LAPPEENRANTA UNIVERSITY OF TECHNOLOGY
LUT School of Energy Systems
LUT Mechanical Engineering

Tommi Nummelin

IMPROVEMENTS ON INFINITELY VARIABLE DRIVE DESIGN

Examiners: Professor Jussi Sopanen
D. Sc. (Tech.) Behnam Ghalamchi

TIIVISTELMÄ

Lappeenrannan teknillinen yliopisto
LUT Energiajärjestelmät
LUT Kone

Tommi Nummelin

Portaattoman mekaanisen vaihteiston kehitystyö

Diplomityö
2017

62 sivua, 40 kuvaa, 1 taulukko

Tarkastajat: Professori Jussi Sopanen
TkT Behnam Ghalamchi

Hakusanat: vaihteistot, polttomoottorit, planeettavaihde, portaaton muuttuva välityssuhde

Polttomoottori saavuttaa parhaan hyötysuhteensa kapealla kierrosnopeusalueella, minkä vuoksi sen jatkeeksi tarvitaan yleensä vaihteisto. Rajattoman määrän erilaisia välityksiä tuottavat portaattomat vaihteistot ovat kiinnostaneet keksijöitä vuosisatojen ajan, mutta yksittäistä teknisesti ylivoimaista, kustannustehokasta ja tilankäytöltään tehokasta ratkaisua ei ole vielä löytynyt. Ideaalinen ratkaisu olisi kaikkea edellä mainittua, ja tuottaisi positiivisten välitysten lisäksi myös vapaavaihteen ja peruutusvaihteen ilman erillisiä kytkimiä.

Työssä tutkitaan kirjallisuuskatsauksen omaisesti erilaisia kitkapintoihin perustuvia ja mekaanisia portaattomia vaihteistoja. Lopuksi esitellään uudenlaisen portaattoman mekaanisen vaihteiston konsepti, jota kehitetään eteenpäin VDI 2221 -tuotekehitysprosessia seuraten. Toimivuutta testataan pienoismallimittakaavan prototyypillä, jonka mittaustulokset analysoidaan. Tulosten pohjalta arvioidaan konseptin käyttökelpoisuutta.

ABSTRACT

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Improvements on Infinitely Variable Drive Design

Master's Thesis
2017

62 pages, 40 figures, 1 table

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Keywords: transmissions, engines, planetary gears, continuously variable transmission

Internal combustion engines are known to have a narrow power band, a range of operating speeds under which the engine is able to operate most efficiently. In most automotive applications, the engine is accompanied by a transmission with different gear ratios to make sufficient power and torque available over the range of vehicle speeds.

Stepless, continuously variable transmissions have had the interest of inventors for centuries, but a single technically superior, cost efficient solution has not yet emerged to overtake mechanical fixed-ratio transmissions in popularity. An ideal solution would be an infinitely variable transmission, producing all the needed positive ratios, neutral and reverse without needing separate clutches.

Different friction-based and mechanical continuously variable transmissions are studied with a literature review. A novel, fully mechanical continuously variable transmission concept is introduced and then developed following the steps of VDI 2221 product development guideline. Small scale functional prototype is built and tested to study the viability of the concept.

ACKNOWLEDGEMENTS

I would like to thank examiners professor Jussi Sopanen and D.Sc. Behnam Ghalamchi for their sage guidance, dearest Janita for putting up with me, and the unknown ancient Mesopotamian person who thought of mixing barley malts with hot water for fueling technological progress.

Tommi Nummelin
Lappeenranta 4.12.2017

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LIST OF SYMBOLS AND ABBREVIATIONS

CCW	Counter-clockwise
C_O	Permissible static load
CVT	Continuously Variable Transmission
CW	Clockwise
FGE	Fundamental Gear Entity
F_r amm.	Maximum permissible load
ICE	Internal Combustion Engine
IVD	Infinitely Variable Drive
IVT	Infinitely Variable Transmission
K_f	Load coefficient
M, T	Torque
N	Number of teeth on a gear
u	Transmission ratio
ω	Rotational speed

1 INTRODUCTION

The task of a transmission is to provide speed and torque conversions from prime mover to its output. In vehicular context this means controlled application of power of internal combustion engine (ICE) to driven wheels. Electric motors and steam engines can operate at any point between zero revolutions per minute and the high end of their power band without stalling, whereas at least a simple clutch mechanism is needed between ICE and the wheels it drives.

When either the velocity or the load (or both) of the vehicle vary, internal combustion engine has to work beyond its optimal operating point. In a conventional mechanical drivetrain with a set of fixed gear ratios, acceleration is simply impossible without raising the engine speed and thus leaving the “sweet spot”.

1.1 Background

Since the dawn of motoring, inventors and engineers have been trying to come up with efficient, cheap, constantly variable transmissions. Yet, if we look at a modern variable-diameter pulley continuously variable transmissions (CVT), it is immediately clear that their basic operating principle is identical to that of Milton Reeves’ early 20th century transmissions.

In a case of a midsized car, using a high efficiency CVT gearbox only reduces the fuel consumption by 3.0 per cent on an average compared to traditional automatic transmissions (National Academy of Sciences, 2015, p. 26).

1.2 Objectives and delimitations

The admittedly ambitious goal of this thesis is to construct a simple, fully mechanical and highly efficient continuously variable transmission. Research objective is to describe and study prior art related to CVTs and then develop a new concept.

A novel continuously variable transmission concept is introduced, then divided into modules and its key concepts tested with two proof-of-concept prototypes. The design process will be shown to the proof of concept stage. Basic functionality of the concept is tested.

In the proof of concept, only the most critical components will be dimensioned. The engineering process leading to chosen bearing sizes, gear profiles and such is not discussed.

2 CONTINUOUSLY VARIABLE TRANSMISSIONS

Internal combustion engine's properties and narrow power bands offer a straightforward explanation for the amount of inventing and capital that has been poured on trying to create a mechanically and financially feasible continuously variable gearbox. In modern automotive world, belt-driven designs or 'variators' have gained popularity despite their relatively low efficiency, as their shortcomings have been found an acceptable trade-off for the lower production cost and ease of implementing. Although they are sometimes seen as a marvel of modern technology, they have been in automotive use since the 1960s. Before that, the need to produce varying output speeds from constant input speeds such as water wheels has driven inventors early as Leonardo da Vinci in 1490 to develop gearless variable transmissions.

In this chapter, a brief review of prior art continuously variable transmissions is first given. Friction-based CVTs are inspected from mostly historic point of view, as they are deemed to be mostly outside the scope of this thesis. Planetary gearsets are then discussed from the viewpoint of enabling infinitely variable transmissions.

2.1 Friction-based CVTs

Continuously variable transmission (CVT) produces a changing transmission ratio between two finite limits without any steps. (Happian-Smith, 2001, p. 437) The most common CVT system is the previously mentioned Reeves drive, or variable-diameter pulley (Fischetti, 2006, p. 92). In both Reeves' CVT and the modern ones, the gear ratio is changed by squeezing a conical pulley together and pulling another's sheaves farther apart (US 785917 A, 1905, p. 3). This operation principle is well presented in Reeves' patent application drawing, figure 1.

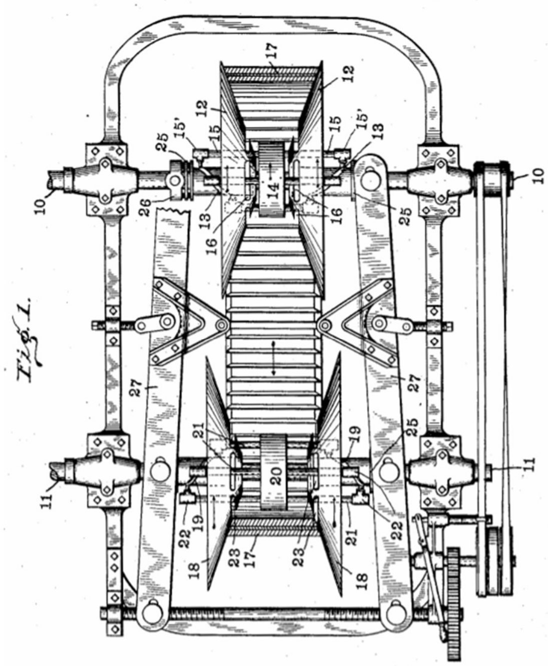

Figure 1. Early 20th century CVT patent by Milton Reeves (US 785917 A, 1905, p. 1)

Two V-belt pulleys are split so that they can be moved along their respective axes of rotation. A belt with a V-shaped cross section is tensioned against the pulleys by adjusting both of them at the same time, as the distance between the shafts does not change. In low-power applications, such as scooters, rubber-belt variable-diameter pulley CVT is ubiquitous. It is often accompanied by a centrifugal clutch acting as a starting device. Modern automotive applications usually involve a steel belt and a wet plate clutch or a torque converter.

In the context of automobiles, many friction-based power transmissions have been used, where in addition to the transmission ratio, the success rates have also varied. Early friction disk drives such as the Lambert drive (fig. 2) occupied a great deal of volume relative to the torque they can transmit.

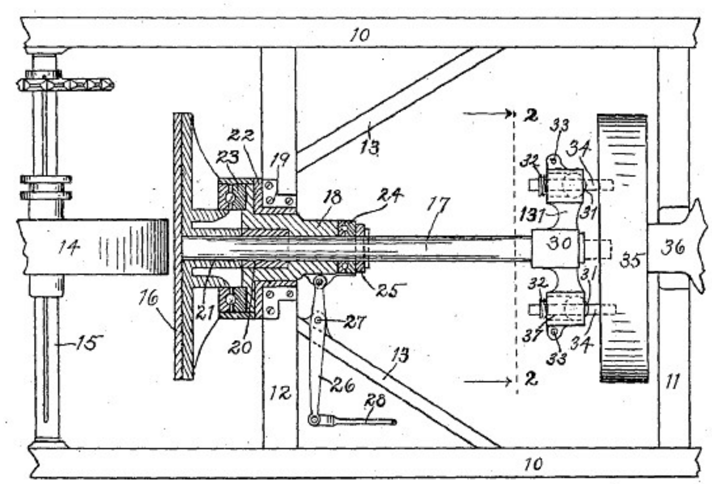

Figure 2. Lambert's friction gearing disk drive transmission (US 954977 A, 1910, p. 1)

Lambert drive was also able to act as an infinitely variable transmission, or produce ratios from negative (reverse) via zero (neutral) to positive (forward). In figure 3, input from the prime mover is on the shaft on the left, output shaft and its sliding friction wheel on the right.

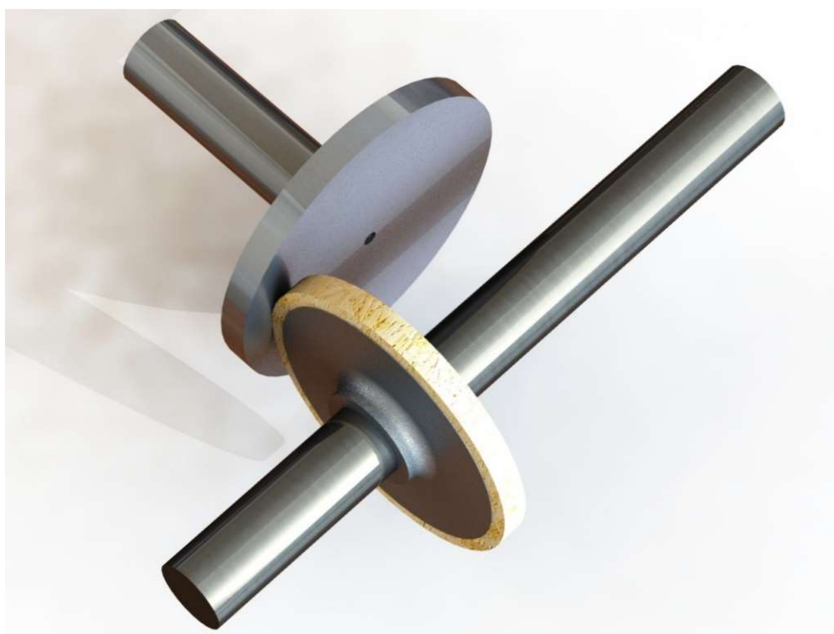

Figure 3. Lambert-style friction drive with aluminium driving disk and fiber-faced driven wheel

When their shared contact point is moved from one perimeter to the opposite extreme, all effective gear ratios between 1:1 and -1:1 are produced given that the wheels have the same diameter. A mechanism is needed to press the output wheel on the plate, so in its presence, it is also easier to produce the neutral gear by relieving the pressure on the wheel rather than running the wheel in the center of the input disk.

Despite lost space, short lifespan of stressed members, and significant power loss, friction gearing transmissions apparently managed to gain some success in the early 1900s motoring world. Some 20 to 30 years before the invention of synchromesh, the ease of use was likely to be quite an asset when comparing friction drives and manual transmissions.

There has also been some new-found interest in cone CVT, where a continuously variable output ratio is generated from two or more reversely conical, parallel shafts generate by sliding their contact point along the length of the shafts (fig. 4).

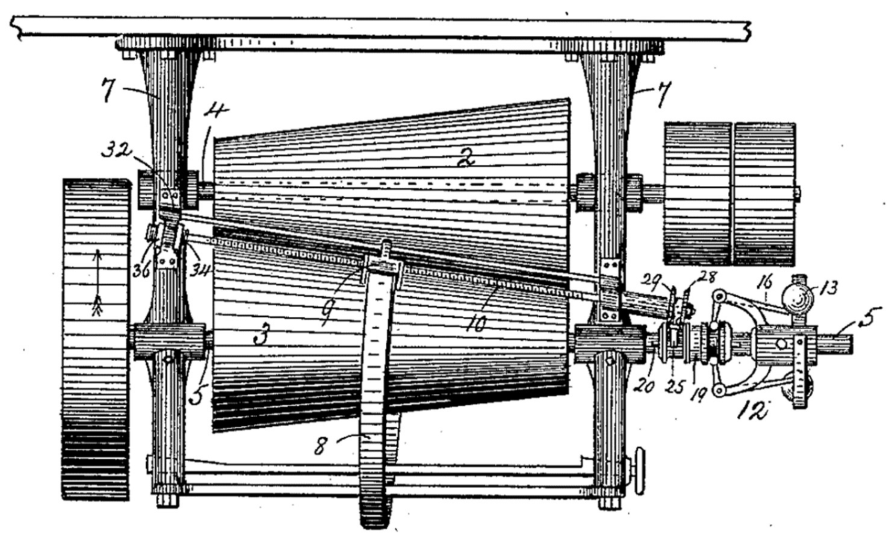

Figure 4. Evans' speed regulating device (US390216 A)

An Italian company called Warko (2008) has recently manufactured some prototypes and claimed novelty, but the operation principle of their power transmission is not much unlike that of Frank Evans' from 1888, seen above.

Some other types of gearboxes also exist where a torque is transmitted with help of friction through rolling elements providing different contact radii in different parts of the element. These include NuVinci planetary CVT (Fallbrook Technologies, 2017, p. 12) with its tilting axis planet spheres and Nissan Motors' Extroid CVT (Nissan, 1999). Nissan's toroidal transmission was the first of its kind in automotive use. The variator section has dual toroidal cavities with rollers to transmit power (fig. 5).

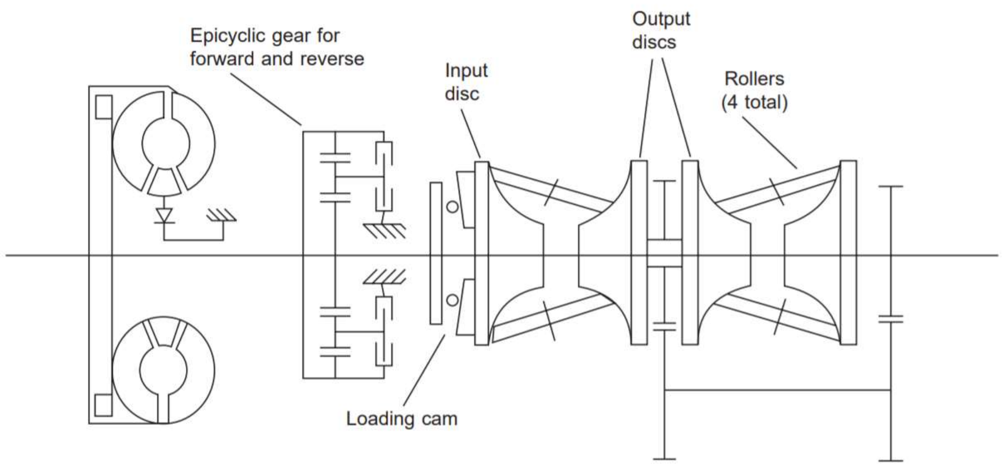

Figure 5. Nissan's toroidal transmission (Happian-Smith, 2001, p. 446)

Forward and reverse gearing is produced via an additional epicyclic gear. However, their elemental weaknesses and engineering challenges are that of their hundreds of years old predecessors: forcing torque through a friction-based contact point generates heat and thus losses, and the contact surface needed to transmit power without detrimental slipping ends up making the gearbox bulky. Since completely mechanical, gear-based, continuously variable transmission would have considerable benefits in terms of usability and efficiency, coming up with a structure that fulfills modern needs of power density and low cost would certainly be of value.

2.2 Mechanical CVTs

Friction-based CVTs are widely used in low-powered applications, such as scooters and lawnmowers. Next, a few methods of producing the same continuous variation with fully mechanical components are introduced.

2.2.1 Ratchet mechanisms

Along with varying belt pulley diameters and moving the contact point on a friction surface, varying transmission ratio can be achieved by changing length of a stroke that transmits power from one shaft to another. If a link performs reciprocating linear movement of varying stroke lengths, it cannot rotate a full revolution around a shaft without risking getting stuck in the dead center. Its trajectory must be limited.

This limitation has two consequences: first, there needs to be a mechanism in some point of the system which only allows power transmission in one direction and moves freely in the other. This can be achieved by a ratchet or a one-way bearing (fig. 6). The second limitation is that the transmission of power will always be intermittent to at least some degree.

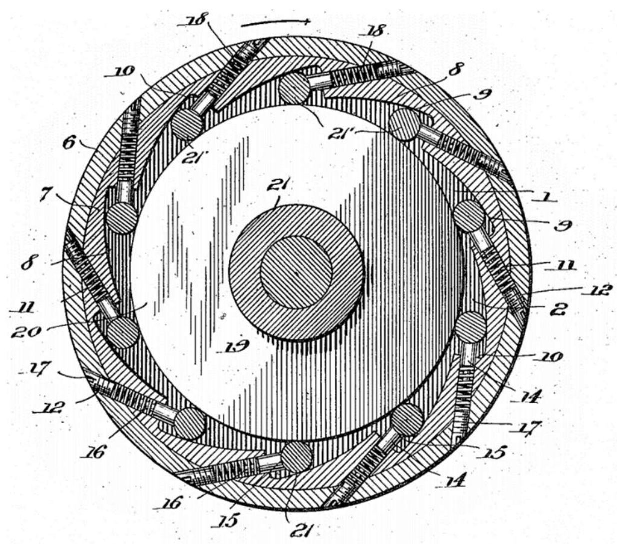

Figure 6. Cutaway drawing of the general workings of a one-way bearing in an early patent (US 1142574 A, 1915, p. 2)

One-way bearings (or freewheel clutches) are a simple construction: there are rolling elements between the hub and an outer race, which are pinned between the surfaces when the speed difference is in one direction, and roll freely when they rotate in another. Some freewheel clutches have figure-eight shaped sprags wedging between the races. They are often used in automatic transmissions as a method of allowing the transmission to shift gears smoothly under load, and are called sprag clutches.

Dozens of patents have been applied for a continuously variable transmission using a lever mechanism. Let us consider a ratcheting CVT of probably the most common type (fig. 7)

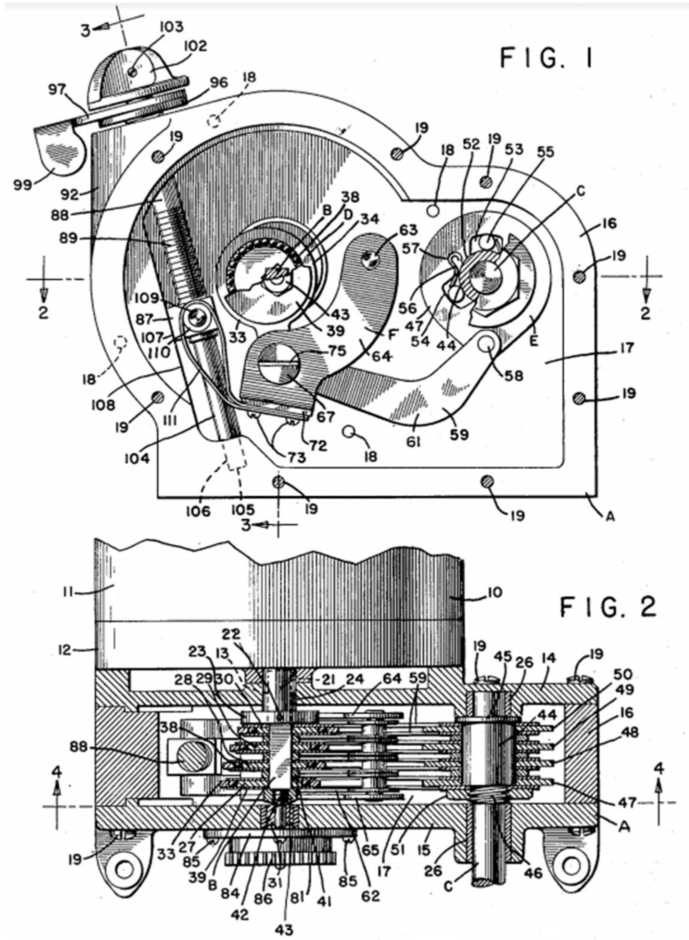

Figure 7. Variable speed power transmission invented by Sterling Stageberg (US 2691896 A, 1954, p. 1).

Adjustable output speed is achieved by changing the distance that one-way clutches rotate the output shaft. Input shaft drives an eccentric connecting rod which pushes the main link connected to the one-way clutch on the output shaft. By moving the control link, the direction of throw of the connecting rod is altered from vertical toward horizontal, making the stroke on the output shaft clutch longer.

This type of transmission has been mass produced for decades by Zero-Max, Inc. It is mostly marketed as an economical secondary drive and a clutch, since it is capable of producing a zero output. Input speeds can be up to 2,000 rpm. (Zero-Max, 2017) Educated guess for why the input speed has been restricted so low would be the fundamental limitation described earlier: despite a four-link structure and redundancy, the transmission will always produce ripple in the discontinuity when one ratcheting phase ends and another takes up.

Zero-Max transmission could be seen as a manual ratcheting CVT. There are also methods to automatically, yet mechanically, move the pivoting point that controls the stroke length.

2.2.2 Constantinesco mechanical torque converter

In the context of modern drivetrains, torque converter usually means a type of fluid coupling normally used to replace mechanical clutch in automatic transmission. However, several rarely used mechanical systems have also been used to multiply torque and to act as an equivalent of a reduction gear. One particularly interesting mechanical torque converter is by a Romanian engineer and inventor George Constantinesco (fig. 8).

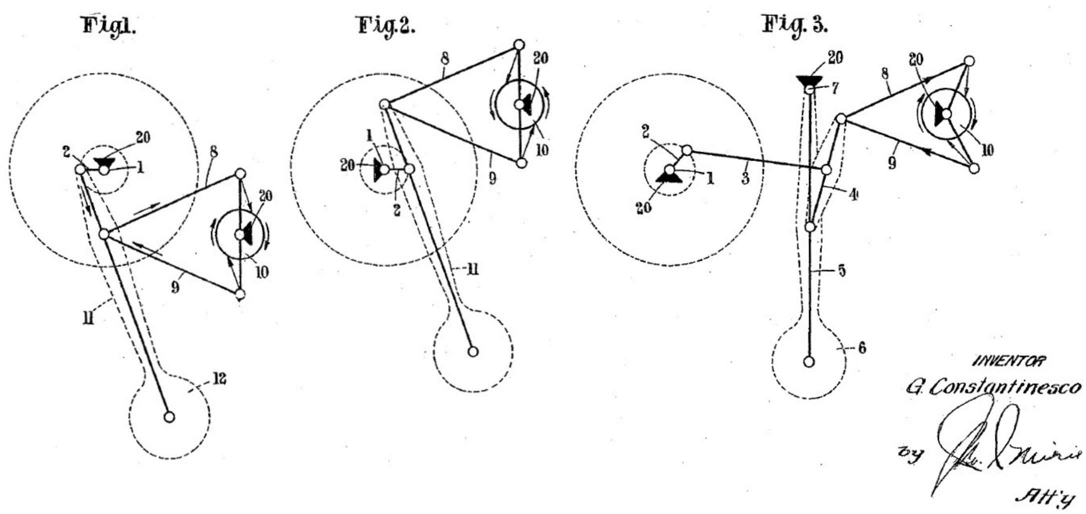

Figure 8. Operating principle of Constantinesco's mechanical torque converter (US 1542668, 1925, p. 1)

As the patent application states: "The object of the invention is to transmit power from the engine to the driven shaft in such a manner that increased resistant torque at the driven shaft may result in an increase of engine speed, so that the power developed by the engine does not unduly decrease with increased resistance." (US 1542668, 1925, p. 11)

This linkage acts between the prime mover *1* and its output shaft *10* driving it via a pendulum *11* with a heavy bob *12*. Output shaft is rotated with a ratcheting mechanism which allows for varying length of strokes to turn itself.

Now, the working principle of the mechanism might not be obvious for the uninitiated. As the resistant torque increases at the driven shaft, the prime mover is about to slow down. While this happens, the pendulum’s oscillation is affected and its swings become shorter. This, then, results in shorter strokes at the ratcheting mechanism; less work is extracted from a single revolution of the prime mover’s shaft. The virtual fulcrum of the driving levers move along resulting in reduced drive ratio and higher output torque.

Typical behaviour of the system can be seen in figure 9.

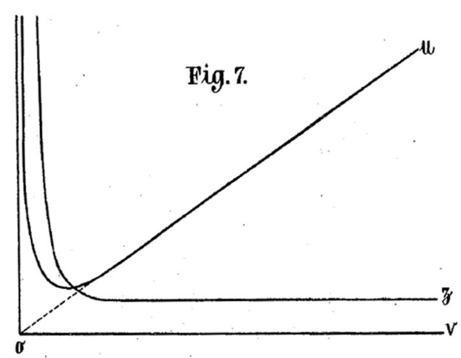

Figure 9. Output speed rising steadily whereas input speed eventually reaches an equilibrium state (US 1542668, 1925, p. 2).

When torque of the prime mover is kept constant, relative values for the speed of the prime mover, torque on the driven shaft and speed of the driven shaft behave as shown.

The mechanism reacts to increased resistance fully automatically and there is not really a way to control it. It can be braked, but speeding it up would prove more difficult. As the masses, pendulum length and other dimensions have to be carefully chosen for each engine power, vehicle mass and average load, it is hard to see how this mechanical torque converter could catch on in usual applications.

2.2.3 Ivanov's toothed CVT

Ivanov (2012) presents a gear differential with two degrees of freedom which acts as a self-regulating continuously variable transmission. This adaptive mechanical transmission is an interesting concept, as it provides a continuously variable ratio with constantly meshing gears, without an external governing device. The principle of this gear arrangement is shown in figure 10.

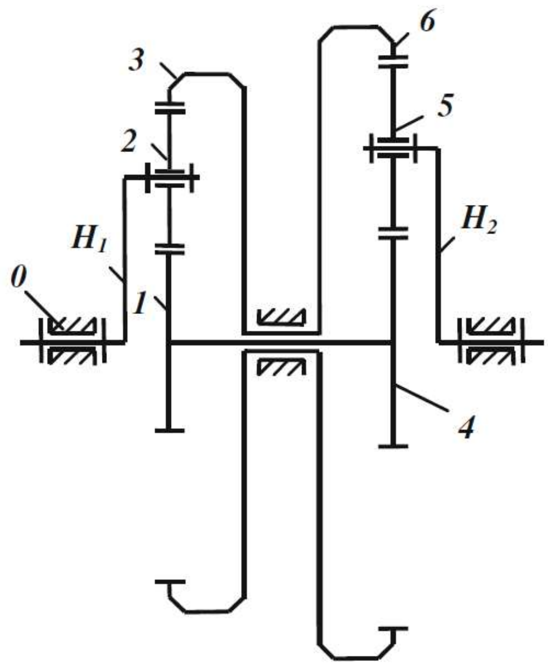

Figure 10. (Ivanov, 2012, p. 3)

Input carrier H_1 drives output carrier H_2 through two planetary arrangements. Ring gears 3 and 6 are tied together, and so are sun gears 1 and 4. External active motive moment M_{H1}, such as an ICE, acts on H_1, and the external active moment of resistance M_{H2} acts on output H_2. Components form a kinematic chain with two degrees of freedom with the closed four-link contour. In state of equilibrium, the principle of virtual work holds true, and it can be stated that input and output power have a total sum of zero. Ivanov then derives similar equilibrium state equations for each satellite, finally gaining output speed.

$$\omega_{H2} = \frac{M_{H1}\omega_{H1}}{M_{H2}} \tag{1}$$

In equation 1, ω_{H2} is the angular velocity of the output carrier, expressed with input speed ω_{H1} and ratio of input torque to output load, $M_{H1}:M_{H2}$.

At a constant input power P_{H1}, the output angular speed is proportionally dependent on a variable output moment of resistance M_{H2}. In other words, the system finds equilibrium by providing a higher ratio when the resistance at wheels decreases. This happens, as mentioned, without any external help, which is one of the defining features outlined in the patent for this gear differential transmission (DE 202012101273 U1, 2012, p. 1).

Transfer ratio u is inverse to the ratio input torque and output resistance.

$$u = \frac{M_{H2}}{M_{H1}} = \frac{\omega_{H1}}{\omega_{H2}} \tag{2}$$

This leads to a maximum transfer ratio which is output resistance divided by input torque. Ivanov’s adaptive gearbox has a lot of advantages, including simple construction and probably low vibration. However, controlling the ratio is often desired in vehicle applications, even if it is not truly necessary. Mechanically adaptive gear differential would also require a clutch, because it cannot produce a neutral ratio.

2.2.4 Generally about swashplate-based arrangements

Next, a few swashplate-based concepts are introduced.

Swashplate is a device used to translate rotating movement into reciprocating linear motion or vice versa. The structure has a tilted disk rotating with a central shaft. This disk has another disk, a follower, which shares the angle of the first disk but does not turn with the shaft. The follower disk's edge oscillates along the driving shaft's length, creating a reciprocating motion in the shafts attached to it. Simple swashplate-driven piston pump structure is presented in figure 11, albeit without a follower disk.

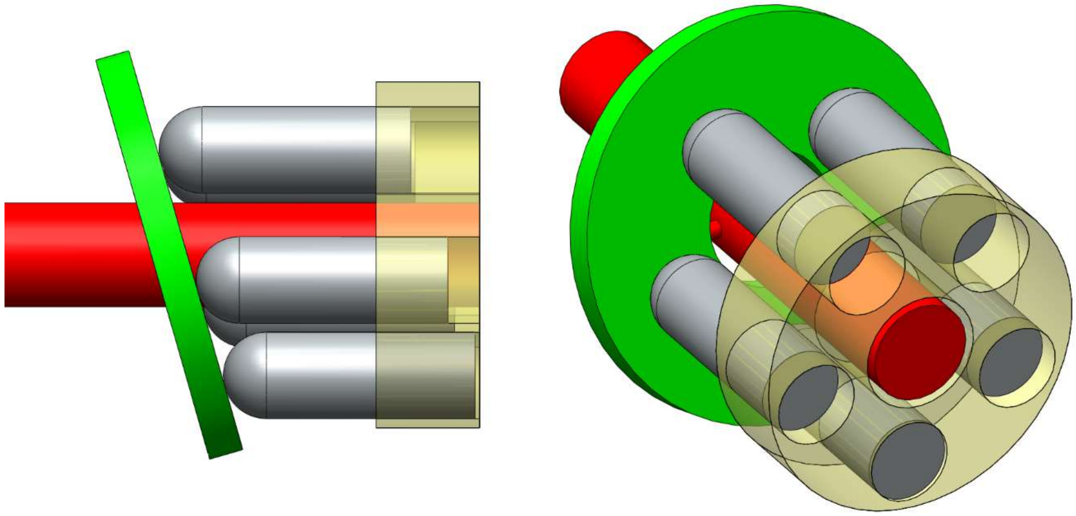

Figure 11. Axial piston pump with a swashplate (green) and a driving central shaft (red)

Swashplates can most often be seen in axial piston pumps and in helicopter rotors, where they provide controlled cyclic pitch and thus selective lift and ”steer” in pilot's chosen direction. One of the earliest applications was an attempt to replace crankshafts and connecting rods on an internal combustion engine (US 1409057, 1922).

Tilting mechanism of a swashplate is not a trivial design task, but it is not very difficult to overcome, either. Considering its fairly simple nature, it is inevitable someone is going to try using it as a source of variation in a CVT. Often the idea is to vary the stroke length and then convert this reciprocating motion into rotational motion. One version this idea can be seen described in Michael Allen’s patent (WO 2014/023926 A1, 2014, p. 1), in figure 12.

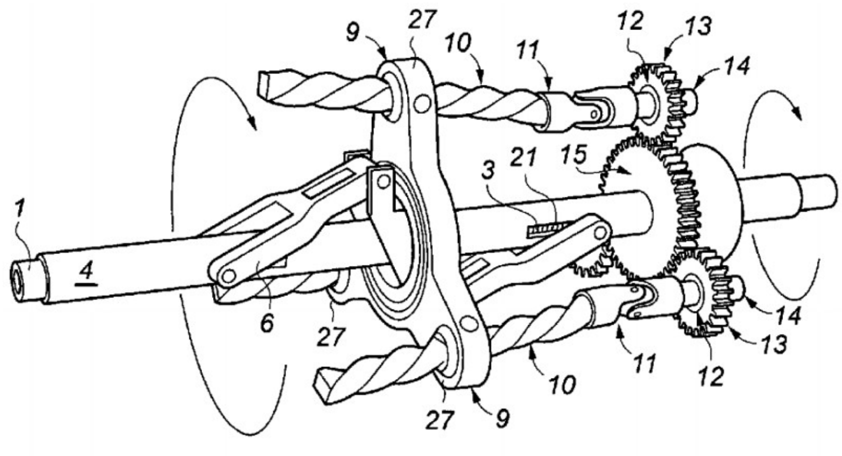

Figure 12. Allen's CVT as described in his patent

In Allen's patent, reciprocating members comprise a helical form which converts linear motion of the swashplate linkage into rotational motion. Main shaft rotates a sun gear, reciprocating members drive planet gears and the output is on the ring gear. Reportedly, a miniature prototype has been built to demonstrate the idea and prove the concept.

Allen's transmission is not completely unique in its working principle. Gustav Eggert came up with a similar method of motion conversion (fig. 13) in his 1932 patent (US 1869189, 1932, p. 1).

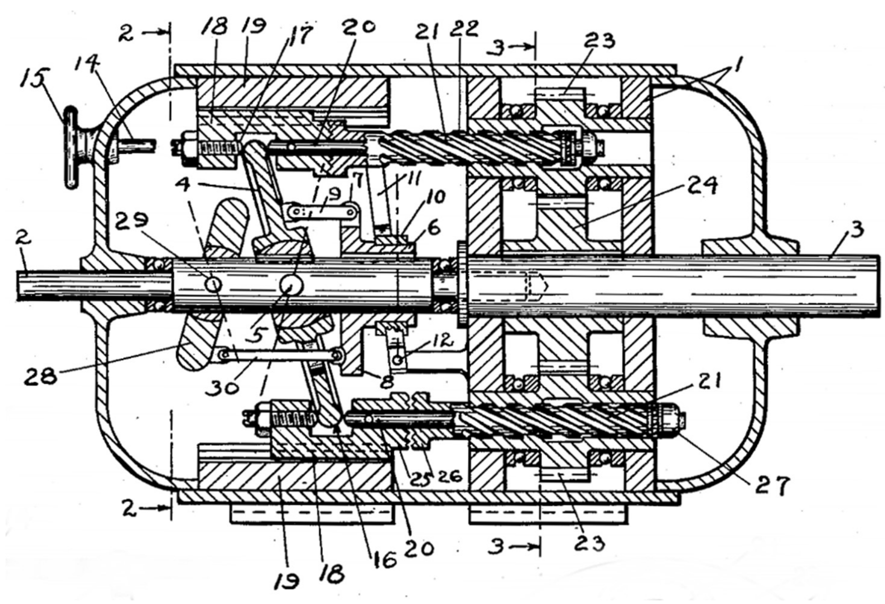

Figure 13. Swashplate pushing helix rods, which drive fixed planet gears

In this concept, helices connect transmission rods to the driven shaft through gearing, converting reciprocating motion of the rods into rotational motion of the driven members. Ratcheting mechanism is also needed for the helix rods, but one is not pictured.

In recent years, Beijing Union University has experimented with an idea rather similar to those of Allen and Eggert. Whereas Allen and Eggert have the linear stroke converted to rotational movement with the help of helical rods and then gears to transmit the power to output shaft, Beijing University's transmission has helical gears that fulfill both tasks (fig. 14).

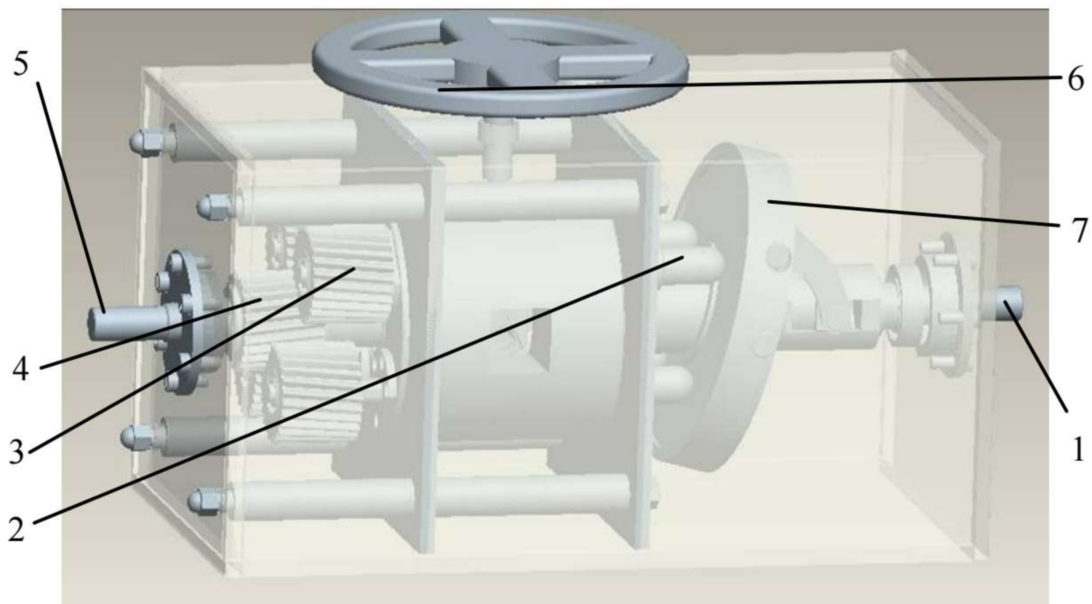

Figure 14. Structure of rotational swashplate pulse CVT (Sun et al., 2012, p. 2)

Input shaft (*1*) can be seen on the right. It drives a swashplate (*7*) which then operates driving helical gears (*3*) through guide rods (*2*). Overrunning clutches are installed between drive gears and guide rods in order to allow drive gears to rotate freely when they are being pulled back by the swashplate. During a push stroke, the clutch engages and the gear will mesh with the output shaft's driven helical gear (*4*). This forces the output shaft to rotate.

A prototype has been manufactured, and its measured efficiency is cited to be around 70%. Due to its uncharacteristic usage of helical gears, lubrication is likely to be a critical factor for this transmission, its efficiency and lifespan.

2.3 Planetary gears

Planetary gears are discussed here because they are a vital arrangement in many infinitely variable transmissions, which are discussed in the next chapter.

Planetary gear is a set of gears where power is transmitted through two or more load paths. This is its distinctive feature compared to simple gear mesh with a single load path. Simplest type of planetary includes a sun gear, planet gears attached to a carrier orbiting the sun, and a ring gear. Figure 15 illustrates such a simple gear train.

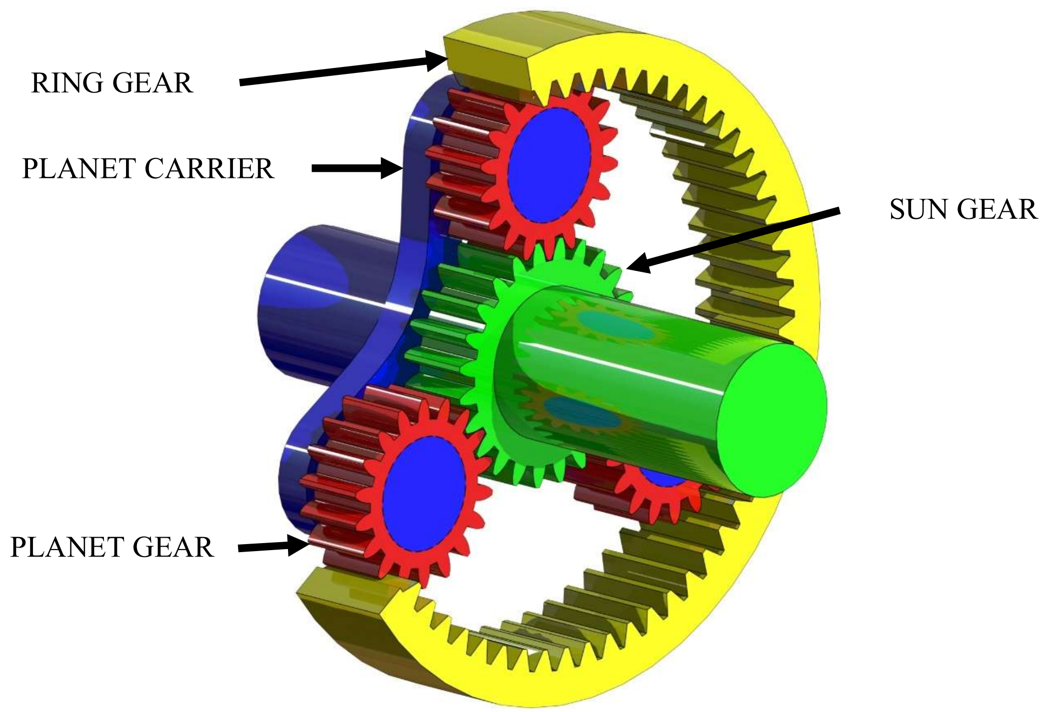

Figure 15. Components of a simple planetary gearset

Planetary gear trains are a subset of epicyclic gearing. Being epicyclic means that a point on a planet gear rotates about its own axis and also about the center of the system, tracing an epicycloid curve. For epicyclic gearing to be planetary, planet gears need to engage both a sun gear and a ring gear.

Lynwander (1983, p.293–295) lists some of the advantages planetary gear trains have over parallel shaft configurations. Because of aforementioned load sharing through several meshes, planetaries are able to transmit more torque that parallel shaft drives requiring similar volume. Relatively smaller and stiffer components contribute to reduced noise and vibration, and thus, increased efficiency. Some applications might benefit from concentric input and output shafts. Finally, this gear arrangement cancels out radial forces, simplifying bearing design as the planetary gear train effectively transmits only torque.

By controlling the rotating components with clutches and brakes, speed changes are achieved. Arranging the gears in different configurations leads to a variety of ratios and power splits. Idealizing the system as friction disks helps deriving the equations describing the mechanics of any simple planetary configuration: a point rotating along the pitch circle of a gear will have the same tangential velocity as another point on another gear in case the two gears are meshing.

Many infinitely variable transmissions rely on epicyclic gear mechanism as means of producing variable output ratios. In this approach, two inputs are needed. Their relative velocities are then adjusted.

Gear assembly with two inputs can be a bit tricky to calculate. The output due to each input can be calculated with the other input held fixed, then summing the results. But before the results can be summed, each input and output combination's velocity reduction ratio needs to be found out.

Simple planetary system with a sun, a carrier and a ring gear can be connected in six different ways, leading to six different reduction ratios. All epicyclic systems have two constraints representing their interactions. First equation shows the relation between sun and planets. (Ferguson, 1983, p. 55–58)

$$N_s\omega_s + N_p\omega_p - (N_s + N_p)\omega_c = 0 \tag{3}$$

Second equation is needed to model the relationship between planet gears and ring gear.

$$N_r\omega_r + N_p\omega_p - (N_r + N_p)\omega_c = 0 \tag{4}$$

Where ω_i are angular velocities for sun, planets, carrier and ring (respectively indexed as *s*, *p* and *r*), and N_i are the number of teeth for each gear in a similar manner. Assuming all gears in the system have the same module, or pitch diameter divided by number of teeth, pitch diameter or radius can be used in the equations instead of number of teeth.

When one element is fixed, its angular velocity is marked equal to zero. Now we can derive reduction ratio for each input-output combination, or $\omega_{input}/\omega_{output}$. These combinations are given in table.

Table 1. Reduction ratios of a simple planetary gearset. In the table, $\alpha = N_r / N_s$.

Input	Fixed	Output	Reduction ratio
Sun	Carrier	Ring	-α
Ring	Carrier	Sun	-1/α
Carrier	Sun	Ring	α /(1+ α)
Ring	Sun	Carrier	(1+ α)/ α
Sun	Ring	Carrier	(1+ α)
Carrier	Ring	Sun	1/(1+ α)

So if the two inputs are sun gear and carrier, the reduction ratios for these combinations separately are $-\alpha$ and $\alpha/(1+\alpha)$. Because reduction ratio is defined as input per output, the output velocity will be the reciprocals of each reduction multiplied by their respective input velocities. Summing these gives:

$$\omega_r = \omega_s \frac{1}{-\alpha} + \omega_c \frac{1+\alpha}{\alpha} = \frac{-\omega_s}{\alpha} + \frac{\omega_c(1+\alpha)}{\alpha} \tag{5}$$

which gives

$$\omega_r = \frac{\omega_c(1+\alpha) - \omega_s}{\alpha} \tag{6}$$

Giving the output angular velocity as a function of two input speeds and number of teeth.

If the mechanism has more than one input and more than one output, some major contributions to systematic methodology of analyzing power flows, torque and kinematics within the system have only been made recently (Esmail & Hassan, 2010). Even with two inputs and just one output, some of the power flow analysis is fairly new. Let us now briefly examine an analysis method presented by Esmail & Hassan (2008, 2009) for two-input epicyclic-type transmission trains.

In a steady state, torques on links about the central axis of the epicyclic gear train have a total sum of zero.

$$T_s + T_c + T_r = 0 \tag{7}$$

Where T_i refers to external torque exerted on given link, be it used as an input or an output. In a similar manner, the sum of power flowing through the system has to be zero because no power is added and losses are neglected. As power is torque multiplied by angular velocity.

$$T_s\omega_s + T_c\omega_c + T_r\omega_r = 0 \tag{8}$$

An approach of fundamental circuits is used in the analysis of two-input epicyclic gear train. A fundamental circuit is defined as a central gear, another gear rotating around its surface, and a link (carrier) to connect them (Freudenstein, 1971, p. 176). When power goes through an epicyclic gear mechanism, at least one fundamental circuit is present. Fundamental circuit is defined active if it takes power from and gives power to environment external to itself. This environment consists of input power, output shaft and other fundamental circuits adjacent to the circuit in question. Figure 16 shows a graph representation of a simple epicyclic gear train.

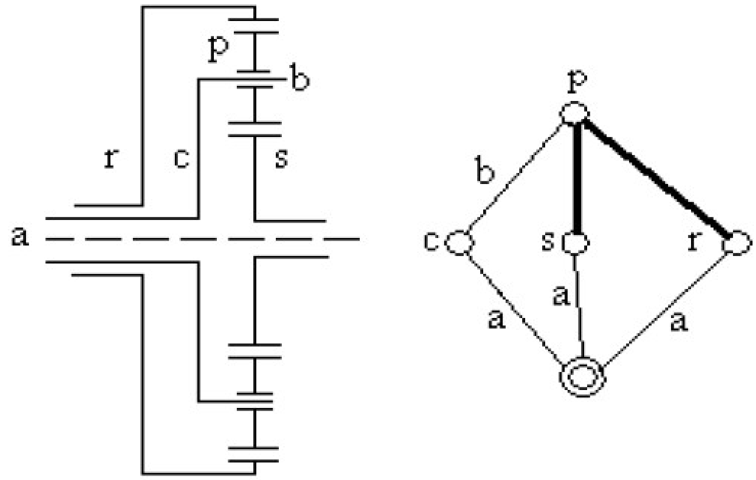

Figure 16. Graph representation of an epicyclic gear train (Esmail & Hassan, 2009, p. 19)

Another concept which needs to be familiarized is one of fundamental gear entity (FGE). It is defined as a mechanism formed by a 2nd level vertex or a heavy-edge chain connected 2nd level vertices together with all their lower vertices connecting them to a root. (Esmail & Hassan, 2009)

Velocity ratio of the epicyclic gear train is the ratio between its input link and output link.

$$N_{p,x} = \pm \frac{Z_p}{Z_x} \tag{9}$$

Where Z_p and Z_x denote the number of teeth on the planet and the sun or the ring gear, respectively, depending on which acts as the output. Positive or negative sign depends on whether output is on the ring or on the sun.

The equation for the fundamental circuit can now be written as:

$$\frac{\omega_x - \omega_c}{\omega_p - \omega_c} = N_{p,x} \tag{10}$$

Solving equations yields:

$$T_c + N_{p,x} \cdot T_s = 0 \tag{11}$$

and

$$T_r + (1 - N_{p,x}) \cdot T_s = 0 \tag{12}$$

These equations are written for each fundamental circuit, which will result in 2(n-2) equations.

Recalling that the sum of torques on any given link is zero, a set of linear equations are obtained for each node. These equations can be written in matrix form:

$$[N]_{AxB}[T] = [T_{in}] \tag{13}$$

Where N_{AxB} is a matrix the elements of which the elements are functions of gear ratios and T denotes the vector of torques. Now the unknown torques can be solved in terms of the input torques T_{in}.

2.4 Infinitely variable transmissions

Strictly speaking, CVT is a transmission that allows a continuous, stepless change of input to output ratio, within a range between finite lower and upper limits. A zero output speed produced from a finite input speed implies an infinite ratio. Thus, an infinitely variable transmission (IVT) with its neutral gear can be seen as an extension to the idea of CVT. (Happian-Smith, 2001, p. 437)

Planetary gears can easily be designed to have such relative sizes between the main components that adjusting the carrier speed in relation to the sun speed, different ratios from negative via neutral to positive ratios can be achieved. Even a variable-diameter pulley variator could be converted into an infinitely variable transmission by having the variator drive a planet carrier whilst the sun is driven directly from the prime mover.

3 DESIGN PROCESS

This chapter describes the phases of a design process. Infinitely Variable Drive project is then introduced and its design phases are shown, reflecting on the general design process.

3.1 Systematic product development methods

Product development is a process of creating a solution for a known need. This can be achieved by either designing a new product or improving a previous design. The engineer's task is to find answers to these problems using natural scientific knowledge and to construct them according to the best contemporary methods available within the given limitations (Pahl, et al., 1990, p. 9).

Although it is identified that generating a concept early on in the process has a significant impact on the efficiency of the design process, initial solutions should be subjected to analysis and evalution, and scrapped if fundamental flaws are found in the conjecture. A new conceptualising cycle is then started. This process can be called heuristic: the designer uses previous experience and prior art, hoping to guide himself towards a viable solution without a guarantee of success. (Cross, 2008, p. 29)

3.2 VDI 2221 design process work flow

VDI 2221 process construes a seven-step work flow structure (fig. 17). Each step has a task and a result that should materialise once the task is completed. (Pahl, et al., 1990, p. 47)

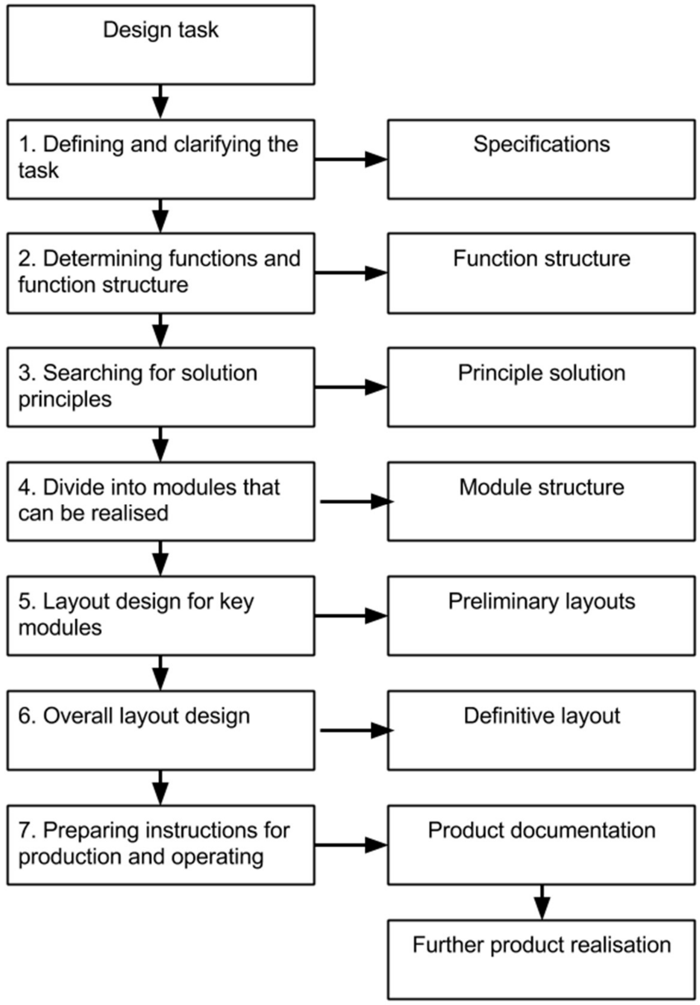

Figure 17. Work flow during design process according to VDI 2221

Precise process descriptions, along with proper documentation, is also vital when introducing new engineers and other workforce to the project.

Exceptionally, the first three steps of VDI 2221 were already taken before the actual project work began. The task was already defined and the desired function was clear: what was yet to do was rest of the concept design for mechanical, infinitely variable transmission. Readily sent patent application laid out the framework around which the rest of the design had to work. Application text describes principle solution, or the third step in VDI 2221 product development process, which is the starting point of the project.

3.3 Infinitely Variable Drive project

Finally, we arrive at the starting point of the actual project itself.

The project was based on a previously created concept of a swashplate-driven infinitely variable differential or IVD. In an effort to limit the design scope, it was decided that further developments of the system should also produce varying ratios by the means of a swashplate mechanism. The structure should also be novel and not infringe intellectual property of already patented ideas.

The basic operating principle (fig. 18) is that tilt angle of a swashplate is changed to generate linear strokes of varying length, which are then fed into planet carrier of a planetary gearset. Sun gear of this planetary gearset is driven by the prime mover, with the output of the planetary on its ring gear.

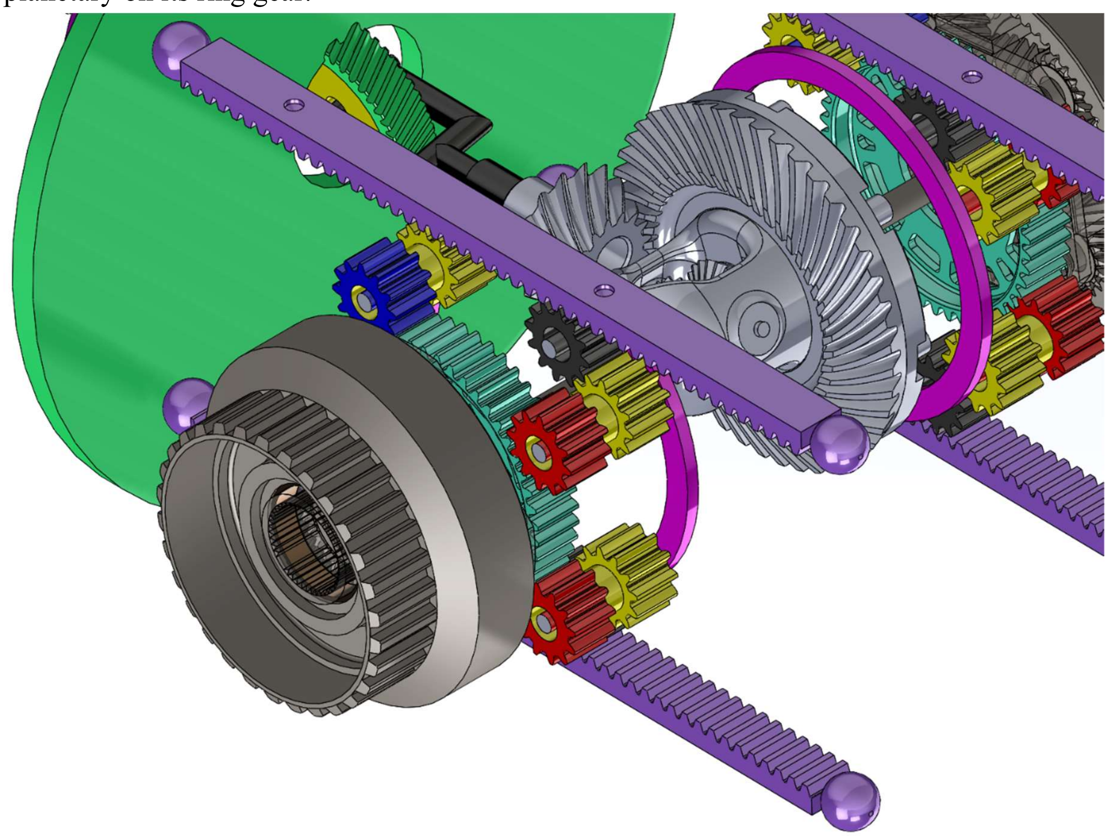

Figure 18. Infinitely variable differential (IVD)

The main problem behind the concept is that the rotational movement of the engine is converted to reciprocating linear movement of the swashplate pins, which again should be converted back to rotational movement in order to drive the planet carrier of the final drive. Such system is easily prone to vibrations and might result in a lower than desired overall efficiency.

3.4 Dividing into modules

Staying on the tracks of VDI 2221, it can be seen that the principle solution is next to be divided into modules.

Module division criteria was the task they perform (fig. 19).

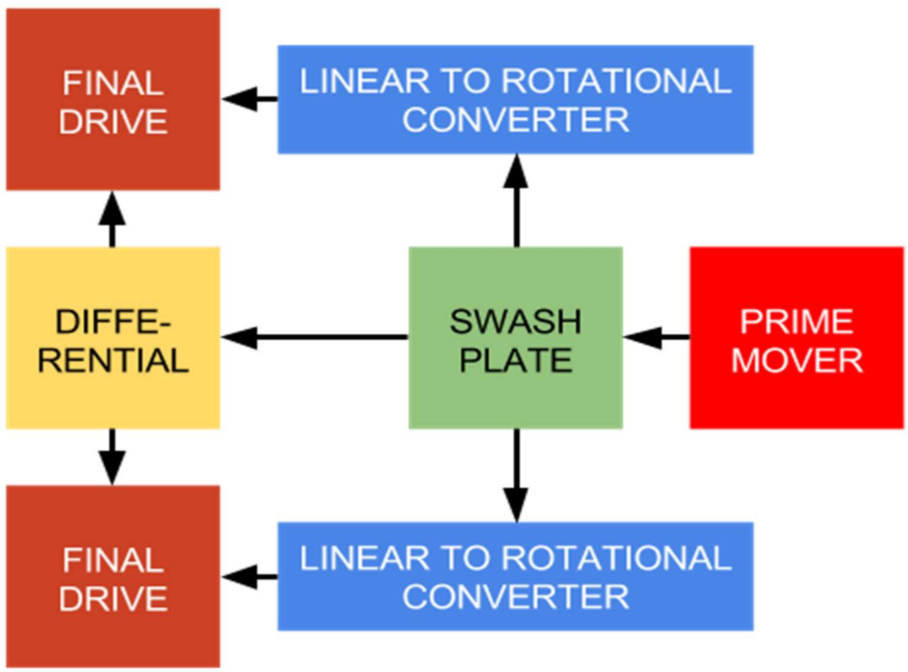

Figure 19. Structural modules of IVD

Swash plate could have also been named “rotational to linear converter” because that is what it essentially does.

After this exercise in sifting, it is easier to see which part of the structure needs more attention. Differential and final drive do not differ greatly from current power transmission systems in design. Swashplate is straightforward. But it can be seen that the module where swashplate’s linear movement is converted back to rotational movement still needs quite a bit work. Initial concept is a loose collective of parts moving in relation to each other.

3.5 Converting linear movement to rotational

It was soon found out that the preliminary layouts, or the next result of VDI 2221 process, would be heavily dictated by the mechanism chosen to convert linear movement to rotational. Supporting structures required by one competing mechanism compared to those needed by another differ greatly.

Some different alternatives to the original ratcheting rack mechanism were studied. One concept mechanism is based on spiral ratchet screwdriver (fig. 20).

Figure 20. Push drill, a mechanism that converts linear motion to rotational (Bradley, 2007)

Spiral ratchet or, colloquially, Yankee screwdrivers have a mechanism that converts linear motion of the handle to rotational motion of the shank. The handle has a pawl built into it which follows a spiral groove. When the handle is pushed along the shank, pawl forces it to rotate in desired direction. Upon lifting the handle, pawl jumps off the groove and the shank does not rotate.

Due to limited space in longitudinal direction, it is not likely sensible to use the screwdriver mechanism as such. In proposed model (fig. 21), the shank reciprocates and does not revolve, but it still carries the spiral groove that pushes against a pin in the bevel pinion.

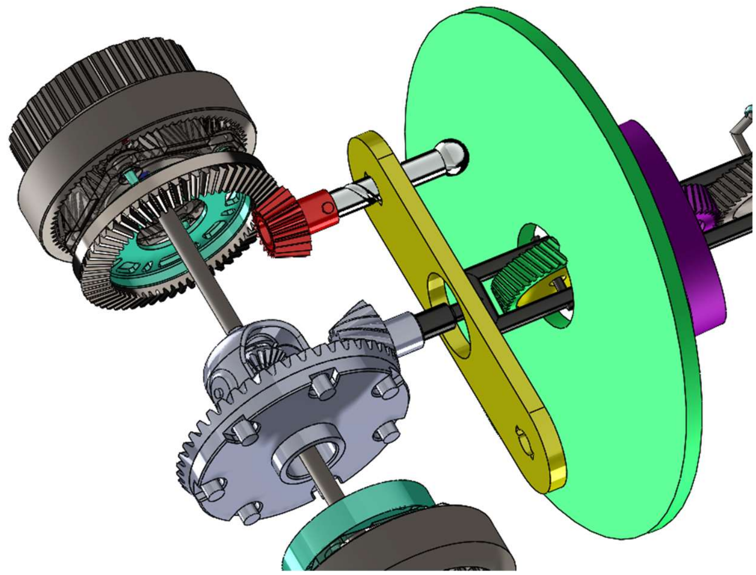

Figure 21. Pump action motion converter

A one way bearing between the shaft and the gear forces the pinion to rotate when the swashplate pushes the pin, and lets it slide freely when the pin returns.

Another motion converter idea was lifted directly from Constantinesco's mechanical torque converter. It consists of two ratchet wheels, sprag clutches or other one-way clutches mounted on the same shaft, side by side, and operated with push and pull levers on opposite sides of the shaft (fig. 22).

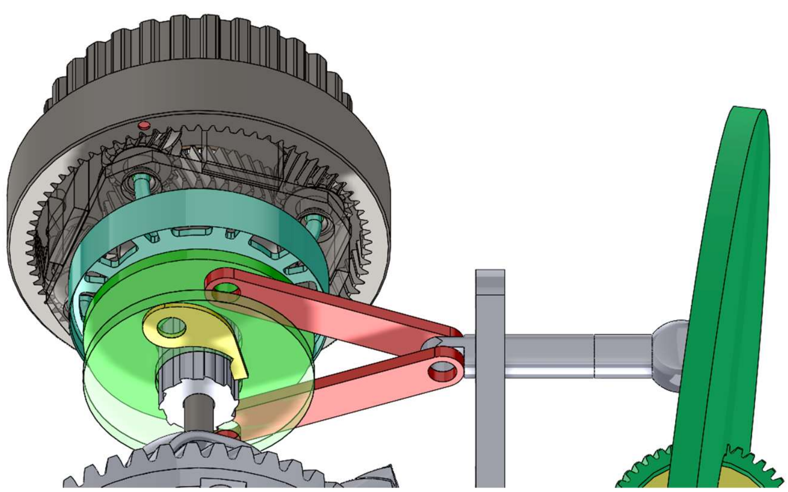

Figure 22. Double ratchet mechanism

Its advantages compared to screwdriver mechanism include reduced complexity and work performed on both push and pull phases of the swashplate movement.

3.6 First prototype

It was decided that a prototype is needed in order to find the key issues with the mechanism. Due to budget and schedule limitations, some restraints had to be set on the prototyping process itself. As many components as possible should be recycled from other machines, as few as possible completely self-designed and manufactured.

Electric motor was chosen as power input for the system. When a 500 W BLDC motor with battery and controller included was found with a reasonable price, it was decided that the rest the setup should be designed around this motor and the torque and power it puts out.
In small scale prototyping and functionality of the mechanism as a whole being the main concern, constructing a complete swashplate mechanism with angle adjusting was ruled out. Thus, instead of building any of the three motion converters presented above, a chain ratchet mechanism was designed (fig. 23).

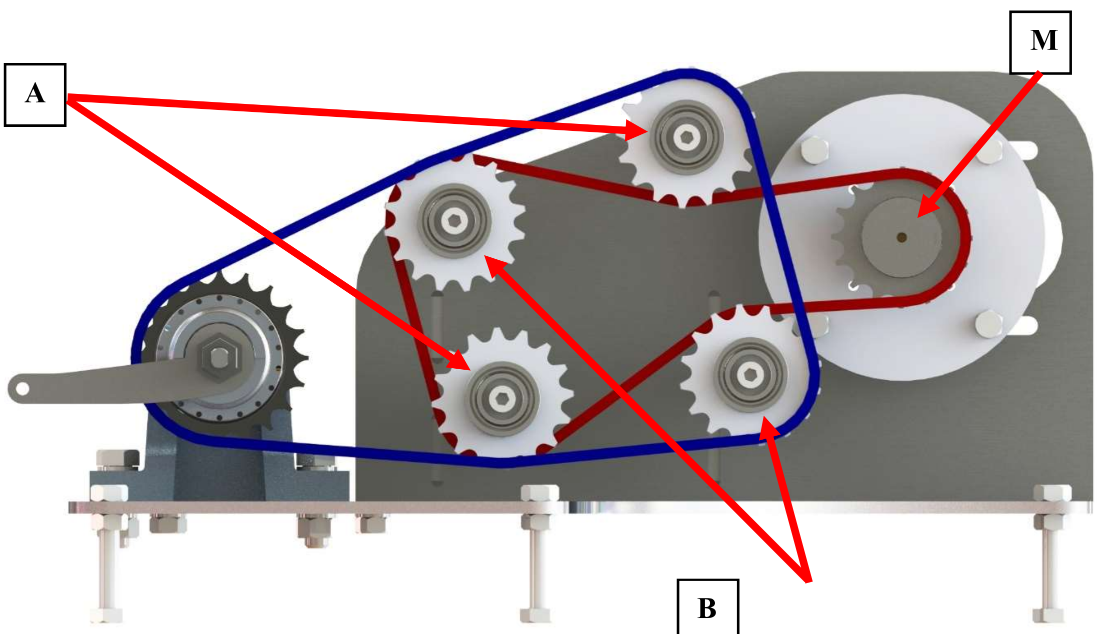

Figure 23. Operating principle of chain ratchet

Considering the VDI 2221 process, this construction could now be seen as one half of the final layout.

Movement created by swinging ends of half a swashplate was simulated with a stepper motor *M* programmed to rotate a given number of degrees back and forth. Stepper motor sits on the right-hand end of the red chain. In some other configuration the chain would be cut at stepper motor's location and its ends attached to tilt plate. Red chain is in contact with four freewheel sprockets. All four engage their shafts when rotated counter-clockwise (CCW); when rotated clockwise (CW), they rotate freely on their shafts. Now when the stepper motor turns CW, sprockets *A* get rotated CCW and sprockets *B* freewheel. When the stepper motor switches direction and rotates CCW, sprockets *B* engage and *A* freewheel. On the freewheel sprockets' shafts, there is another set of sprockets fixed on the shaft. So now when the stepper motor rotates back and forth, the second set of sprockets drive a secondary blue chain constantly in the CCW direction, ideally creating a smooth output on hub on the left-hand corner of figure.

It is known that this arrangement does not replicate swashplate's movement perfectly, but it was also thought to demonstrate the major issues such a design would have before spending resources on swashplate design. Stepper motor was programmed to rotate imitating sinusoidal output of a rod attached to swashplate.

Because the concept as a whole still needed to be questioned, parts of it were first to be tested separately. Main concern among designers was smoothness of power delivery. It was decided that this could be tested by building a partial model of the system (fig. 24) and only include mechanism between one wheel and one side of the swashplate.

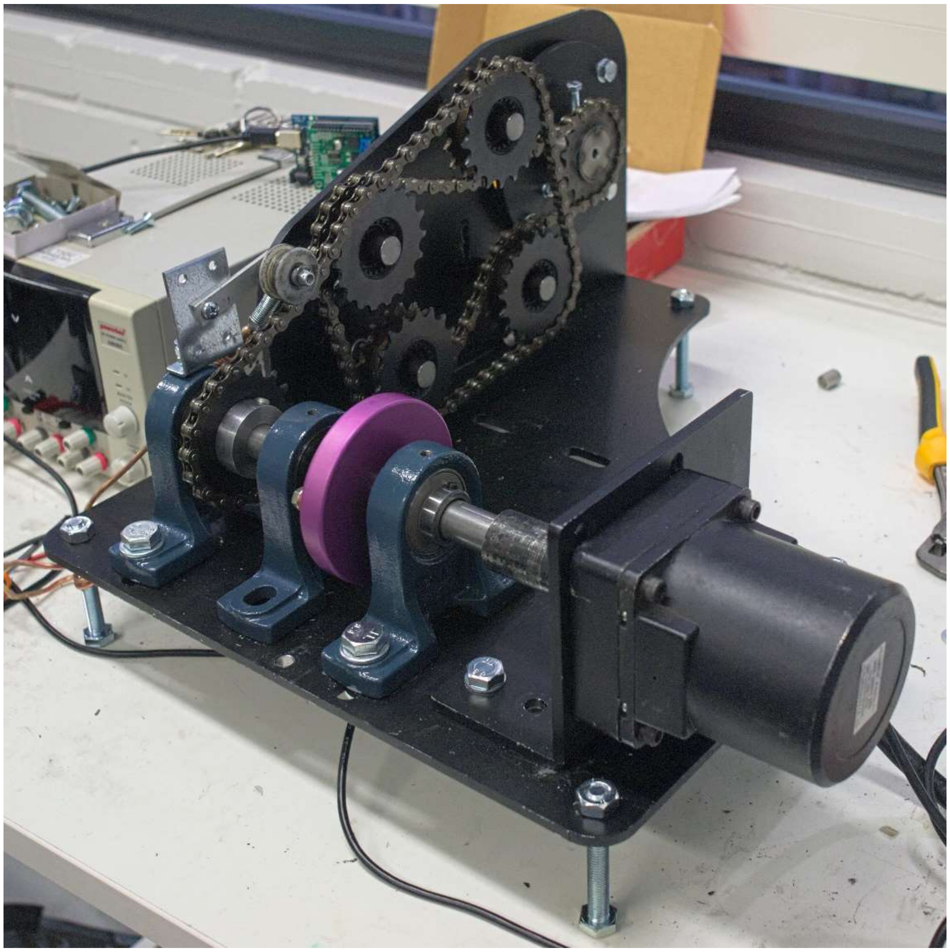
Figure 24. Ratcheting CVT test bench.

Acquiring a differential was also avoided.

Three-speed Shimano hub was partially disassembled and its brake shoes removed. Because the normally stationary sun gear (fixed on the hub's main shaft) is rotated, the jerking motion of the planetary could engage the brakes had the shoes not been removed. For reasons which were never thoroughly investigated, using bicycle hub yielded unpredictable results. Thus, it was replaced with a gearset from an educational scale model (fig. 25).

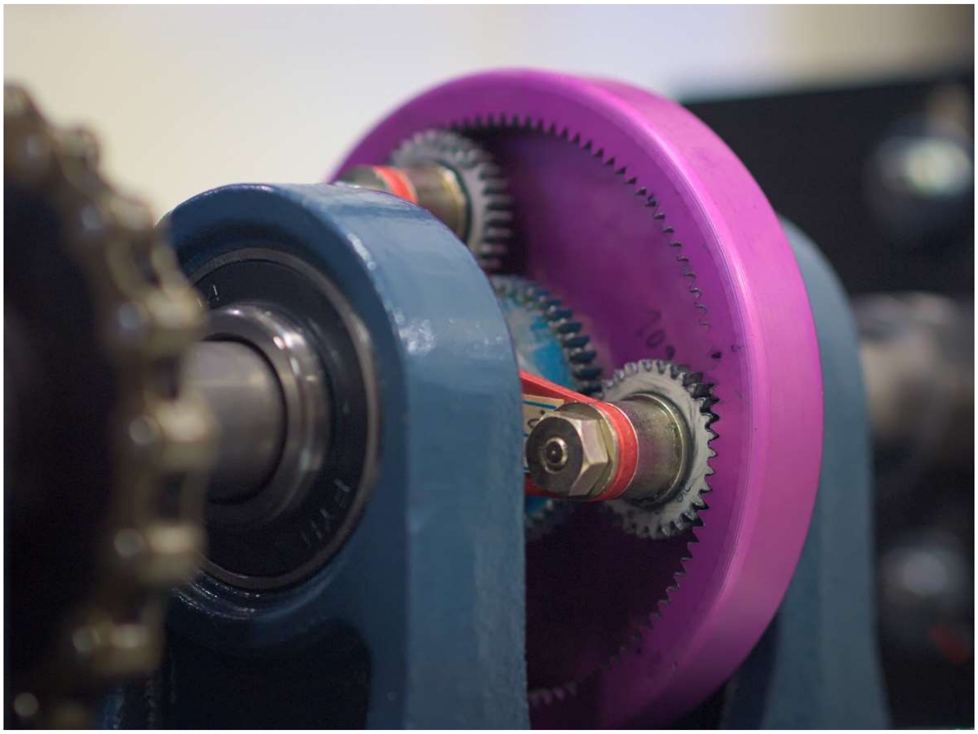

Figure 25. Educational planetary gearset.

In a complete system, swashplate and planetary sun would be rotated by the same motor. Now that the swashplate is not used, the sun is attached directly to an electric motor. This saves the effort of designing and building bevel gear and connection from it to the sun.

The test setup was also equipped with a simple brake mechanism: two plastic blocks tightened with wing nuts. Power output of the system is on the outer edge of the hub which does not carry a lot of inertia. Brake is supposed to give the transmission some kind of a load to work against in absence of rotating masses and vehicle mass.

3.6.1 Testing results

The test setup used fairly standard single-pawl bicycle freewheels. Ratcheting freewheels always have a certain amount of backlash depending on pawl's position relative to corresponding tooth and number of teeth in the gear itself. Turning these affordable Chinese freewheels used in the prototype one can hear 20 clicks for each revolution of the sprocket, which means the worst-case scenario for backlash is 18 degrees before the output is engaged. The engagement delay comes into the picture when direction of rotation in the primary chain changes, contributing to torque ripple. Backlash is constant, so it is most significant in proportion when stroke is short. Freewheel sprocket has a rolling diameter of 62 mm, so with the pawl in a most unfortunate position, the linear stroke has to travel almost 10 mm before secondary chain moves.

Some improvement could have been achieved by using freewheels with multiple pawls. However, often freewheels with four pawls have two sets of two pawls half a space away from being opposite the other, meaning two pawls (not one) carry the torque through the mechanism. Despite potential, backlash is not a quarter of that of single-pawl freewheel's, but just half of it.

3.6.2 Structural issues

Backlash issue can be partially solved by changing ratcheting freewheels to a stepless type of overrunning clutch. Sprag clutches, for example, are used in conventional automatic transmissions as a method of allowing the gearbox to smoothly change gears under load.
Some concern was voiced regarding sprag clutches' ability to transmit torque. Properties of It can be concluded that when selecting a sprag type clutch for a ratcheting transmission application, the torque transmission properties of the clutch need to be carefully examined and its loads calculated. One-way clutches have a crucial role minimizing the torque ripple.

Some of the torque ripple can be defined structural. Zero-Max ratcheting CVT has four links with a 90-degree phase difference to each other. Project CVT also has four, but they have been divided between two outputs. Because only a push forward is a working stroke, dividing the swashplate outputs to left and right side will result in significant torque ripple.

Pushrods on either side are 90 degrees apart (fig. 26).

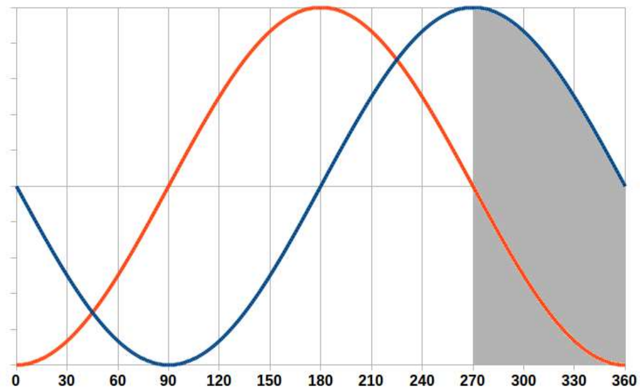

Figure 26. Relative pushrod positions on one side.

In the figure, X-axis is degrees of swashplate rotation and Y-axis marks relative displacement from full swing backwards to as forward as the pushrod can reach.

Let us consider a situation where the first pushrod (red line) begins its work stroke, and mark this point as zero degrees of swashplate rotation. When the swashplate has rotated 90 degrees further, the second pushrod catches on and starts its work stroke. After three fourths of a full revolution, both pushrods are retreating. The area marked grey in the graph shows where neither pushrod is performing work.

Finally, the design does not have a truly neutral gear, but a certain swashplate angle at which the output is averagely neutral. Due to pulsating power input from swashplate, the planetary gearset component it is driving is expected to freewheel. There will still be some kinetic energy in the component between one stroke ending and another about to begin, so it will keep rotating. Isolating this driven component from the rest of the gear train, it will have rotational energy E_r.

$$E_r = \frac{1}{2} I \omega^2 \tag{14}$$

Where is I is the moment of inertia around the axis of rotation and is ω angular velocity. Kinetic energy and rotation speed do not have a linear relationship. This leads to a situation where increasing the input speed will push the swashplate-driven gearset component a longer freewheeling travel. Most importantly, the travel between power strokes is longer in relation to the gear train component directly driven from the engine. In conclusion, changing the input speed will change the ratio slightly, and thus the swashplate angle with which the neutral speed is achieved changes with input speed.

3.7 Second prototype

The concept was then refined and simplified before building another prototype.

3.7.1 Proposed improvements

Examining some key features of first prototype's structure, it can be seen that the differential dividing the drive in two does not per se affect the final transmission ratio.

Pushrods attached to the swashplate only perform work in one direction. With two one-way clutches, it is simple to construct a mechanism that converts reciprocating linear motion to rotational motion in both push and pull phases (fig. 27)

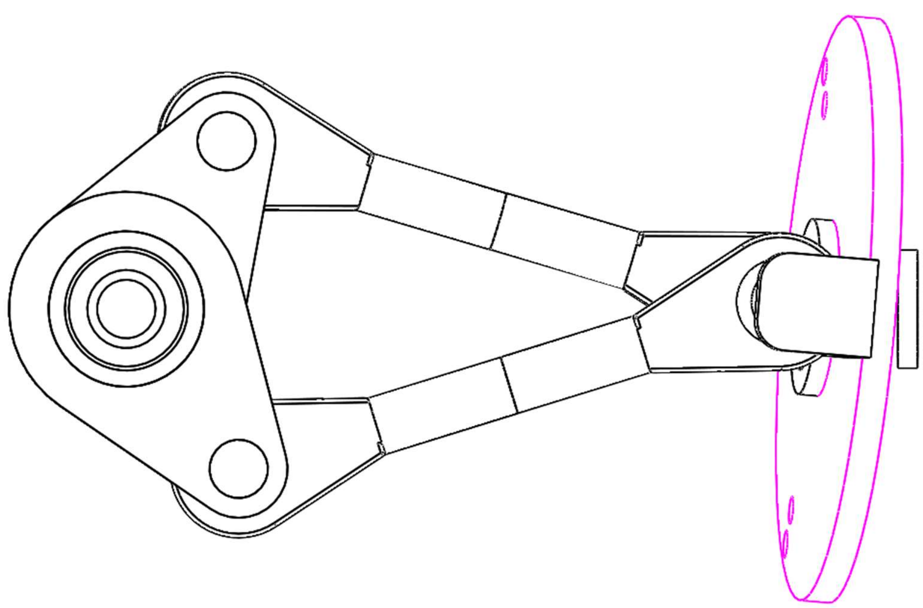

Figure 27. Reciprocating to rotational motion

With the differential between the left and right sides of the swashplate, there is only so much one side of the plate can do. One way to simplify the construction would be to only drive one output with the swashplate.

Depending on where the design emphasis is, it could be that smoother output is more desirable than dividing the load between two gears. Any odd number of outputs will produce a situation where two gears will never be working simultaneously. For example, let us consider a system with three pushrods, 120 degrees apart from each other (fig. 28).

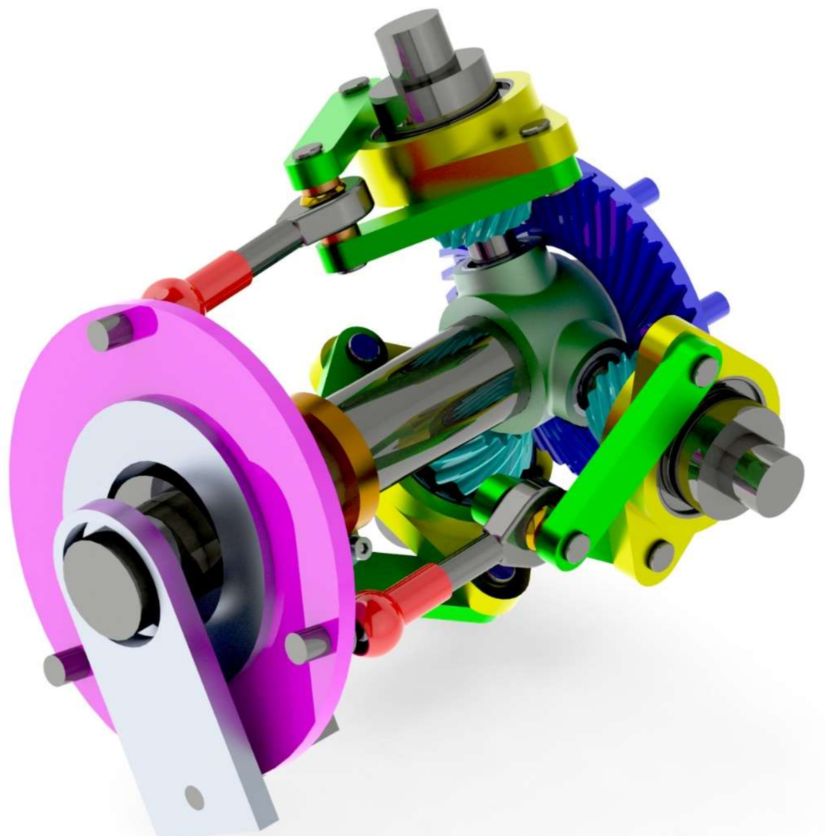

Figure 28. Three-pushrod swashplate CVT

Idealized pushrod locations would now look like this (fig. 29).

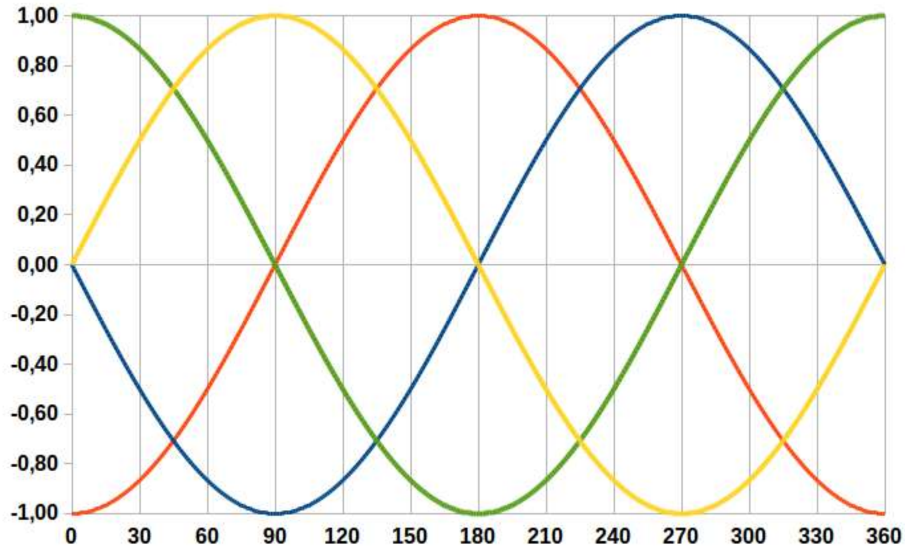

Figure 29. X-axis displacements of pushrods

Derivative of position is velocity and derivative of sine is cosine, so it can be stated that the velocity graph of the pushrods has a similar shape with position graph, following it with a 90-degree phase difference. Because the fastest-moving pinion is the one dictating the pace of the output bevel, the speed curve follows the highest point on the graph.

Opposite pushrods (phase difference 180 degrees) work in parallel, one pushing and the other pulling at the same speed. Because all pushrod-driven pinions rotate against a shared output gear, they rotate at the same speed. Because of one-way clutches, the two gears having the highest velocity perform the work and two others freewheel.

Swashplate mechanism has a simple tilt angle adjusting mechanism (fig. 30)

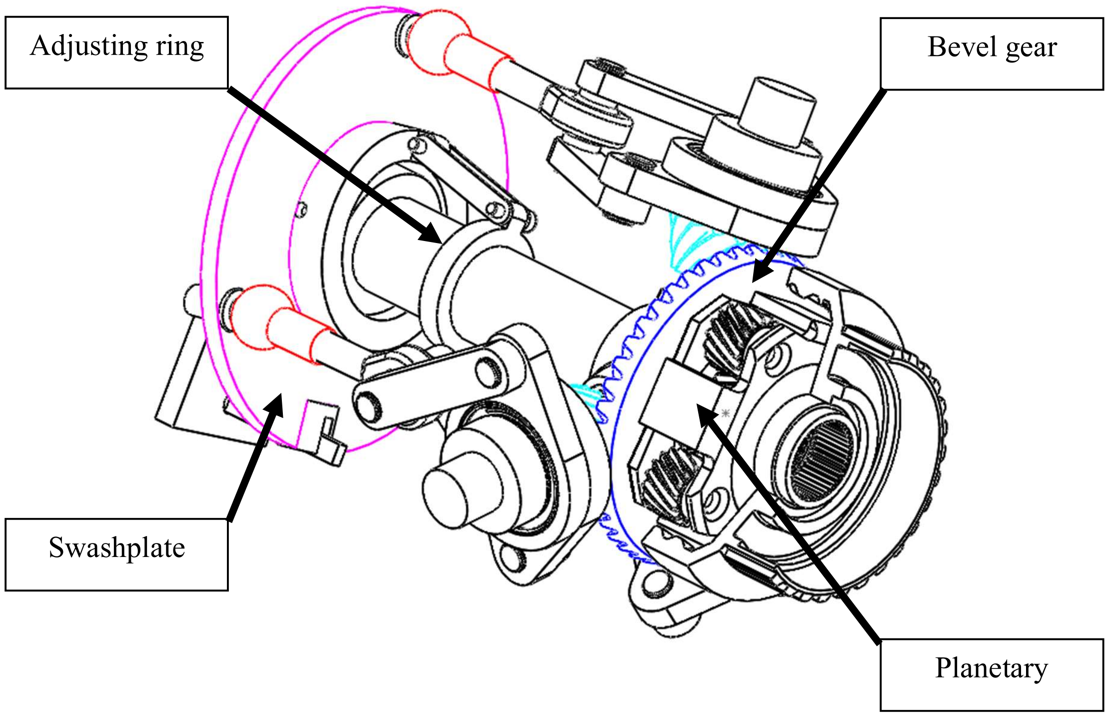

Figure 30. More detailed view of three-pushrod swashplate arrangement

Adjusting ring rotates with the swashplate mechanism and the driving shaft. Swashplate is separated with a thrust bearing and does not rotate. Along the same lines, the adjusting ring would have a bearing between itself and whatever mechanism is used to move it around. This could be done with a lever fork, lead screw or something comparable.

A planetary gearset can be placed right after the swashplate and linear to rotational converter. Arranging the pairing of these two so that the main shaft drives the sun gear and bevel gear drives the planet carrier makes the transmission an infinitely variable one. In the example structure, the maximum angle of rotation is 28,7 degrees between the two extremities of the swashplate. Because the pushrod operates the gear when moving in and when backing out, the final rotation angle of the gear during one revolution of the swashplate is twice as much. Because of the freewheeling and phase difference, the gear actually performing the work changes over the duration of the revolution, but nevertheless all three will still end up rotating the same number of degrees.

Three-pushrod design does minimize the number of pushrods, but it has certain problems as well. Because only one of the two sprag clutches has any meaningful resistance at any given part of the pushrod's cycle, the setup will try to swing in the freewheeling direction. This one unnecessary degree of freedom could be eliminated by using two pushrods attached to the same point on the swashplate perimeter.

Instead of trying to burden the same rod in both push and pull phases, having six separate pushrods attached to three points on the perimeter as pairs would create a simpler structure.

Bevel gear has a ratio of 1:3. Consider that each swashplate revolution is converted into two working strokes. Let us convert these strokes into revolutions. Remembering that in this particular structure, the maximum angle was 28,7 degrees

$$\frac{28{,}7° \cdot 2}{360°} = 0{,}1594 \tag{15}$$

Thus, the final ratio is derived.

$$0{,}1594 \cdot \frac{1}{3} = 0{,}053 = \frac{5{,}3}{100} \tag{16}$$

Output of the bevel gear rotates 5,3 rounds for every 100 revolutions of the swashplate's main shaft.

In order to achieve infinite variation or reverse, neutral and forward with this particular planetary gearset, the maximum speed forward needs to be somewhere near the main shaft's speed. One way to achieve this would be to mount an overdriving auxiliary shaft between the output bevel and the planetary gearset. Another conclusion could be that planetary gearset and bevel gears should be dimensioned otherwise.

3.7.2 Construction and preliminary results

It was decided that another prototype should be constructed to examine the improved concept.

Saimaa University of Applied Sciences had recently acquired a new 3D printer, so the concept was structured to better suit rapid prototyping methods and employ standard components where possible. A simple mechanism was designed for adjusting swashplate angle by hand during operation. Finally, the printed parts and store-foraged components were assembled and paired with the 12-volt electric motor previously used with the first prototype (fig. 31).

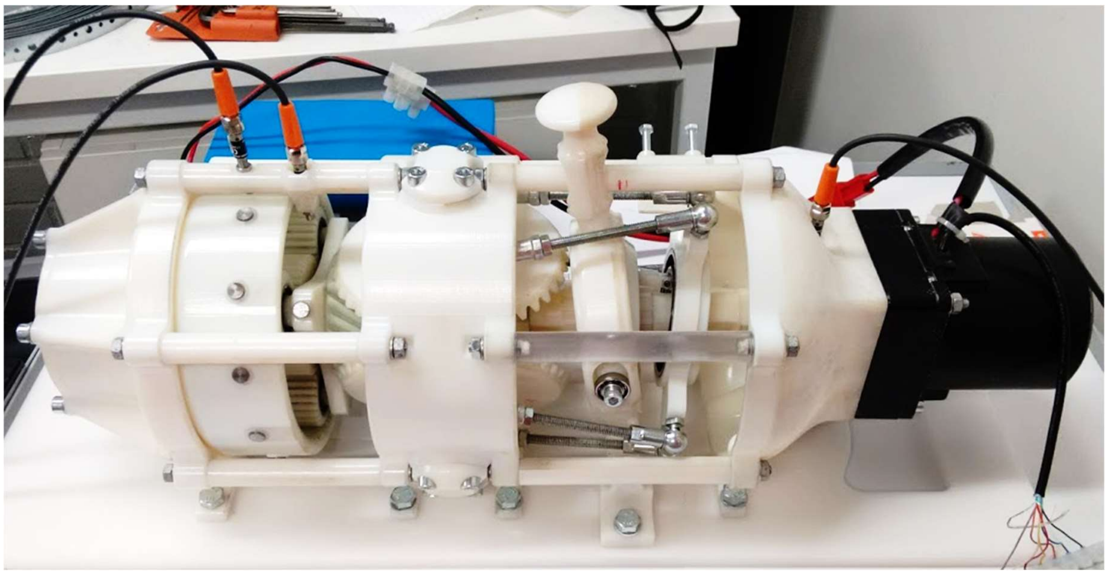

Figure 31. Second swashplate infinitely variable drive prototype

Some sensors were put in place to measure input speed, output speed on bevel gear, and ring gear speed. The arrangement is simple with heads of steel screws passing inductive sensors at a close distance, creating pulses which can be counted and then converted to frequency.

Angle adjusting mechanism is fully manual. Figure 32 offers a view into it.

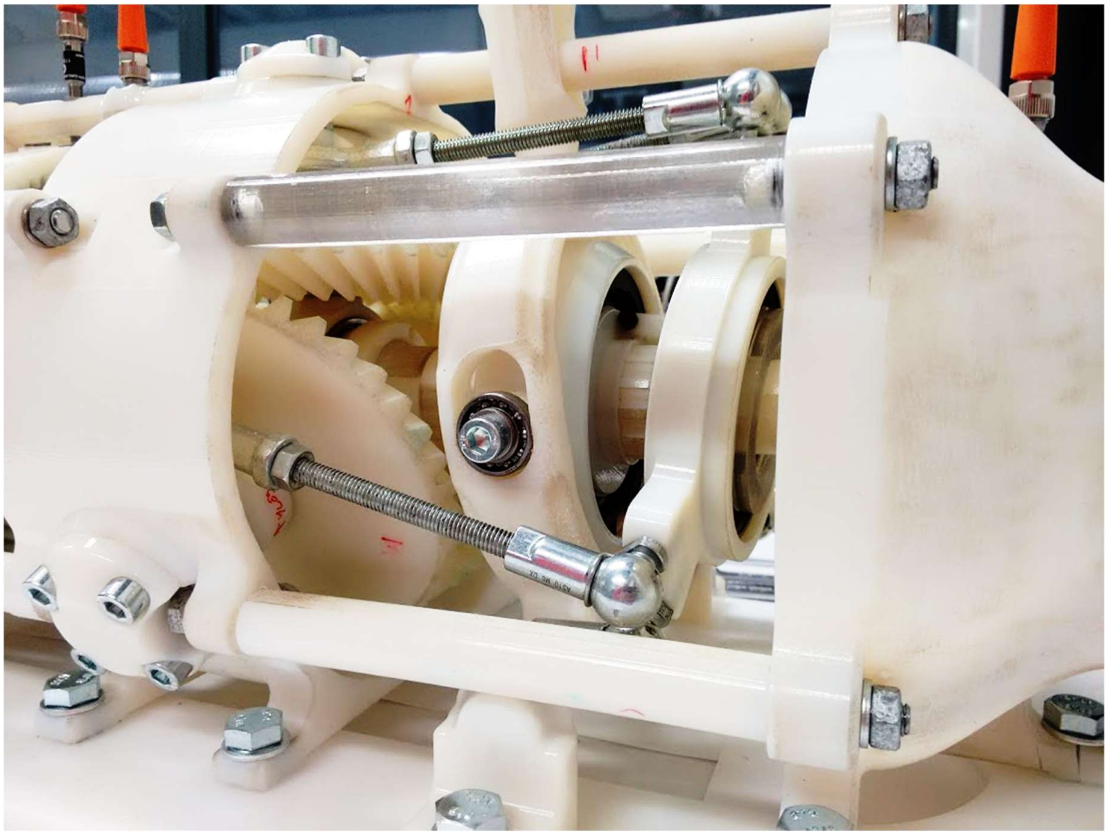
Figure 32. Closer look at the speed-varying mechanism

Pushing the adjusting knob towards the bevel gears increases swashplate angle. The knob is tightened to a guide shaft with a screw.

After assembly, sensors were hooked up into a datalogger and some initial tests were performed. First, the swashplate angle was set to zero, which produces no movement on the pushrods, leading to a stationary planet carrier (fig. 33)

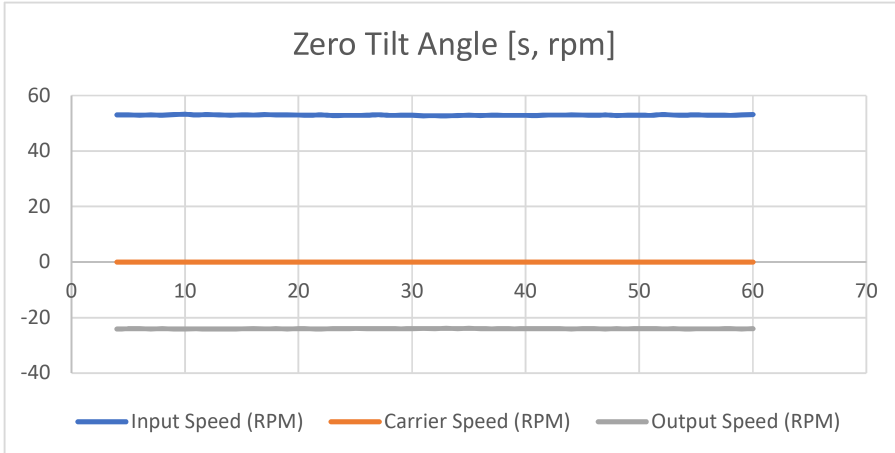

Figure 33. Swashplate tilt angle set to zero

As can be expected, when the planet carrier is stationary, the gearbox produced a steady negative ratio of -0,453. First, the test drive was performed without any load, then with some brakes on, both giving the same results.

The intention was that at zero tilt angle the gearbox produces its largest negative ratio, and that by increasing the tilt angle this negative ratio will diminish to zero before the output is eventually positive. The next task was to find out whether the neutral gear can be found (fig. 34)

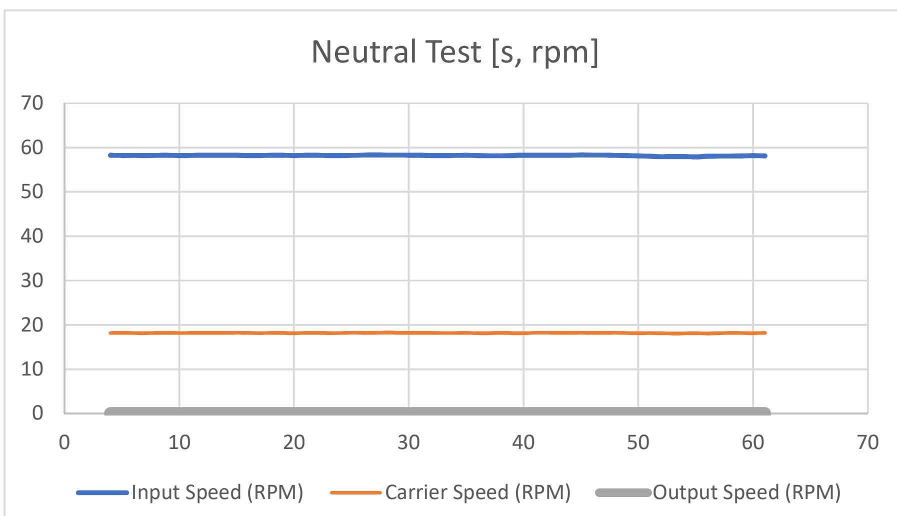

Figure 34. Testing neutral ratio

It was found that the neutral ratio can be achieved, and that the system behavior is relatively smooth.

Next, the swashplate angle was increased in order to achieve a positive forward ratio. Results are presented in figure 35.

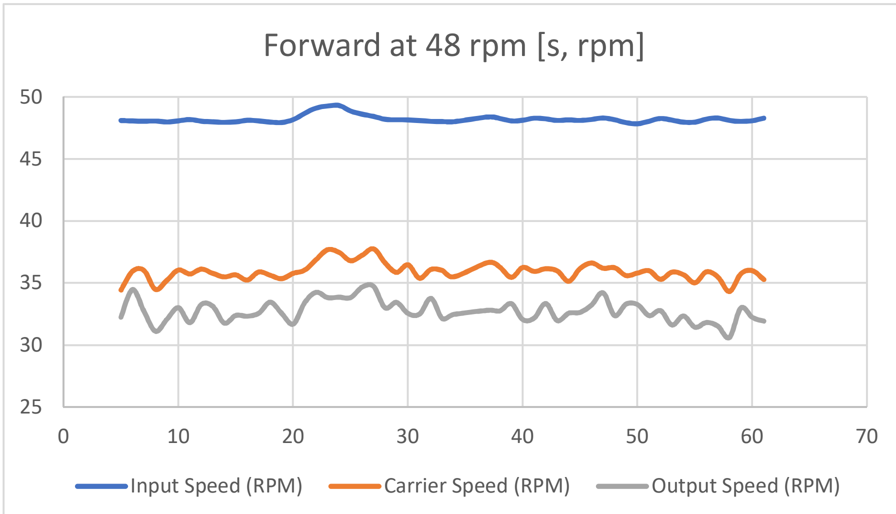

Figure 35. Forward speed test with load

This test was performed with a load braking down the gearbox output. Average transmission ratio of 0,679 was achieved. As the graph shows, when the tilt angle is increased, the carrier's rotation becomes a little twitchy.

Using the same swashplate angle but dropping off the load on the output shaft yields completely different results (fig. 36)

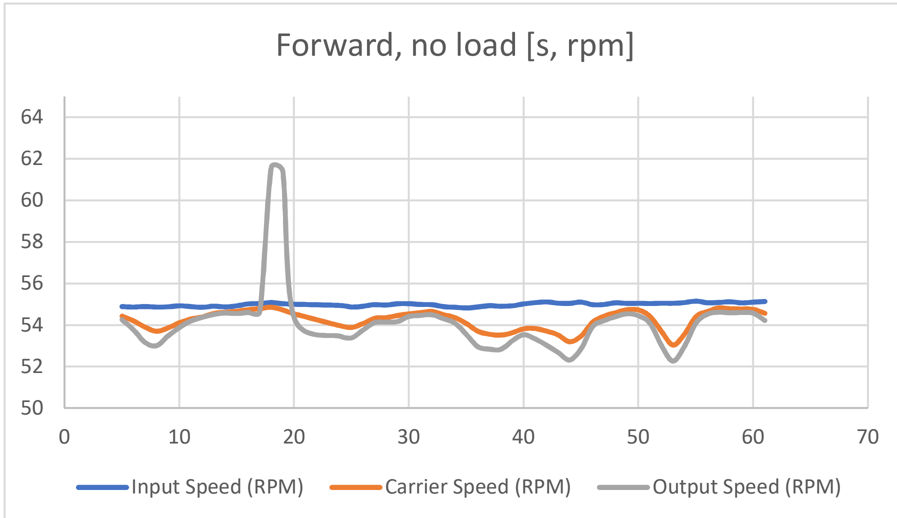

Figure 36. Forward speed test without load

When there is no force present to brake down the carrier, sprag clutches start to overrun and ratio becomes only a little short of 1:1. Because of this same overrunning issue, swashplate angles between zero and neutral drive did not yield consistent results.

4 DISCUSSION

Testing the proof of concept lead to some initial findings.

Main issue with the arrangement was with the sprag clutches. When planet carrier is rotating slower than the sun gear, sprag clutches start to overrun, thus keeping carrier speed equal to sun speed. To rephrase this situation: carrier is not being ran by swashplate mechanism, but the sun gear. A brake should be applied to planet carrier in order to prevent it from running with sun gear, but this is hardly economical. However, with no brakes or other forces applied, all parts are driven by input speed, or at the speed of the sun gear. Freewheeling clutches are going to freewheel, when their output rotates faster than their input; an unladen planet carrier tends to follow the speed of the sun gear, because there is nothing to slow it down and keep it from doing so.

Because of this overrunning issue, reverse speed could not be produced with this arrangement. With the chosen gear set, negative final ratio would have required planet carrier to rotate slower than the sun gear. However, completely stopping the planet carrier yields reversing mode, but this being only a single ratio and not a flexible variable band of ratios from reverse through neutral to drive mode, more developing is needed.

Overrunning issues can partially be overcome by redesigning the planetary gear so that the carrier speed is higher than the sun gear speed. Simulating and calculating this arrangement shows that overdrive and direct drive could be achieved, but neutral and reverse would still need carrier to be slower in relation to sun.

Some minor inconveniences were also listed. Because the space was limited and the design needed to be as compact as possible, rocker arms could not be positioned ideally. In further iterations, their joint locations should be as close as possible to even 120º distribution around the perimeter of the swashplate. That is, if there are three bevel pinions driving an output. Positioning two joints over each other would ensure smooth power delivery, but studying other solutions can be considered if they are mechanically less complicated.

Swashplate also acts as a rotating unbalance. Eliminating sources of vibration is obviously a top priority in powertrain design, but as the stroke needs to be variable length and thus the swashplate angle changes, counterweights are not sufficient to cancel out vibration. If mass was added to the plate to balance the rotating couple and static unbalance at one particular angle, it would move mass center further away from the tilt axis of the plate and thereby increase static unbalance at other angles. If a larger scale prototype is designed, some space should be reserved for a balancing mechanism. Smith's expired patent shows one convenient method (fig. 37).

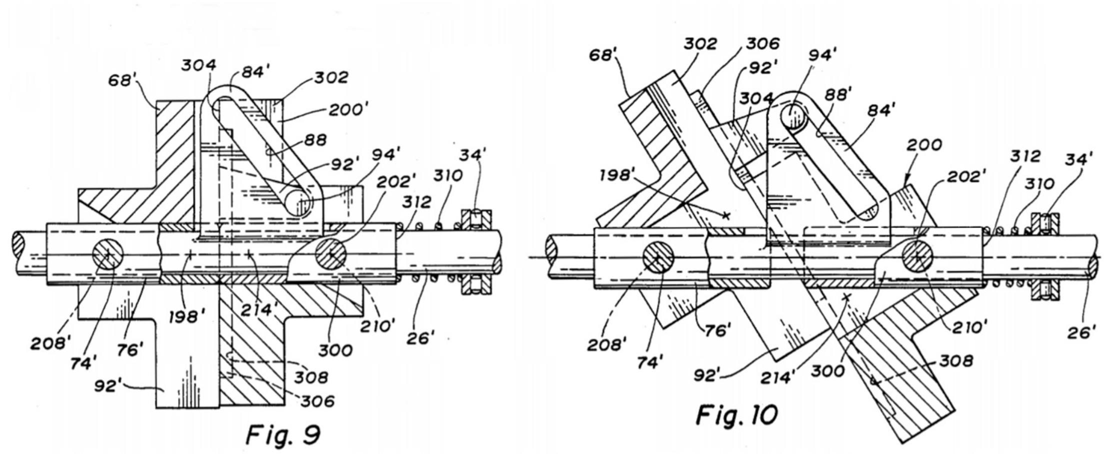

Figure 37. Balancing swashplate with an adjacent disk (US 4836090 A, 1989, p. 5)

This swashplate structure has a balance disk that tilts conjointly with primary disk to maintain static balance throughout the plate's range of motion. This arrangement is beneficial especially when swashplate tilt axis cannot be placed in the middle of the plate.

Angle adjusting mechanism is not as precise as it could be. Its most notable shortcoming is its lack of measuring device: apart from permanent marker lines on white plastic, there is not really a way to interpret swashplate position. Linear displacement sensor could track linear movement an axis. Because their signal is linear enough for most needs, relatively simple on-board electronics could convert measured linear displacement to tilt angle. For the adjusting part, leadscrew rotated by a stepper motor would be the first option to explore.

4.1 Challenges with dimensioning

In both concepts, there is a need to extract linear movement from the perimeter of the swashplate. Due to needed degrees of freedom, using standard axial ball joints appears attractive. But if they need to carry pulling forces, it might not be sensible. Let us examine the structure of a standard ball joint, presented in figure 38.

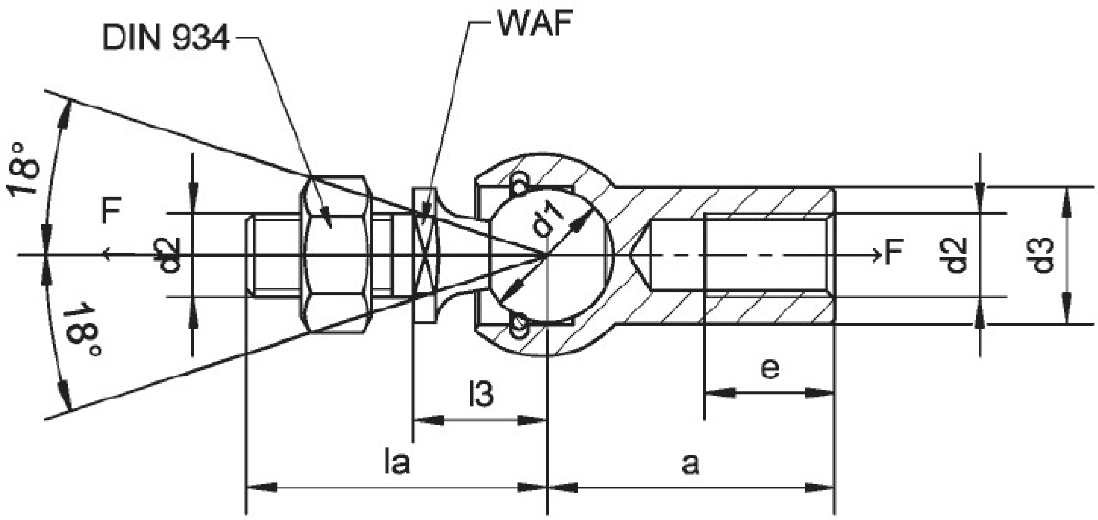

Figure 38. Structure of an axial ball joint (DIN 71802)

The figure presents an axial joint similar to a DIN 71802 angle joint. Respective sphere shaft sleeve is described in better detail in DIN 71805. As the figure shows, there is only a circlip keeping the assembly together. The circlip, semi-flexible in its nature, retreats to its groove when the joint is being pulled. On one hand, this makes it easier to disassemble the joint. On the other, it is also why the axial ball joint can hardly tolerate any pulling forces relative to the pushing pressure they can withstand. For example, a size 19 ball joint (ball diameter d_1 = 19 mm) has a minimum extraction force of 100 newtons according to DIN 71805.

Angular ball joints could be used instead, but then the design would have to deal with the eventual bending moment resulting from the distance between the line of action of the pushrod and the mounting point of the joint. Another option could be to use a universal joint, but as they are typically intended to be used as rotating shaft couplings, their DIN 808 standard does not describe any type of axial fastening. This is also why the standard does not cover permitted axial loads, but just the allowed torques.

Using a spherical bearing, as in a rod end, could be more feasible. Most manufacturers offer static load ratings for their standard product range. Dynamic loads produce greater stress on the member than static loads, which is why a load coefficient needs to be used to accommodate for the alternation. Chiavette (2017) uses formula for permissible dynamic load.

$$F_r\ amm. = C_o \cdot K_f \tag{17}$$

In equation, F_r *amm.* is maximum permissible load on the rod end, C_o is permissible radial static load on the rod end, and K_f is load coefficient. Applied nominal radial force should be less than permissible, naturally. Constant load constitutes as $K_f = 1$, simple dynamic load (constantly positive or negative) is 0,5 and alternating dynamic load is 0,25. Even when accounting for alternating dynamic load of swashplate, a mere M5 size rod end has permissible dynamic load of 150 N, or 50% more compared to size 19 ball joint extraction force.

5 SUMMARY AND CONCLUSIONS

IVD was originally created with vehicle powertrains in mind, but trying to extract high power density from it appear to need unfavorable amounts of mechanical complexity. Simplifying the construction to a CVT could help it find markets somewhere else. Applications currently using Zero-Max unidirectional adjustable speed drives could be a good starting point for market research efforts. There are probably industrial applications where only a constant speed input is available but a variable speed output is needed. Whether the differences between swashplate drive and Zero-Max work in swashplate's favor is a research question best left for marketing people.

Using the swashplate variator to build an infinitely variable transmission might prove to be too ambitious and, due to its mechanical complexity, not economically feasible. If the planetary gear train is completely left out and reverse gear is pursued with some alternative arrangement instead, the swashplate mechanism alone could be used as a continuously variable transmission. After all, it would produce relatively smooth power transmission with little torque ripple, and its neutral ratio would be easy to find at swashplate's zero angle. Neutral gear position would not falter by varying input speed, either. Furthermore, should vehicular applications remain the main target of the next round of iterations, it could be studied whether continuous variation is necessary when the vehicle is in reverse. At least some vehicles could do with a fixed reverse gear ratio, which could be easier to achieve than a full range of infinite variation.

If only neutral and positive ratios are needed, swashplate CVT could replace belt-driven variators. One potential application is combine harvester. Typically, the ICE in a combine harvester is operated constantly in its most efficient speed to power a hydraulic pump. While the drivetrain is hydrostatic, auxiliary drives are powered mechanically from the engine (fig. 39).

Figure 39. Pulleys and belts of Sampo-Rosenlew Comia C10 (Ahonen & Tuomi, 2016)

To accommodate for the varying thickness of the crop, threshing cylinder and fan are driven by a variable-diameter pulley. After some hours of operation, the variator belt becomes loose and the operator must check belt deflection (fig. 40)

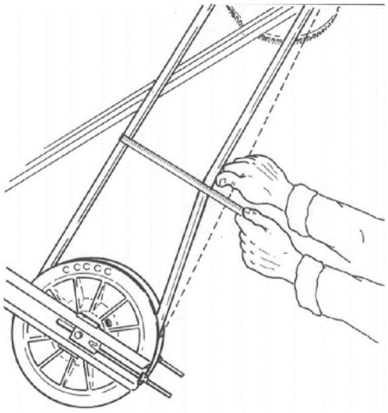

Figure 40. Checking combine harvester's variator belt deflection (Sampo-Rosenlew, 2011, p. 96)

Some combine harvesters have spring-loaded jockey pulley to maintain proper tension, but they are nevertheless subject to maintenance. If the threshing cylinder was rotated with a swashplate variator, this V-belt drive could be replaced with a chain, reducing the need of adjustment and other maintenance. Looking at all the pulleys and belts under the hood of a conventional combine harvester, some simplicity could be welcome.

LIST OF REFERENCES

Ahonen, H. & Tuomi, T. 2016. Erikoinen kulkuneuvo: Leikkuupuimuri – Sampo Rosenlew Comia C10. Article in Tekniikan Maailma, 16.9.2016. [Referred 4.12.2017]. Available at: https://tekniikanmaailma.fi/erikoinen-kulkuneuvo-leikkuupuimuri-sampo-rosenlew-comia-c10/

Bradley, B. 2007. Spiral-screwdriver, photograph, viewed 13.2.2017. https://commons.wikimedia.org/wiki/File:Spiral-screwdriver.jpg

Chiavette Unificate S.p.a. Spherical bearing – Rod end – ISO 12240: Technical Data. [Online datasheet]. Bologna, Italy: Chiavette Unificate S.p.a. [Referred 2.5.2017]. Available at: http://www.chiavette.com/rod-ends-and-spherical-bearings-technical-data

Cross, N. 2008. Engineering Design Methods: Strategies for Product Design. 4th edition. Chichester, West Sussex, United Kingdom: John Wiley & Sons Ltd, 2008. 217 pages. ISBN 978-0-470-51926-4.

DE 202012101273 U1. 2012. Einrichtung zur automatischen und kontinuierlichen Drehmoment- und Drehzahlveränderung einer Abtriebswelle je nach Fahrwiderstand. Ivanov, Konstantin S. Almaty, Kazakhstan. (Ivanov, K.S.) DE201220101273, 10.4.2012. Publ. 2.5.2012. 14 pages.

DIN 808: Machine tools; cardan joints; sizes, connecting dimensions, loading, installation. Edition 1984-08. Berlin, Germany: Deutsches Institut für Normung E.V.

DIN 71802: Ball joints with or without circlips. Edition 1992-01. Berlin, Germany: Deutsches Institut für Normung E.V.

DIN 71805: Angle joints; housings, circlips. Edition 1990-01. Berlin, Germany: Deutsches Institut für Normung E.V.

Esmail, E.L. & Hassan, S.S. 2008. Torque and Power-Flow Analyses of Two Input Epicyclic-Type Transmission Trains. ASME 2008 International Mechanical Engineering Congress and Exposition, Vol. 17: Transportation Systems, pp. 155–164. Boston, Massachusetts, United States of America: The American Society of Mechanical Engineers. DOI: 10.1115/IMECE2008-66377

Esmail, E.L. & Hassan, S.S. 2009. Two-Input Epicyclic-Type Transmission Trains with Application to Tandem Bicycling. Al-Qadisiya Journal For Engineering Sciences, 2(3).

Esmail, E.L. & Hassan, S.S. 2010. An Approach to Power-Flow and Static Force Analysis in Multi-Input Multi-Output Epicyclic-Type Transmission Trains. Journal of Mechanical Design, 132(1). DOI:10.1115/1.4000644.

Fallbrook Technologies. 2017. Traction Drive Overview. [Press release]. Cedar Park, Texas, United States of America: Fallbrook Intellectual Property Company LLC. [Referred 4.12.2017]. 13 pages. Available at: http://www.fallbrooktech.com/sites/default/files/videos/Traction%20Drive%20Overview.pdf

Ferguson, R.J. 1983. Short cuts for analyzing planetary gearing. Machine Design, 55(12), p. 55–58.

Fischetti, M. 2006. No More Gears. Scientific American, 294(1), p. 92–93. DOI:10.1038/scientificamerican0106-92

Freudenstein, F. 1971. An Application of Boolean Algebra to the Motion of Epicyclic Drives. Journal of Engineering for Industry, 93(1), p. 176.

Happian-Smith, J. 2001. An Introduction to Modern Vehicle Design. Butterworth-Heinemann Ltd. 2001. 601 pages. ASIN B00QQ6B93Y.

Ivanov, K.S. 2012. Synthesis of Toothed Continuously Variable Transmission (CVT). In: Lovasz EC., Corves B. (editors) Mechanisms, Transmissions and Applications. Mechanisms and Machine Science, vol 3. Dordrecht, Netherlands: Springer. DOI: 10.1007/978-94-007-2727-4_24

Koyo Seiko Co., Ltd. Sprag type one-way clutches. [Online datasheet]. Osaka, Japan: Koyo Seiko Co., Ltd.: 2016. [Referred 5.3.2017]. Available at: http://www.peilinbrg.com/html/images/car/4.pdf

Lynwander, P. 1983. Gear drive systems. New York, United States of America: Marcel Dekker, Inc., 1983. 407 pages. ISBN 0-8247-1896-8.

National Academy of Sciences. 2015. Cost, Effectiveness and Deployment of Fuel Economy Technologies for Light-Duty Vehicles. Washington, D.C., United States of America: The National Academies Press, 2015. 612 pages. ISBN 978-0-309-37388-3. http://www.nap.edu/catalog/21744/cost-effectiveness-and-deployment-of-fuel-economy-technologies-for-light-duty-vehicles. Viewed 22.6.2016.

Nissan Motor Co., Ltd. 1999. Extroid CVT For Application to Rear-Wheel-Drive Cars Powered by Large Engines. [Technical Note]. Tokyo, Japan: Nissan Motor Co., Ltd.: 1999. [Referred 15.11.2017]. Available at: http://www.nissan-global.com/PDF/tcvt_e.pdf

Pahl, G. & Beitz, W. 1990. Koneensuunnitteluoppi. 2nd edition. Helsinki, Finland: Metalliteollisuuden Kustannus Oy, 1990. 608 pages. ISBN 951-817-468-7.

Sampo-Rosenlew. 2011. SR2000 Combine Harvester Instruction Book. 154 pages. Available at: http://www.sampo-rosenlew.fi/upload/kirjat/Puimuri/SR2000/Manuaali/sr2000_manual_eng_02_2011.pdf

Sun, J., Fu, W., Lei, H., et al. 2012. Rotational swashplate pulse continuously variable transmission based on helical gear axial meshing transmission. Chinese Journal of Mechanical Engineering. Vol. 25 (6), pp. 1138–1143. doi:10.3901/CJME.2012.06.1138

US 1142574 A. 1915. Ratchet mechanism. Huck, J.T. San Francisco, California, United States of America. (Huck, J.T.) US 816492, 4.2.1914. Publ. 8.6.1915. 4 pages.

US 1409057. 1922. Mechanism for the interconversion of reciprocating and rotary motion. Michell, A.G.M. Melbourne, State of Victoria, Australia. (Michell, A.G.M.) US 239384, 11.6.1918. Publ. 7.3.1922. 7 pages.

US 1542668. 1925. Method and means for transmitting power. Constatinesco, G. Waybridge, Surrey, England. (Constantinesco, G.) US 570986, 26.6.1922. Publ. 16.6.1925. 10 pages.

US 1869189. 1932. Transmission. Eggert, G.B. Saginaw, Michigan, United States of America. (Eggert, G.B.) US 393949, 20.9.1929. Publ. 26.7.1932. 6 pages.

US 2691896 A. 1954. Variable Speed Power Transmission. Stageberg, S. Minneapolis, Minnesota, United States of America. (Stageberg, S.) US 121264, 14.10.1949. Publ. 19.10.1954. 7 pages.

US 390216 A. 1888. Speed regulating device. Evans, F. Somerville, Massachusetts, United States of America. (Evans, F.) US 271495, 23.4.1888. Publ. 2.10.1888.

US 4836090 A. 1989. Balanced variable stroke axial piston machine. Smith, Richard H. Birmingham, Michigan, United States of America. (General Motors Corporation, Detroit, Michigan, United States of America) US 206070, 13.6.1988. Publ. 6.6.1989.

US 785917 A. 1905. Speed varying mechanism. Reeves Pulley Co, Columbus, Indiana, United States of America. (Reeves, Milton O.) US 186121, 21.12.1903. Publ. 28.3.1905.

US 954977 A. 1910. Friction driving mechanism. Lambert, J.W. Madison, Indiana, United States of America. (Lambert, J.W.) US 471055, 7.1.1909. Publ. 12.4.1910.

Warko. 2008. Warko's working principle. [Website]. [Referred 15.11.2017]. Available at: http://www.warko.it/home/en/warko/principio-di-funzionamento

WO 2014/023926 A1. 2014. A continuously variable gear ratio changing system. Allen, M.C. Pollokshields, Glasgow, Great Britain. (Allen, M.C.) PCT/GB2013/000331, 7.8.2013. Publ. 13.2.2014. 30 pages.

Zero-Max. 2017. Adjustable Speed Drives. [Product page on Zero-Max's website]. [Referred 1.3.2017]. Available at: http://www.zero-max.com/adjustable-speed-drives-c-21-l-en.html