the crank pin along the crank pin collar shaft,
- g) a slotted rack holder comprising one or more racks, which is restricted to only move along a direction of a longitudinal axis of the one or more racks, and a crank pin slot for receiving the crank pin, with a longitudinal axis of the crank pin slot orthogonal to the one or more racks,

B) at least one angular velocity module comprising:

- a) an input shaft,
- b) at least one driving Geneva pin wheel mounted on the input shaft and driving
- c) at least one driven Geneva slot wheel mounted co-axially on the auxiliary input shaft, at a fixed orientation to the axis of the crank pin shaft and

C) at least one rectifier module comprising:

- a) a pinion engaged with the one or more racks, and mounted on
- b) a pinion shaft through
- c) a computer-controlled clutch, a one-way bearing, or a ratchet mechanism;

arranged such that a uniform rotation of the driving Geneva pin wheel via the input shaft, causes a non-uniform angular velocity of the auxiliary input shaft via the driven Geneva slot wheel, causing the crank pin to revolve the auxiliary input shaft reciprocating the one or more racks substantially along a longitudinal direction of the one or more racks at a substantially constant velocity and slowing down briefly during direction reversal and accelerating to the constant velocity, where a magnitude of the reciprocation is proportional to the offset distance of the crank pin and the auxiliary input shaft, and this reciprocation of the rack causes an alternating rotation of the pinion and the rotation of the pinion is converted to a unidirectional rotation of an output gear, or output sprocket mounted on the pinion shaft via the computer controlled clutch, the one way bearing or the ratchet mechanism.

5. The infinitely variable transmission of claim 2, wherein rotation ratio of driving Geneva pin wheel of the driven Geneva slot wheel, when expressed using Cartesian coordinates  $(X_1, Y_1)$  and  $(X_2, Y_2)$  respectively, as a function of angle  $\theta$  are

where  $\phi(\theta)$  is a solution to a piece-wise differential equation

$$N * \frac{d\phi}{d\theta} * \sin(N * \phi) =$$

*function of any linear or nonlinear curve connecting points  $\left(\frac{i * 2 * \pi}{N * n}, 0\right)$  to  $(\theta_{1i}, k_i)$*

$$\text{if } \frac{i * 2 * \pi}{N * n} < \theta < \theta_{1i},$$

$$k_i \text{ if } \theta_{1i} < \theta < \theta_{2i},$$

*function of any linear or nonlinear curve connecting points  $(\theta_{2i}, 0)$  to  $(\theta_{3i}, -k_i)$*

$$\text{if } \theta_{2i} < \theta < \theta_{3i},$$

$$-k_i \text{ if } \theta_{3i} < \theta < \theta_{4i},$$

*function of any linear or nonlinear curve connecting points  $(\theta_{4i}, 0)$  to*

$$\left(\frac{(i+1) * 2 * \pi}{N * n}, -k_i\right) \text{ if } \theta_{4i} < \theta < \frac{(i+1) * 2 * \pi}{N * n};$$

or

$$N * \frac{d\phi}{d\theta} * \cos(N * \phi) =$$

$$k_i \text{ if } \frac{i * 2 * \pi}{N * n} < \theta < \theta_{1i},$$

*function of any linear or nonlinear curve connecting points  $(\theta_{1i}, k_i)$  to  $(\theta_{2i}, -k_i)$*

$$\text{if } \theta_{1i} < \theta < \theta_{2i},$$

$$-k_i \text{ if } \theta_{2i} < \theta < \theta_{3i},$$

*function of any linear or nonlinear curve connecting points  $(\theta_{3i}, -k_i)$  to  $(\theta_{4i}, k_i)$*

$$\text{if } \theta_{3i} < \theta < \theta_{4i},$$

$$k_i \text{ if } \theta_{4i} < \theta < \frac{(i+1) * 2 * \pi}{N * n},$$

where boundary conditions are

5

$$\Phi(0) = 0;$$

$$\Phi\left(\frac{(i+1) * 2 * \pi}{N * n}\right) = \frac{(i+1) * 2 * \pi}{N};$$

$$\Phi(\theta_{1i}) = \frac{i * 2 * \pi}{N} + \Phi_1;$$

$$\Phi(\theta_{2i}) = \frac{i * 2 * \pi}{N} + \Phi_2;$$

$$\Phi(\theta_{3i}) = \frac{i * 2 * \pi}{N} + \Phi_3;$$

$$\Phi(\theta_{4i}) = \frac{i * 2 * \pi}{N} + \Phi_4;$$

where

CTR is a center to center distance of the driving Geneva pin wheel and the driven Geneva slot wheel,

$\theta$  is an angular displacement of the driving Geneva pinwheel;

$\Phi$  is an angular displacement of the driven Geneva slot wheel;

$i$  refers to an  $i$ -th revolution the input disk from 0 to  $N*n-1$  with a 1st rotation being  $i=0$ ;

$N$  is a number of times the input disk rotates when the driven Geneva pin wheel rotates once;

$n$  is a number of times the driven Geneva slot wheel rotates when the driving Geneva slot wheel once;

regions where the piece-wise function for the rack velocity is constant are functional regions and regions where the piece-wise function for the rack velocity is not constant are non-functional regions which can be linear or non-linear functions of  $\theta$ ;

$\theta_{1i}, \theta_{2i}, \theta_{3i}, \theta_{4i}$  are specific angular positions of the driving Geneva pin wheel, the values of which are solved for using a solution to the piece-wise differential equation;

$\Phi_1, \Phi_2, \Phi_3, \Phi_4$  are specific angular positions of the driven Geneva slot wheel corresponding to angular positions  $\theta_{1i}, \theta_{2i}, \theta_{3i}, \theta_{4i}$  of the driving Geneva pin wheel respectively, and are a cutoff between the functional and non-functional regions, values of  $\Phi_1, \Phi_2, \Phi_3, \Phi_4$  which can to be solved for by using arbitrary values for  $\theta_{1i}, \theta_{2i}, \theta_{3i}, \theta_{4i}$ ;

and  $k_i$  are constants, which are all equal.

6 The infinitely variable transmission of claim 2, further comprising one or more additional driving Geneva pin wheel and driven Geneva slot wheel pairs, wherein the pairs of driving Geneva pin wheel and driven Geneva slot wheel are stacked in layers and a sum of all functional regions of all the pairs of driving Geneva pin wheel and driven Geneva slot wheel in each angular velocity module is greater than or equal to  $360^\circ$  and is placed such that the Geneva pin wheel and slot wheel pairs are in the functional region in sequence with an overlap between the functional regions of consecutive driven Geneva slot wheels.

- 7) The infinitely variable transmission of claim 2, wherein the angular velocity modules are oriented such that their Geneva pin wheel and the Geneva slot wheel are in the functional region in sequence with an overlap when the input disk completes approximately one rotation, ensuring that at least one angular velocity module is in the functional region at any given time.
- 8) The infinitely variable transmission of claim 18, wherein an amount of overlap between each pair of consecutively engaged rectifier modules are substantially identical.
- 9) The infinitely variable transmission of claim 2, further comprising a dead weight and a wheel that transfers motion from the rack to a dummy rack with teeth identical to the rack and located 180 degrees relative to the rack, and the dummy rack moves in a substantially opposite direction of the rack.
- 10) The infinitely variable transmission of claim 2, further comprising a dummy crank pin having a mass substantially identical to a mass of the crank pin that slides in an opposite direction of the crank pin.
- 11) The infinitely variable transmission of claim 2, further comprising:
  - a differential assembly comprising an input miter bevel gear and a pair of substantially co-axial output miter bevel gears operably connected with the input miter bevel gear such that the output miter bevel gears rotate in opposite directions, each output miter bevel gear having a through-bore substantially at a central axis thereof and substantially co-axial with each other;
  - a through-shaft positioned through the through-bores of the output miter bevel gears; and a pair of collars operably coupled with the through-shaft and rotatably fixed therewith, each collar configured to move axially along the through-shaft independently of the other collar and configured to engage with one of the output miter bevel gears;
  - wherein the power link shaft is operably coupled with the input miter bevel gear to cause rotation of the input miter gear.
- 12) The infinitely variable transmission of claim 11, wherein:
  - when a first one of the collars is engaged with a first one of the output miter bevel gears and a second one of the collars is not engaged with a second one of the output miter bevel gears, the through-shaft rotates about its longitudinal axis in a first direction corresponding to a rotational direction of the first one of the output miter bevel gears; and
  - when the second one of the collars is engaged with the second one of the output miter bevel gears and the first one of the collars is not engaged with the first one of the output miter bevel gears, the through-shaft rotates about its longitudinal axis in a second direction corresponding to a rotational direction of the second one of the output miter bevel gears.
- 13) The infinitely variable transmission of claim 11, wherein when neither of the collars is engaged with the output miter bevel gears, the through-shaft is free to rotate in any direction about its longitudinal axis.

- 14) The infinitely variable transmission of claim 11, wherein when each of the collars is engaged with a respective one of the output miter bevel gears, the through-shaft is restricted from rotating about its longitudinal axis.
- 15) The infinitely variable transmission of claim 14, wherein the input shaft is connected to a ring gear, a carrier or a sun gear, the output from the output gear thru an output shaft is connected to another one of the ring gear, the carrier or the sun gear and the final output is connected to another one of the ring gear, the carrier or the sun gear.
- 16) The infinitely variable transmission of claim 14, wherein a final output from a planetary gear system temporarily stores energy in a fly-wheel-system and later delivers power back to the input shaft or to a wheel-system.
- 17) A infinitely variable transmission comprising:
  - A) at least one scotch yoke module comprising:
    - h) a crank pin perpendicularly mounted on
    - i) a crank pin collar having a non-circular orifice and sliding on
    - j) a co-axial crank pin collar shaft having a matching non-circular cross-section, and the crank pin collar shaft is mounted perpendicularly on
    - k) a notched auxiliary input shaft, such that a longitudinal axis of the crank pin is coplanar and parallel and at an offset distance to the longitudinal axes of the auxiliary input shaft, wherein the offset distance can be altered by displacing the crank pin by
    - l) a crank pin displacement mechanism comprising:
      - iii. a sliding collar disposed co-axially with the auxiliary input shaft, wherein one of the sliding collar and the auxiliary input shaft has a non-circular cross-section and another of the sliding collar and the auxiliary input shaft has a matching non-circular orifice, such that the sliding collar and the auxiliary input shaft rotate synchronously with each other while having the ability to slide axially relative to each other,
      - iv. a link assembly comprising
        - d) a link pivoting on the sliding collar and the crank pin collar through the notch
        - e) a sliding collar pivot pin on one end of the link and pivoting the crank pin collar with
        - f) a crank pin collar pivot pin on another end
    - m) at least one thrust bearing that is co-axially placed in contact with the sliding collar, such that an external force applied on the thrust bearing causes an axial displacement of the thrust bearing and the sliding collar with respect to the auxiliary input shaft, which alters the offset distance by moving the crank pin collar together with the crank pin along the crank pin collar shaft,

- n) a slotted rack holder comprising one or more racks, which is restricted to only move along a direction of a longitudinal axis of the one or more racks, and a crank pin slot for receiving the crank pin, with a longitudinal axis of the crank pin slot orthogonal to the one or more racks,

B) at least one angular velocity module comprising:

- d) an input shaft,
- e) at least one driving Geneva pin wheel mounted on the input shaft and driving
- f) at least one driven Geneva slot wheel mounted co-axially on the auxiliary input shaft, at a fixed orientation to the axis of the crank pin shaft and

C) at least one rectifier module comprising:

- a) a pinion engaged with the one or more racks, and mounted on
- b) a pinion shaft through
- c) a computer-controlled clutch, a one-way bearing, or a ratchet mechanism;

arranged such that a uniform rotation of the driving Geneva pin wheel via the input shaft, causes a non-uniform angular velocity of the auxiliary input shaft via the driven Geneva slot wheel, causing the crank pin to revolve the auxiliary input shaft reciprocating the one or more racks substantially along a longitudinal direction of the one or more racks at a substantially constant velocity and slowing down briefly during direction reversal and accelerating to the constant velocity, where a magnitude of the reciprocation is proportional to the offset distance of the crank pin and the auxiliary input shaft, and this reciprocation of the rack causes an alternating rotation of the pinion and the rotation of the pinion is converted to a unidirectional rotation of an output gear, or output sprocket mounted on the pinion shaft via the computer controlled clutch, the one way bearing or the ratchet mechanism.

18. An infinitely variable transmission comprising:

A) at least one scotch yoke module comprising:

- a. a crank pin revolving around
- b. a notched auxiliary input shaft, at an offset distance between longitudinal axes of the crank pin and the auxiliary input shaft that remain parallel to each other, and the offset distance that can be altered from zero when the crank pin is co-axial to the auxiliary input shaft to a non-zero real number by displacing the crank pin along a radial slot of
- c. an input disk rigidly mounted on the auxiliary input shaft, by

B) a crank pin displacement mechanism comprising:

- a. a sliding collar disposed co-axially with the auxiliary input shaft with a feature preventing relative angular displacement while allowing relative translation,

- b. a link assembly comprising
  - i. a link pivoting the crank pin through the notch
  - ii. a crank pin pivot pin on one end and pivoting the sliding collar about
  - iii. a sliding collar pivot pin on another end of the link,
- c. at least one thrust bearing that is co-axially placed in contact with the sliding collar, such that an external force applied on the thrust bearing causing an axial displacement of the thrust bearing along with the sliding collar with respect to the auxiliary input shaft, alters the offset distance by moving the crank pin along the radial slot of the input disk,
- d. a slotted rack holder comprising one or more racks, which is restricted to only move along a direction of the longitudinal axis of the one or more racks, and a crank pin slot for receiving the crank pin, with a longitudinal axis of the crank pin slot orthogonal to the one or more racks,
- C) at least one angular velocity module comprising:
  - a. an input shaft,
  - b. one or more driving circular or non-circular gear mounted on the input shaft and driving
  - c. at least one driven circular or non-circular gear that is mounted free to spin on a fixed shaft, where the driven non-circular gear further functions as a carrier of a planetary gear system, with
  - d. at least one free to spin planet gear meshing with a stationary sun gear mounted on the fixed shaft and is axially attached to
  - e. a primary cam that is operably engages with
  - f. a secondary cam that is mounted on the auxiliary input shaft and
- D) at least one rectifier module comprising:
  - a. a pinion engaged with the rack, and mounted on
  - b. a pinion shaft through
  - c. a computer-controlled clutch, a one-way clutch or a ratchet mechanism

arranged such that a uniform rotation of the driving non-circular gear via the input shaft, causes a non-uniform angular velocity of the auxiliary input shaft via the driven non-circular gear and the planetary gear system, causing the crank pin to reciprocate the rack substantially along a longitudinal direction of the rack at a substantially constant velocity and slowing down briefly during direction reversal and accelerating to the constant velocity, where a magnitude of the reciprocation is proportional to the offset distance of the crank pin

and the auxiliary input shaft, and this reciprocation of the rack causes an alternating rotation of the pinion and this alternating rotation of the pinion is converted to a unidirectional rotation of the pinion shaft via the computer-controlled clutch, the one-way clutch or the ratchet mechanism.

19. An infinitely variable transmission comprising:

A) at least one scotch yoke module comprising:

- a. a crank pin revolving around
- b. a notched auxiliary input shaft, at an offset distance between longitudinal axes of the crank pin and the auxiliary input shaft that remain parallel to each other, and the offset distance that can be altered from zero when the crank pin is co-axial to the auxiliary input shaft to a non-zero real number by displacing the crank pin along a radial slot of
- c. an input disk rigidly mounted on the auxiliary input shaft, by

B) a crank pin displacement mechanism comprising:

- a. a sliding collar disposed co-axially with the auxiliary input shaft with a feature preventing relative angular displacement while allowing relative translation ,
- b. a link assembly comprising
  - i. a link pivoting the crank pin through the notch
  - ii. a crank pin pivot pin on one end and pivoting the sliding collar about
  - iii. a sliding collar pivot pin on another end of the link,
- c. at least one thrust bearing that is co-axially placed in contact with the sliding collar, such that an external force applied on the thrust bearing causing an axial displacement of the thrust bearing along with the sliding collar with respect to the auxiliary input shaft, alters the offset distance by moving the crank pin along the radial slot of the input disk,
- d. a slotted rack holder comprising one or more racks, which is restricted to only move along a direction of the longitudinal axis of the one or more racks, and a crank pin slot for receiving the crank pin, with a longitudinal axis of the crank pin slot orthogonal to the one or more racks,

C) at least one angular velocity module comprising:

- a. an input shaft,
- b. one or more driving circular or non-circular gear mounted on the input shaft and driving
- c. at least one driven circular or non-circular gear that is mounted free to spin on a fixed shaft, where the driven non-circular gear further functions as a carrier of a planetary gear system, with

- d. at least one free to spin planet gear meshing with a stationary ring gear that is mounted on a frame and is axially attached to
  - e. a primary cam that is operably engages with
  - f. a secondary cam that is mounted on the auxiliary input shaft and
- D) at least one rectifier module comprising:
- a. a pinion engaged with the rack, and mounted on
  - b. a pinion shaft through
  - c. a computer-controlled clutch, a one-way clutch or a ratchet mechanism

arranged such that a uniform rotation of the driving non-circular gear via the input shaft, causes a non-uniform angular velocity of the auxiliary input shaft via the driven non-circular gear and the planetary gear system, causing the crank pin to reciprocate the rack substantially along a longitudinal direction of the rack at a substantially constant velocity and slowing down briefly during direction reversal and accelerating to the constant velocity, where a magnitude of the reciprocation is proportional to the offset distance of the crank pin and the auxiliary input shaft, and this reciprocation of the rack causes an alternating rotation of the pinion and this alternating rotation of the pinion is converted to a unidirectional rotation of the pinion shaft via the computer-controlled clutch, the one-way clutch or the ratchet mechanism.

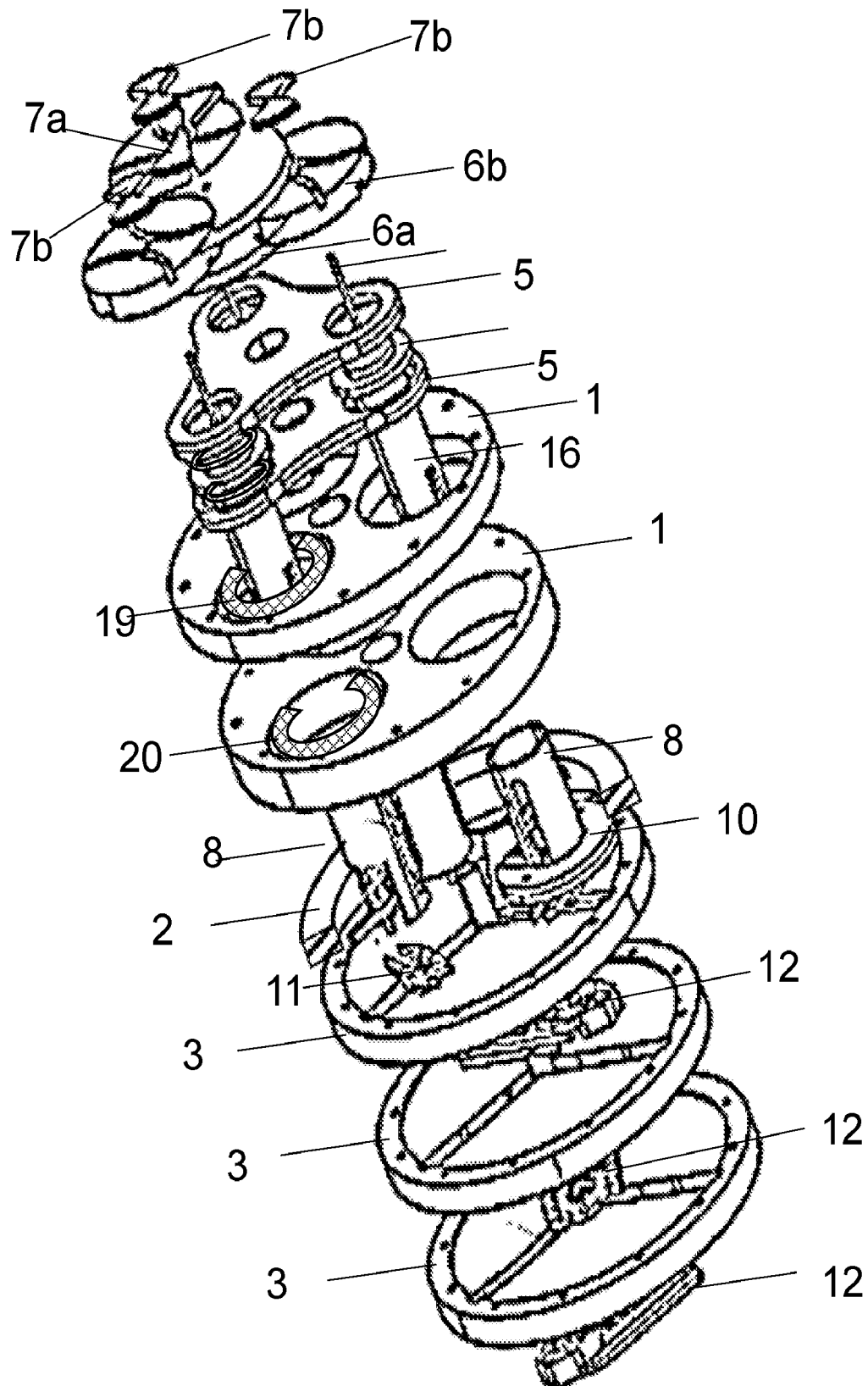
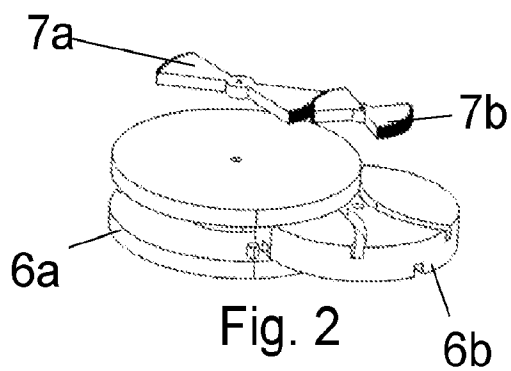


Fig.1



2/11

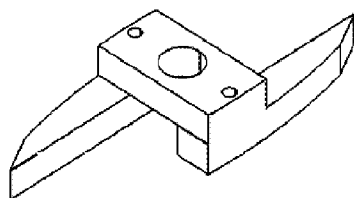
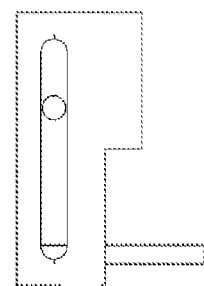
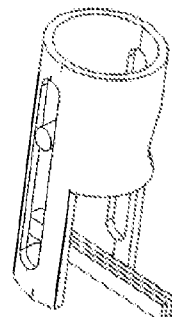
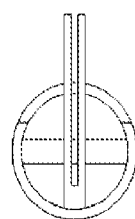
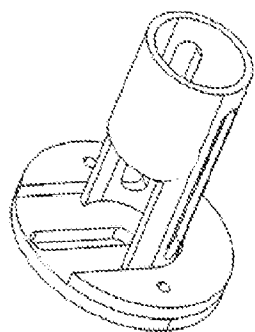
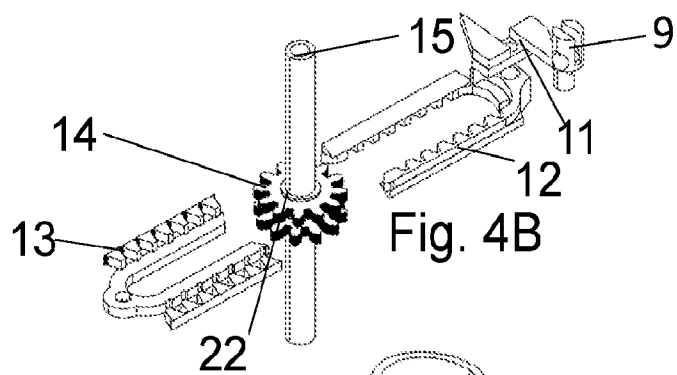
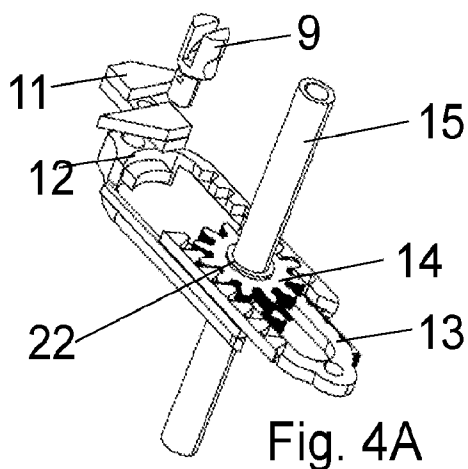
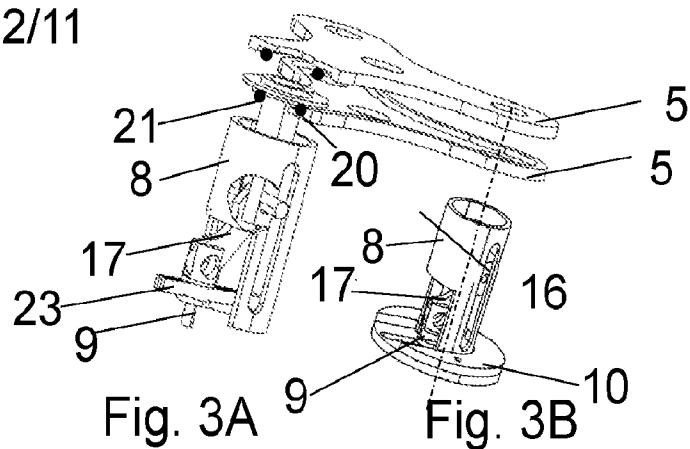


Fig. 7b

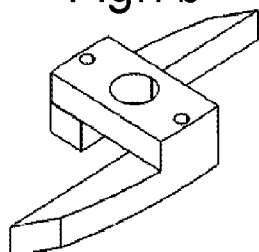
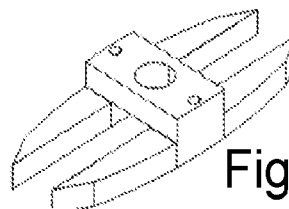


Fig. 7c



3/11

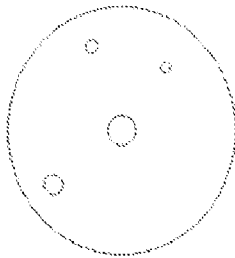


Fig. 8A



Fig. 8B

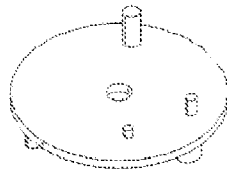


Fig. 8C

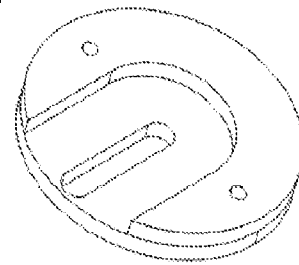


Fig. 9

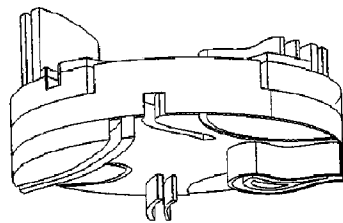


Fig. 10A

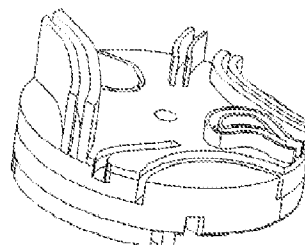


Fig. 10B

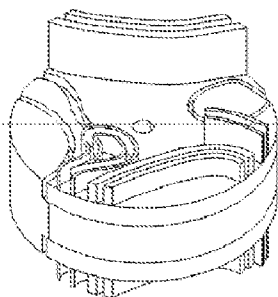


Fig. 10C

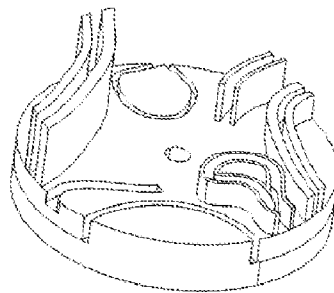


Fig. 11

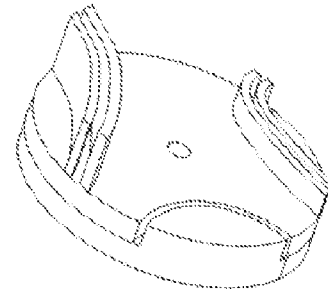


Fig. 12

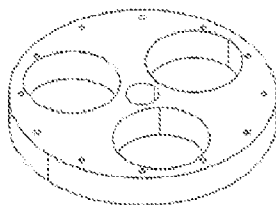


Fig. 13

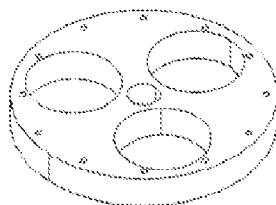


Fig. 14

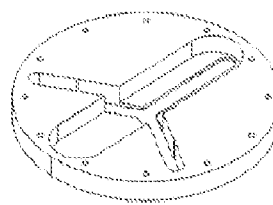


Fig. 15

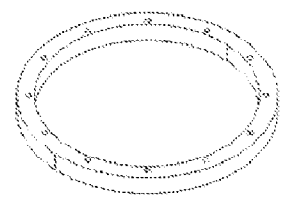


Fig. 16

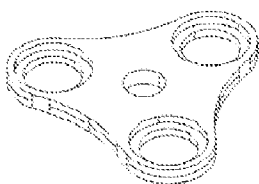


Fig. 17



Fig. 18



Fig. 19

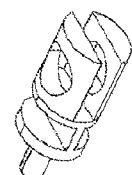


Fig. 20

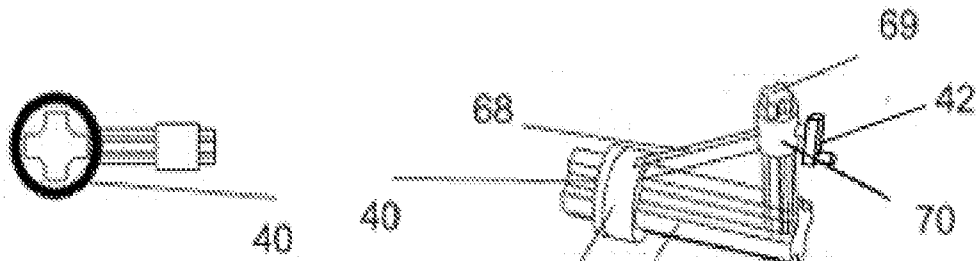


Fig. 21A

Fig. 21D

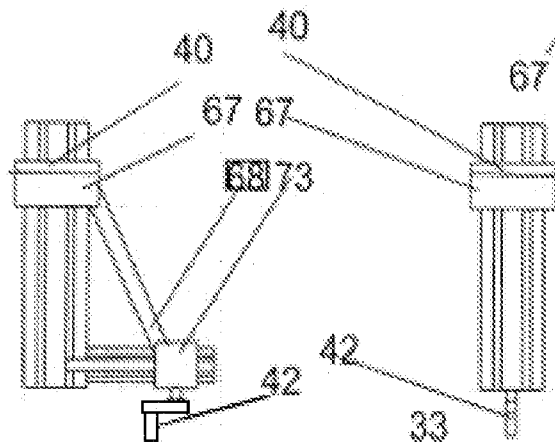


Fig. 21B

Fig. 21C

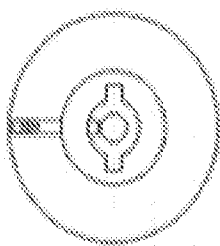


Fig. 22A

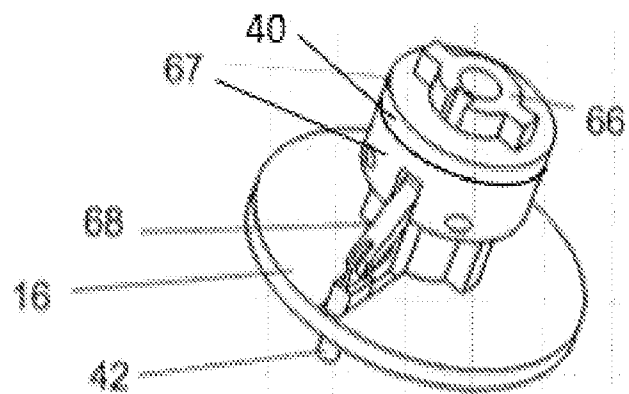


Fig. 22D

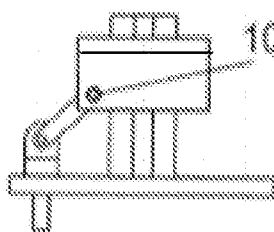


Fig. 22B

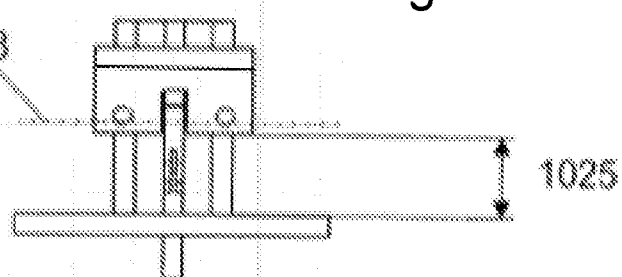


Fig. 22C

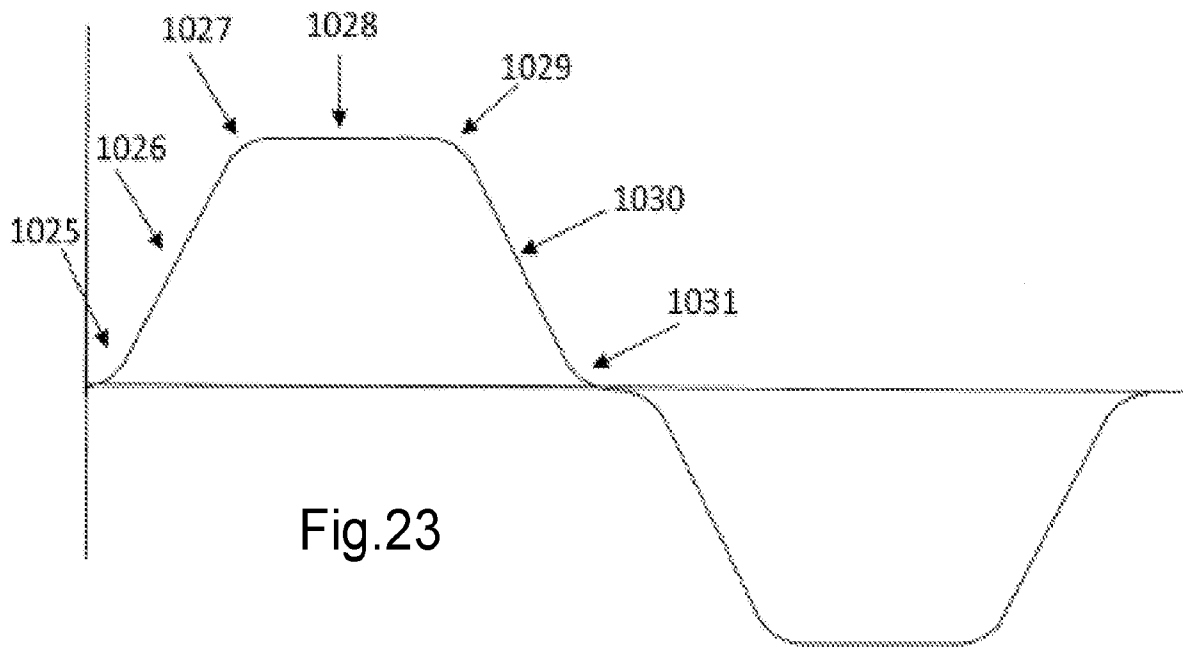


Fig.23

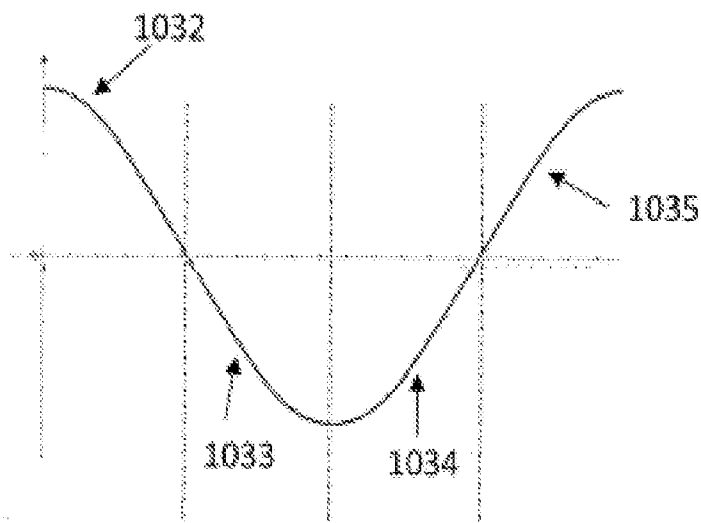


Fig.24

6/11

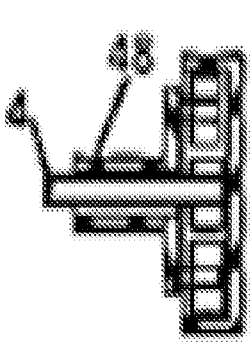


Fig. 25

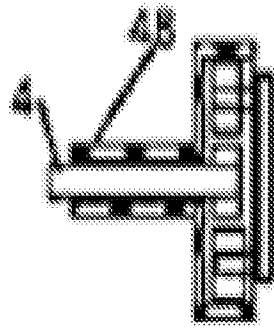


Fig. 26

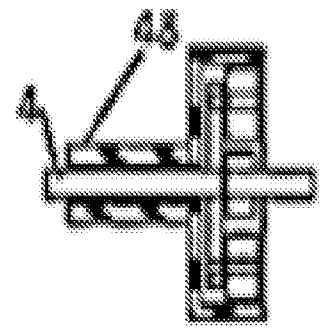


Fig. 27

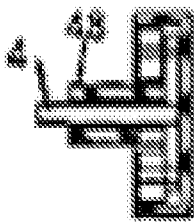


Fig. 28

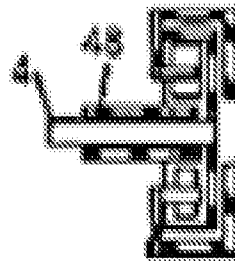


Fig. 29

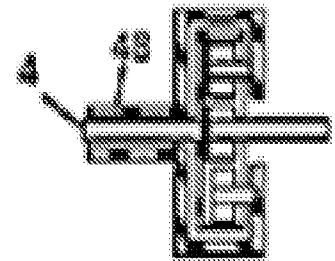


Fig. 30

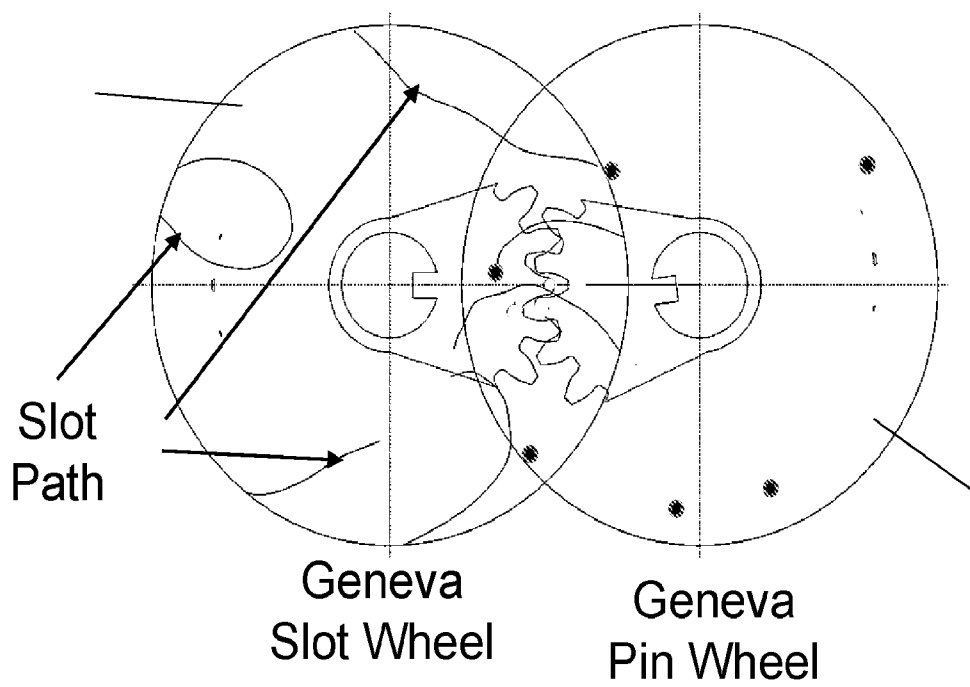


Fig. 31

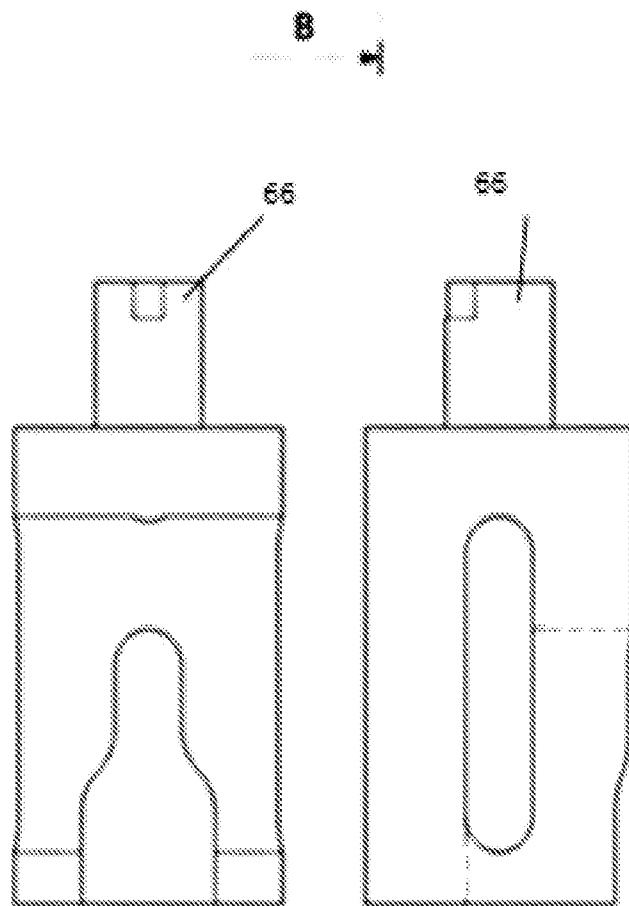
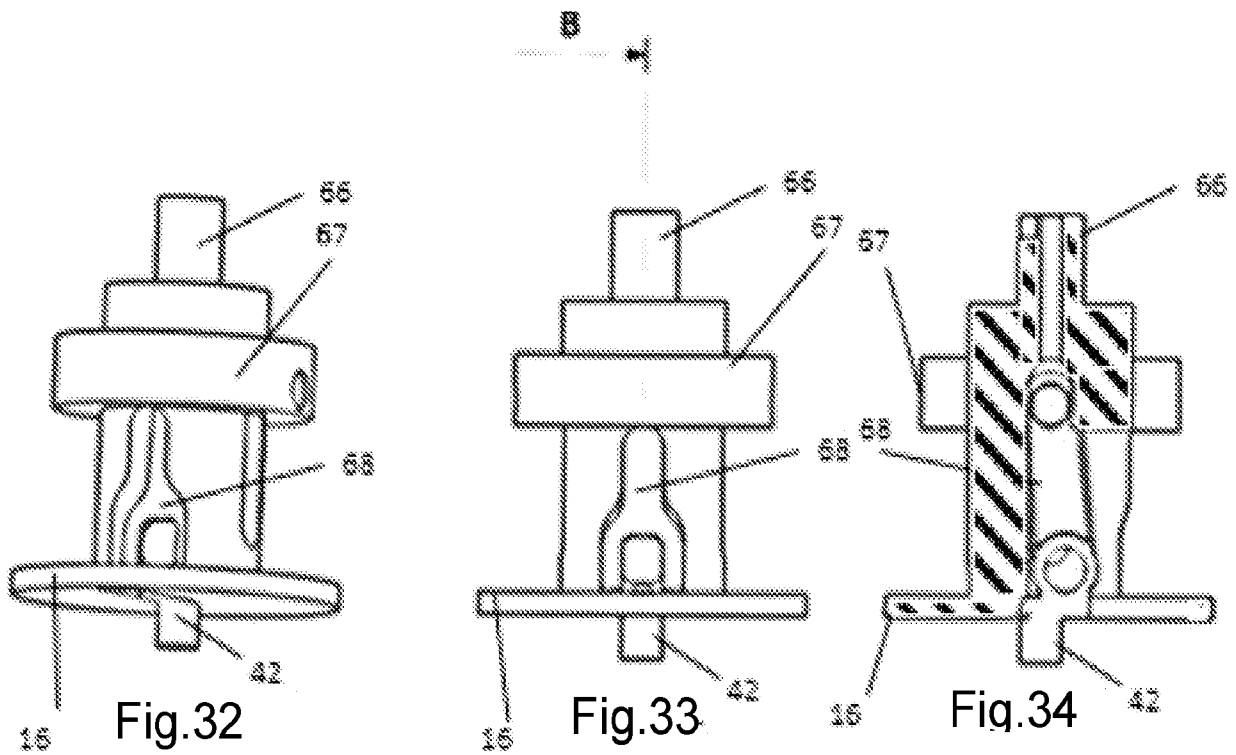


Fig.35

Fig.36

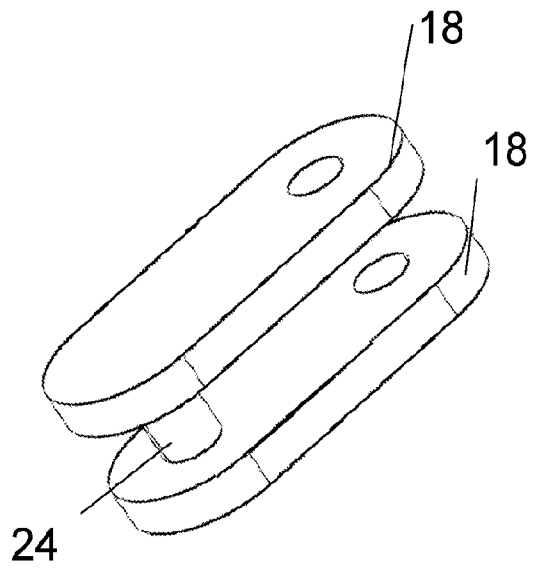


Fig.37

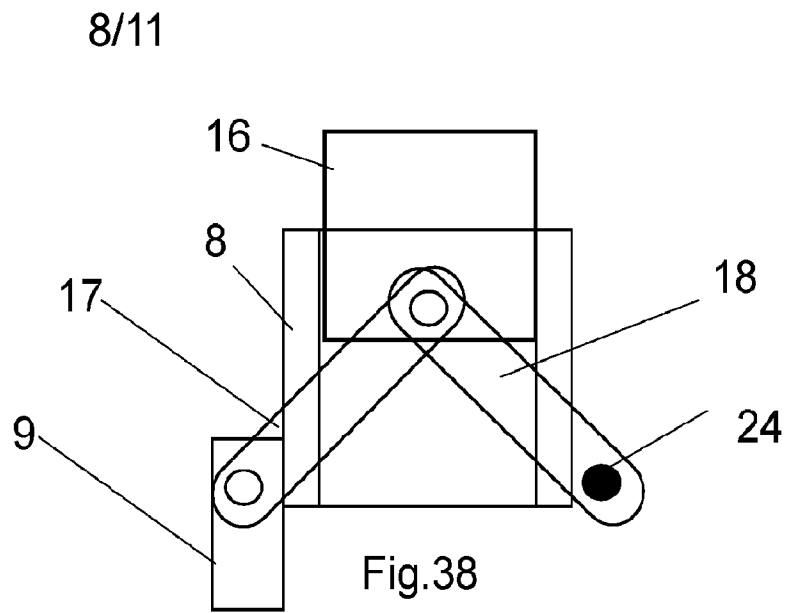


Fig.38

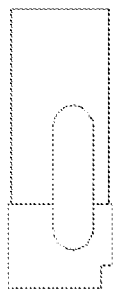


Fig.39A

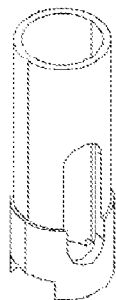


Fig.39D



Fig.39B



Fig.39C

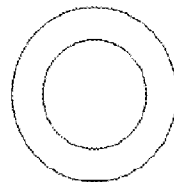


Fig.40A

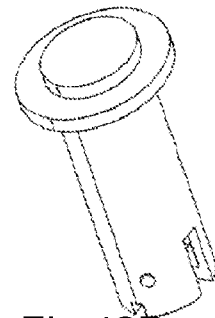


Fig.40D

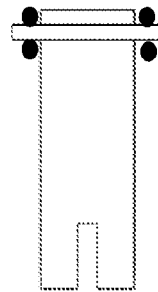


Fig.40B

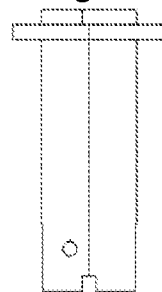


Fig.40C

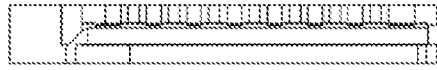


Fig.41A

9/11

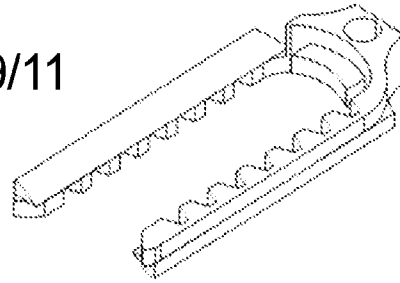


Fig.41E

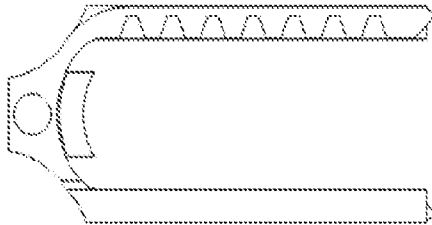


Fig.41B



Fig.41D

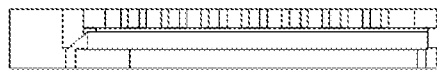


Fig.41C

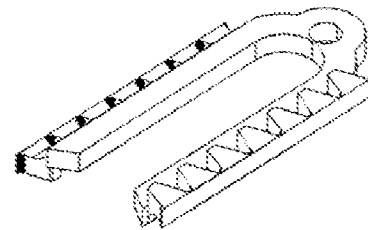


Fig.42E

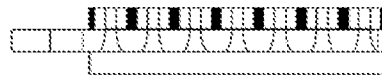


Fig.42A

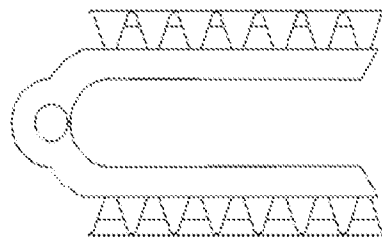


Fig.42B

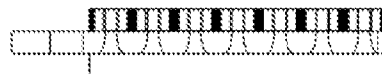


Fig.42C

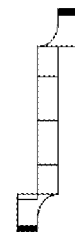


Fig.42D

10/11

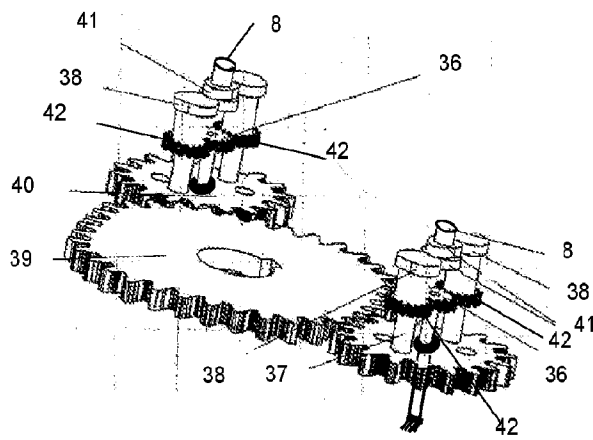


Fig. 43A

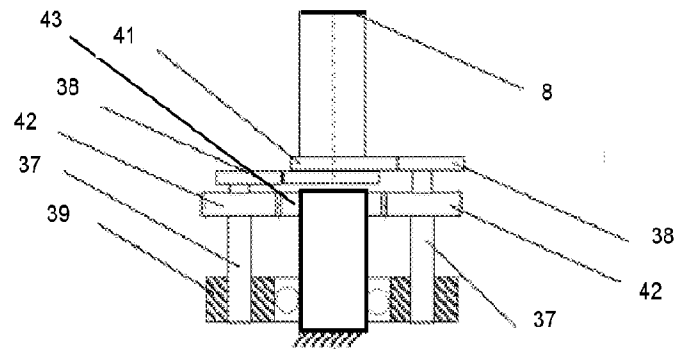


Fig. 43B

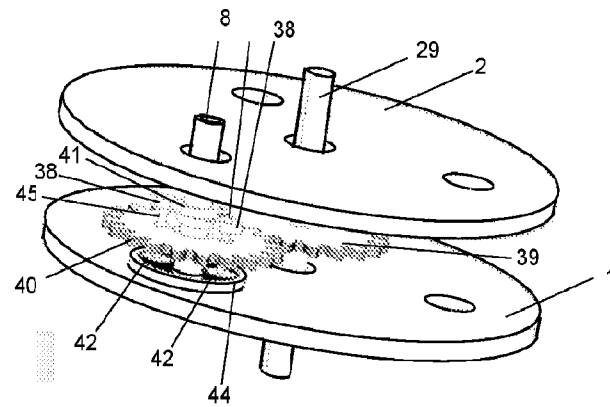


Fig. 44A

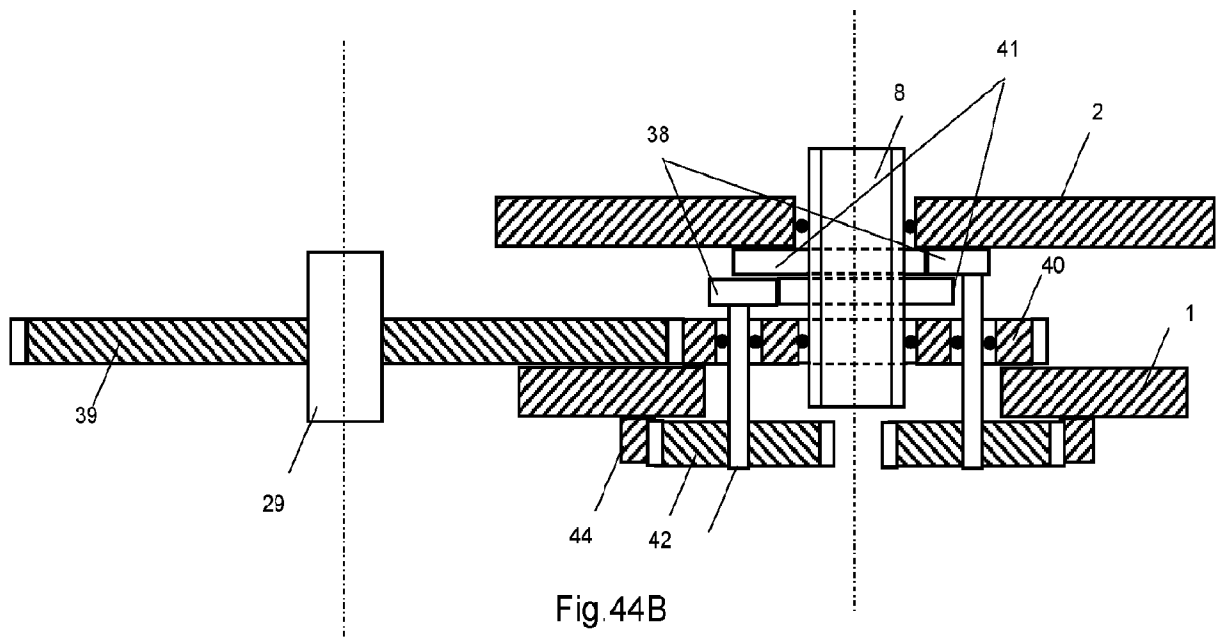


Fig. 44B

11/11

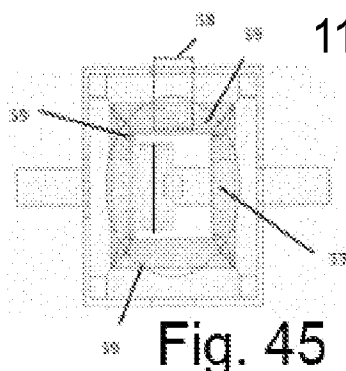


Fig. 45

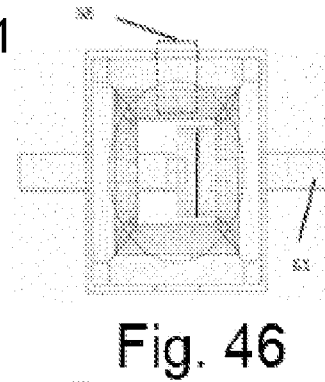


Fig. 46

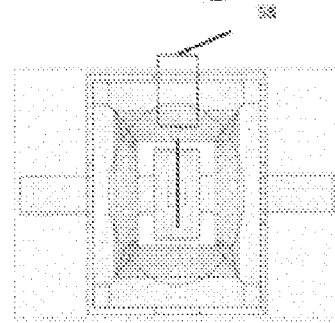


Fig. 47

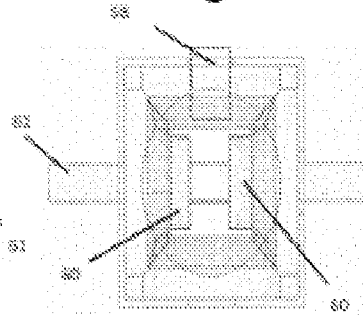


Fig. 48

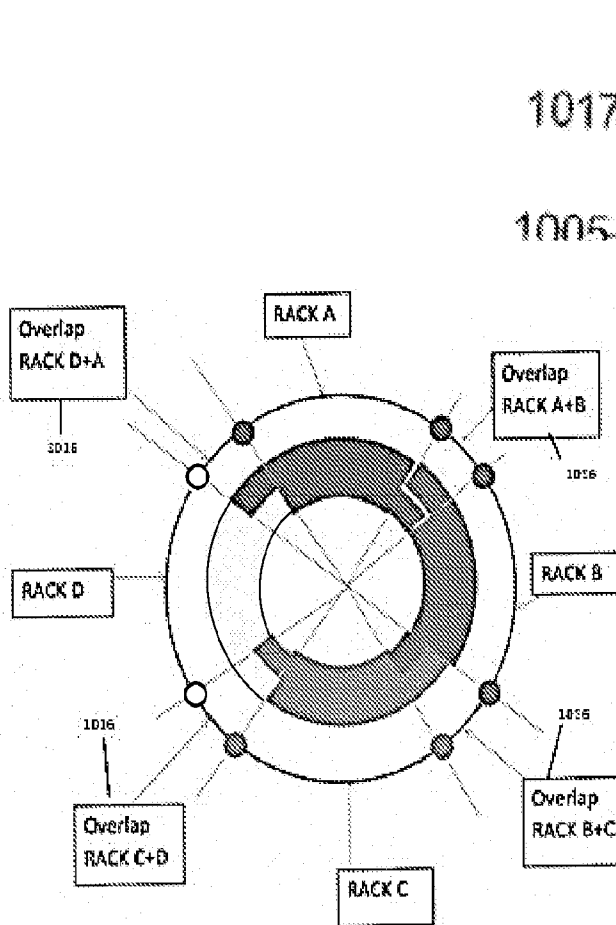


Fig. 50

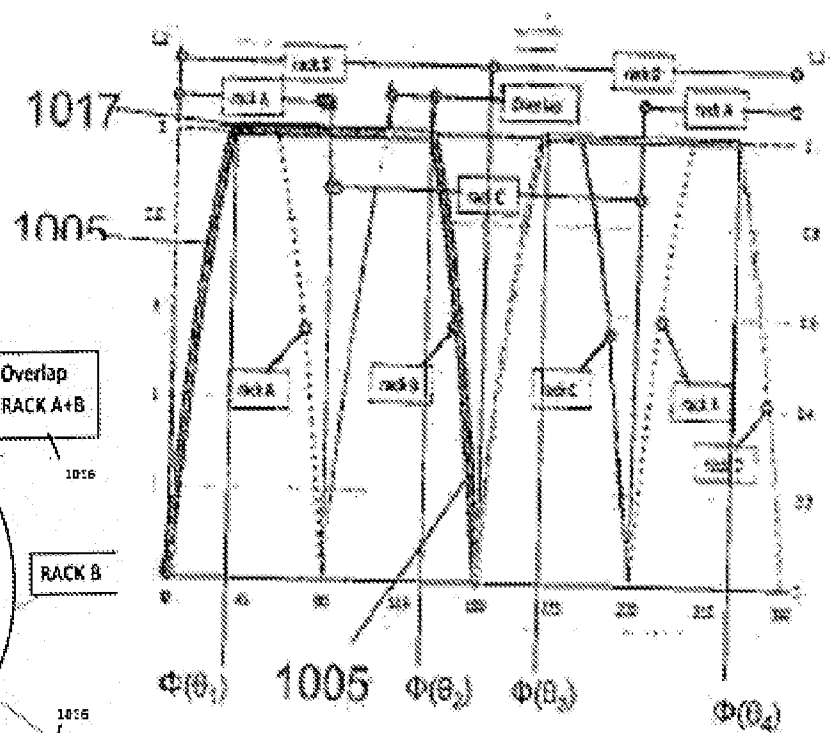


Fig. 49

## INTERNATIONAL SEARCH REPORT

International application No.

PCT/US 21/17984

A. CLASSIFICATION OF SUBJECT MATTER  
 IPC - F16H 29/20; F16H 29/00; F16H 27/06 (2021.01)  
 CPC - F16H 29/00; F16H 29/20; F16H 27/06

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)  
 See Search History document

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched  
 See Search History document

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)  
 See Search History document

## C. DOCUMENTS CONSIDERED TO BE RELEVANT

| Category* | Citation of document, with indication, where appropriate, of the relevant passages                           | Relevant to claim No. |
|-----------|--------------------------------------------------------------------------------------------------------------|-----------------------|
| A         | US 2019/0316662 A1 (Rajendran et al.) 17 October 2019 (17.10.2019), entire document especially Fig. 1-2, 66D | 1-19                  |
| A         | US 2008/0115623 A1 (Naude) 22 May 2008 (22.05.2008), entire document, especially Fig. 9; para [0001], [0041] | 1-19                  |
| A         | US 2015/026779 A1 (Rajendran et al.) 24 September 2015 (24.09.2015), entire document                         | 1-19                  |
| A         | US 5,906,134 A (Yamada) 25 May 1999 (25.05.1999), entire document                                            | 1-19                  |

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

☐ See patent family annex.

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"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

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Date of the actual completion of the international search

01 June 2021 (01.06.2021)

Date of mailing of the international search report

**JUN 15 2021**

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