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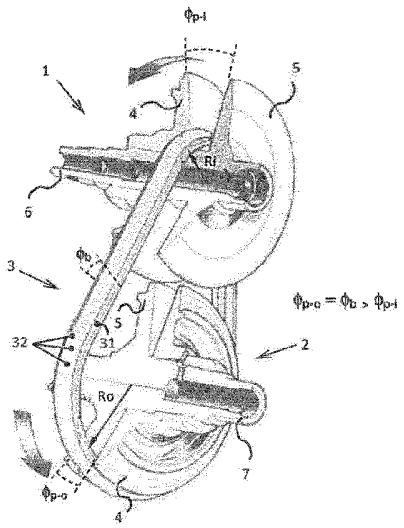
(54) **A continuously variable transmission with pulleys and a drive belt.**

(57) A continuously variable transmission comprises two pulleys (1, 2), each with two pulley discs (4, 5) that are mutually oriented at a pulley angle (Φp-i; Φp-o), and a drive belt (3) with an endless carrier (31) and with a number of transverse members (32) that each arrive in contact with the pulley discs (4, 5) via contact faces thereof that are mutually oriented at a belt angle (Φb). The pulley angle (Φp-i) of one (1) of the transmission pulleys (1, 2) is set smaller than the pulley angle (Φp-o) of the other one (2) of the transmission pulleys (1, 2) and the belt angle (Φb) is set essentially equal to the larger pulley angle (Φp-o) of the said other one pulley (2). This transmission is capable of transmitting a particularly high driving power with a particularly high efficiency.



NL C 1040573

A CONTINUOUSLY VARIABLE TRANSMISSION WITH PULLEYS AND A DRIVE BELT

The present disclosure relates to a continuously variable transmission provided with two pulleys and a drive belt, whereof each pulley includes a pair of conical discs that define a tapered circumferential groove and whereof the drive belt is arranged around such transmission pulleys, located, in part, in the pulley groove of the respective pulleys, while being clamped by and between the pulley discs thereof. The drive belt comprises a plurality of discrete transverse members for contacting the pulleys, as well as one or more endless, i.e. annular carriers for carrying and guiding the transverse members in the transmission. This type of drive belt is also known as a push belt.

The endless carrier of the drive belt is typically composed of a plurality of mutually nested, continuous flexible metal rings and is also known as a ring set. The endless carrier is at least partly inserted in a recess provided in the transverse members. In case the drive belt comprises only one endless carrier, such carrier is typically mounted in a central recess of the transverse members. However, usually the drive belt is provided with two endless carriers that are each mounted in a respective one of two recesses of the transverse members that open up towards a respective axial or lateral side of the drive belt.

The transverse members of the drive belt are slidingly arranged along the circumference of the endless carrier or carriers in a virtually continuous row, such that these members are able to transmit forces which are related to a movement of the drive belt. The transverse members have two main body surfaces that, at least partly, extend substantially parallel with respect to each other and that are separated from each other over the (local) thickness of the transverse member by a circumferential side surface thereof. As seen along the circumference of the carrier, the transverse members have a comparatively small dimension, i.e. thickness, such that a several hundred thereof are present in the drive belt. Adjoining transverse members must be able to tilt relative to one another, in order for the belt to follow a curved trajectory (part) in the pulley groove. To accommodate and control such relative tilting, at least one of the two main body surfaces of two adjoining transverse members in the drive belt that are mutually in contact is provided with a so-called rocking edge in the form of an axially extending, convexly rounded section of a respective main body surface.

Parts of the side surface of the transverse members, which parts are predominantly oriented in the axial direction, i.e. widthwise, are intended for frictionally engaging the pulley discs by being clamped there between. These pulley disc engaging parts of the side surface of the transverse members are denoted the contact faces and

are mutually oriented at an acute -so-called- belt angle, which belt angle closely matches a -so called- pulley angle, defined by and between the pulley discs of the pulleys.

The friction contact between the contact faces of the transverse members and the pulley discs allows a force to be transmitted there between, such that the drive belt can transfer a drive torque and a rotational movement from one pulley to the other. Furthermore, because of the conical shape of the pulley discs in combination with the belt angle between the contact faces of the transverse members of the drive belt, these transverse members also experience a radially outward directed force component of the clamping force that is exerted at/by each pulley. This latter force component puts the drive belt, in particular the endless carrier(s) thereof, under tension and also enables a displacement of the respective parts of drive belt located in the pulley grooves in mutually opposite radial directions in-between a smallest and a largest radial position thereof. Hereby, the actual, respective radial positions of the drive belt at the two pulleys determine a speed ratio of the transmission.

In the Japanese patent publication No. 2000-213609A it is observed that if the belt angle is increased relative to the pulley angle, a maximum drive torque that can be transmitted at a given clamping force favorably increases as well, at least initially. According to JP-2000-213609A, figure 5, such transmissible drive torque can be raised by approximately 18% at most, by applying a belt angle that is approximately 0.2 degrees larger than the pulley angle, at least relative to the drive torque that is transmissible by a transmission wherein the belt angle and the pulley angle are exactly the same.

Although such increase in transmissible torque is beneficial and highly sought after in the design of the present type transmission, is has been found to come only at the cost of a reduced overall efficiency of the transmission. In particular, it has been observed that, at least in a largest speed ratio of the transmission (which ratio is defined as a rotational speed of an output pulley of the transmission, which output pulley is associated with a load, divided by a rotational speed of an input pulley of the transmission, which input pulley is associated with an engine or motor driving the load), friction losses between the drive belt and the transmission pulleys increase as the belt angle is increased relative to the pulley angle. In practice, when the transmissible drive torque is maximized in the above-mentioned manner, i.e. by applying a belt angle that is 0.2 degrees larges than the pulley angle, a reduction of approximately 0.25% has been measured for the efficiency of the power transmission in the said largest speed ratio of the transmission. In particular in an automotive, e.g. passenger car, application of the transmission, this reduced efficiency is highly undesirable.

The present disclosure thus aims to reduce the said efficiency loss in the largest transmission ratio of the known transmission, while maintaining the relatively high transmissible torque thereof that is provided by applying a belt angle that is larger than the pulley angle. According to the present disclosure such aim is realized in the continuously variable transmission in accordance with the claim 1 hereinafter. In the latter, novel transmission the pulley angle defined by the input pulley, i.e. the input pulley angle, is set smaller than the pulley angle defined by the output pulley, i.e. the output pulley angle, and the belt angle is set larger than the input pulley angle and, preferably, essentially equal to the output pulley angle.

In the above arrangement, the belt angle is larger than the input pulley angle to provide the said known increase in the transmissible drive torque in accordance with the know art. Furthermore, the difference between the output pulley angle and the belt angle is decreased relative to the known transmission, to counteract the observed efficiency loss in the said largest speed ratio thereof the known transmission. This novel design of the transmission in terms of the pulley and belt angles departs from and relies on the observation that, on the one hand, the friction contact between the drive belt and the input pulley in the smallest speed ratio of the transmission is the dominant factor in relation to the said transmissible torque and that, on the other hand, the friction contact between the drive belt and the output pulley in the largest speed ratio of the transmission is the dominant factor in relation to the said overall efficiency of the transmission.

It was further observed that both such dominant friction contacts respectively occur at a smallest radial position of the drive belt in the pulley groove of the respective pulley. Therefore, in a more detailed embodiment of the above, novel transmission, the pulley angle of the input pulley is set to increase in relation to the radial position of the drive belt in the pulley groove. Preferably in this latter embodiment, the input pulley angle is set essentially equal to the belt angle at least in the largest radial position thereof in the pulley groove. By this feature the efficiency of the transmission may be improved even further.

It is noted that, in practice, the belt angle of the known drive belt varies between the transverse members as a result of the inaccuracies and tolerances occurring in manufacturing. For example, in the conventional transmission a typical value of the belt angle is 22.0°, which belt angle may vary between the transverse members thereof for 0.3 of a degree, e.g. 22.0°±0.3°. Within the context of the present disclosure, a nominal belt angle is defined as the average value of the angle defines between the contact faces of the individual transverse members of the drive belt. The pulley angle can typically be set within a more narrow tolerance band of less than ±0.3° around a nominal

value of the pulley angle, typically within ±0.2° or less.

The novel transmission design according to the present disclosure will now be explained in greater detail in the following description along one or more exemplary embodiments thereof and with reference to the attached drawing figures, whereof:

figure 1 provides a schematic perspective view of the continuously variable transmission with a drive belt running over two pulleys, which drive belt includes an endless carrier and a number of transverse members;

figure 2 shows a cross section of the known drive belt oriented in the circumference direction thereof;

figure 3 provides a width-wise oriented view of a transverse member of the known drive belt;

figure 4 provides a schematic representation of a transverse member of the drive belt that has arrived in friction contact with a pulley in accordance with the known art; and

figure 5 provides a schematic perspective view of a novel continuously variable transmission that is designed in accordance with the present disclosure.

In the drawing figures equal reference signs indicate equal or similar structures and/or parts.

The schematic illustration of a continuously variable transmission in Figure 1 shows a drive belt 3 which runs over two pulleys 1, 2 and which includes a flexible endless carrier 31 and an essentially contiguous row of transverse members 32 that are mounted on and arranged along the circumference of the endless carrier 31. The pulleys 1, 2 each include a pair of conical discs 4, 5 that define a tapered circumferential groove that opens towards the radial outside while enclosing an acute angle; the so-called pulley angle Φ_p. The drive belt 3 is, in part, located in these pulley grooves, while being clamped by and between the pulley discs 4, 5 of the respective pulley 1, 2.

In the illustrated configuration of the transmission, the upper pulley 1 will rotate more quickly than the lower pulley 2. By changing the distance between the two conical discs 4, 5 of the pulleys 1, 2, the radial positions or running radii Ri, Ro of the drive belt 3 at the pulleys 1, 2 are changed in mutually opposite radial directions and, as a result, a ratio between rotational speeds of the two pulleys 1, 2, which speed ratio is linked to the ratio between the respective running radii Ri, Ro, is varied. More in particular, the speed ratio is defined as a rotational speed of an output pulley 2 of the transmission, which output pulley 2 is associated with a load, divided by a rotational speed of an input pulley

1 of the transmission, which input pulley is associated 1 with an engine or motor driving the load.

In Figure 2, the drive belt 3 is shown in a cross section thereof facing in the circumference or length direction L of the belt 3, i.e. facing in a direction perpendicular to the axial or width direction W and the radial or height direction H thereof. This figure 2 shows the presence of two endless carriers 31, which are shown in cross-section in this figure 2, that carry and guide the transverse members 32 of the drive belt 3, whereof one transverse member 32 is shown in front elevation in this figure 2.

The transverse members 32 and the endless carriers 31 of the drive belt 3 are typically made of steel. The transverse members 32 take-up a clamping force exerted between the discs 4, 5 of each pulley 1, 2 via contact faces 37 thereof, one such contact face 37 being provided at each axial side of the transverse member 32. These contact faces 37 are mutually diverging in radial outward direction such that an acute angle is defined there between that is denoted the belt angle Φ_b of the drive belt 3.

The transverse members 32 are able to move, i.e. to slide along the endless carriers 31 in the circumference direction L, so that a torque can be transmitted between the transmission pulleys 1, 2 by the transverse members 32 pressing against one another and pushing each other forward along the endless carriers 31 in a direction of rotation of the drive belt 3 and the pulleys 1, 2. In the exemplary embodiment that is illustrated in this figure 2, the endless carriers 31 are composed of five individual endless bands each, which endless bands are mutually concentrically nested to form the endless carrier 31. In practice, the endless carriers 31 often comprise more than five endless bands, e.g. nine or twelve or possible even more.

The transverse member 32 of the drive belt 3, which is also shown in a side elevation in Figure 3, is provided with two cut-outs 33 located opposite one another, which cut-outs 33 each open towards a respective axial side of the transverse member 32 and each accommodate a (small section) of a respective endless carriers 31. A first or base portion 34 of the transverse member 32 thus extends radially inwards from the endless carriers 31, a second or middle portion 35 of the transverse member 32 is situated in between the endless carriers 31 and a third or top portion 36 of the transverse member 32 extends radially outwards from the endless carriers 31. The radially inner side of each cut-out 33 is delimited by a so-called bearing surface 42 of the base portion 34 of the transverse member 32, which bearing surface 42 faces radially outwards, generally in the direction of the top portion 36 of the transverse member 32, and contacts the inside of an endless carrier 31.

A first or rear surface 38 of the two main body surfaces 38, 39 of transverse

member 32 that face in mutually opposite circumference directions L, is essentially flat. The other or front main body surface 39 of the transverse member 32 is provided with a so-called rocking edge 18 that forms, in the radial direction H, the transition between an upper part of the front surface 39, extending essentially in parallel with its rear surface 38, and a lower part thereof that is slanted such that it extends towards the rear surface 38. In figure 2 the rocking edge 18 is indicated only schematically by way of a single line, however, in practice the rocking edge 18 is mostly provided in the shape of a convexly curved transition surface. The said upper part of the transverse member 32 is thus provided with an essentially constant dimension between the main body surfaces 38, 39, i.e. as seen in the circumference direction L, which dimension is typically referred to as the thickness of the transverse member 32.

Referring back to figure 1, the transmission is depicted in its smallest speed ratio, wherein the drive belt 3 is located at its smallest running radius Ri at the input pulley 1 and at its largest running radius Ro at the output pulley 2, such that the rotational speed of the input pulley 1 will be higher than that of the output pulley 2. In particular in this smallest speed ratio of the transmission, the torque that is maximally transmissible by the transmission is an important characteristic in the automotive application thereof. According to JP-2000-213609A, to maximize such transmissible torque, the belt angle Φ_b of the known transmission is set slightly larger than the pulley angle Φ_p thereof, as is schematically illustrated in figure 4 in relation to the contact between one contact face 37 of the drive belt 3 and one pulley disc 4; 5 of the pulleys 1, 2.

According to JP-2000-213609A, relative to the normally applied, nominal pulley angle of 22°, the belt angle should not be set equal to such nominal pulley angle but approximately 0.2° larger, i.e. the nominal belt angle should amount to, approximately, 22.2° in order to maximize the transmissible torque. In particular, according to JP-2000-213609A the transmissible torque could thereby be increased by around 18%. According to the present disclosure, however, the exact optimum value of such difference between the nominal value of the belt angle Φ_b and the nominal value of the pulley angle Φ_p will depend on many factors, such as these nominal values as such and a (relative) axial stiffness of the design and shape of the transverse members 32 and the pulley discs 4, 5. For example, in a contemporary transmission design the maximum transmissible torque was realized with a belt angle Φ_b of 22.6° relative to the pulley angle Φ_p of 22.0°.

This known transmission comes with the disadvantage that a (friction) loss associated with the power transfer between the pulleys 1, 2 by means of the drive belt 3 during operation of the transmission was found to increase, as the difference between

the belt angle Φ_b and the pulley angle Φ_p increases. In order to favorably reduce such power losses and/or to favorably improve the operating efficiency of the continuously variable transmission, it is presently proposed to set only the nominal pulley angle Φ_p of the input pulley 1, i.e. the output pulley angle $\Phi_{p\text{-}i}$, smaller than the belt angle Φ_b, while the nominal pulley angle Φ_p of the output pulley 2, i.e. the output pulley angle $\Phi_{p\text{-}o}$ is set essentially equal to the nominal belt angle Φ_b. This particular, novel transmission design in accordance with the present disclosure is schematically illustrated in figure 5. In this particular transmission design, a friction loss between the drive belt 3 and the output pulley 2 is minimized, while the advantage of the maximized transmissible torque in the largest speed ratio of the transmission that is determined by the friction contact between the drive belt 3 and the input pulley 1 is maintained.

A practical range for the difference between the input pulley angle $\Phi_{p\text{-}i}$ and the output pulley angle $\Phi_{p\text{-}o}$ and/or the belt angle Φ_b is given by at least -0.2° up to -1.2° (wherein the minus sign means that the input pulley angle $\Phi_{p\text{-}i}$ is smaller than the output pulley angle $\Phi_{p\text{-}o}$ and/or the belt angle Φ_b). Furthermore, in this numeric example, the input pulley angle amounts to 21.4° or 0.6° less than the pulley angle of the output pulley angle and/or the drive belt angle of 22.0°.

The present disclosure, in addition to the entirety of the preceding description and all details of the accompanying figures, also concerns and includes all the features of the appended set of claims. Bracketed references in the claims do not limit the scope thereof, but are merely provided as non-binding examples of the respective features. The claimed features can be applied separately in a given product or a given process, as the case may be, but it is also possible that to apply any combination of two or more of such features therein.

The invention(s) represented by the present disclosure is (are) not limited to the embodiments and/or the examples that are explicitly mentioned herein, but also encompasses amendments, modifications and practical applications thereof, in particular those that lie within reach of the person skilled in the relevant art.

CONCLUSIES

1. Continu variabele overbrenging met een ingangspoelie (1), een uitgangspoelie (2) en een drijfriem (3), welke ingangs- en uitgangspoelies (1, 2) elk zijn voorzien van twee, in hoofdzaak conische schijven (4, 5), die onderling onder een poeliehoek (Φ_p; Φ_{p-i}; Φ_{p-o}) zijn geplaatst en die in contact staan met contactvlakken (37) aan weerszijden van de drijfriem (3), welke contactvlakken (37) aan weerszijden van de drijfriem (3) onderling onder een bandhoek (Φ_b) zijn geplaatst, waarin het contact tussen enerzijds de drijfriem (3) en anderzijds de ingangspoelie (1) of de uitgangspoelie (2) tussen een kleinste en een grootste radiale positie ten opzichte van de respectievelijke poelieschijven (4, 5) kan variëren, met het kenmerk, dat, tenminste ter plaatse van de kleinste radiale positie van de drijfriem (3) ten opzichte van de respectievelijke poelieschijven (4, 5), de poeliehoek (Φ_{p-i}) van de ingangspoelie (1) kleiner is dan de poeliehoek (Φ_{p-o}) van de uitgangspoelie (2).

2. Continu variabele overbrenging volgens de conclusie 1, met het kenmerk, dat ter plaatse van de kleinste radiale positie van de drijfriem (3) ten opzichte van de respectievelijke poelieschijven (4, 5), de poeliehoek (Φ_{p-i}) van de ingangspoelie (1) minimaal 0,2 en maximaal 1,2 graden kleiner is dan de poeliehoek (Φ_{p-o}) van de uitgangspoelie (2).

3. Continu variabele overbrenging volgens de conclusie 1, met het kenmerk, dat een nominale waarde van de poeliehoek (Φ_{p-o}) van de uitgangspoelie (2) ongeveer 22 graden bedraagt en dat ter plaatse van de kleinste radiale positie van de drijfriem (3) ten opzichte van de respectievelijke poelieschijven (4, 5), de nominale poeliehoek (Φ_{p-i}) van de ingangspoelie (1) ongeveer 21,4 graden bedraagt.

4. Continu variabele overbrenging volgens één der voorgaande conclusies, met het kenmerk, dat, tenminste ter plaatse van de grootste radiale positie van de drijfriem (3) ten opzichte van de respectievelijke poelieschijven (4, 5), de poeliehoek (Φ_{p-i}) van de ingangspoelie (1) althans nagenoeg gelijk is aan de poeliehoek (Φ_{p-o}) van de uitgangspoelie.

5. Continu variabele overbrenging volgens één der voorgaande conclusies, met het kenmerk, dat de bandhoek (Φ_b) tenminste gelijk is aan de poeliehoek (Φ_{p-i}) van de

ingangspoelie (1) en ten hoogste gelijk is aan de poeliehoek ($\Phi_{p\text{-}o}$) van de uitgangspoelie (2).

6. Continu variabele overbrenging volgens één der voorgaande conclusies, met het kenmerk, dat de bandhoek (Φ_b) althans nagenoeg gelijk is aan de poeliehoek ($\Phi_{p\text{-}o}$) van de uitgangspoelie (2) ter plaatse van de kleinste radiale positie van de drijfriem (3) ten opzichte van de poelieschijven (4, 5) daarvan.

7. Continu variabele overbrenging volgens één der voorgaande conclusies, met het kenmerk, dat de drijfriem (3) is voorzien van een in zichzelf gesloten, i.e. eindloze drager (31) en een aantal, verschuifbaar op de eindloze drager (31) aangebrachte, elkaar langs de omtrek van de eindloze drager (31) opvolgende dwarsdelen (32), waarvan elk dwarsdeel (32) is voorzien van een voorvlak (39) en van een achtervlak (38), waartussen het dwarsdeel (32) zich in dikterichting uitstrekt en waartussen elk dwarsdeel (32) in breedterichting aan weerszijden van een contactvlak (37) is voorzien, bestemd voor het wrijvingscontact met de ingangs- en de uitgangspoelies (1, 2) van de overbrenging, waarbij de contactvlakken (37) van elk dwarsdeel (32) onderling onder een hoek staan en waarbij de genoemde bandhoek (Φ_b), althans een nominale waarde daarvoor, het rekenkundig gemiddelde van de hoek tussen de contactvlakken (37) van alle dwarsdelen (32) van de drijfriem (3) betreft.

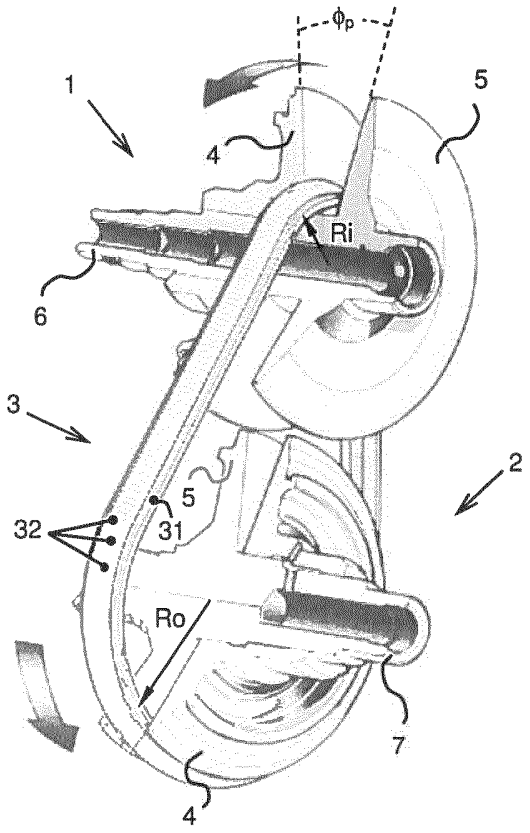


FIG. 1

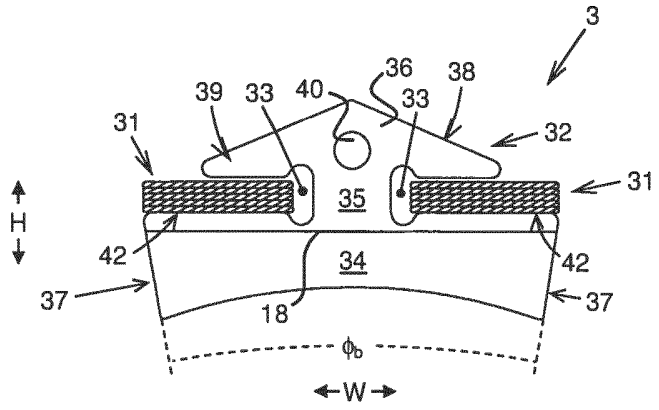


FIG. 2

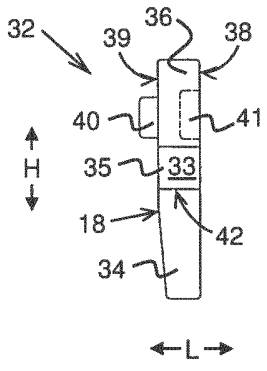


FIG. 3

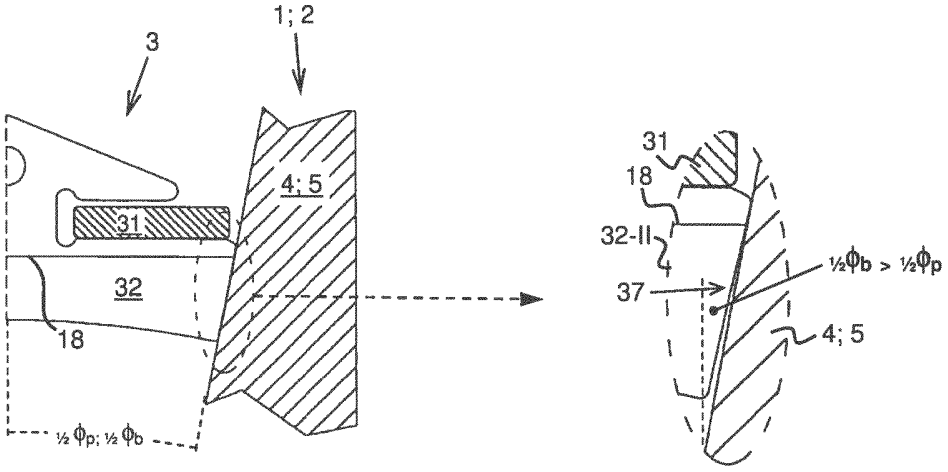

3
1; 2
4; 5
31
32
18
½ ϕp; ½ ϕb
31
18
32-II
37
½ϕb > ½ϕp
4; 5

FIG. 4

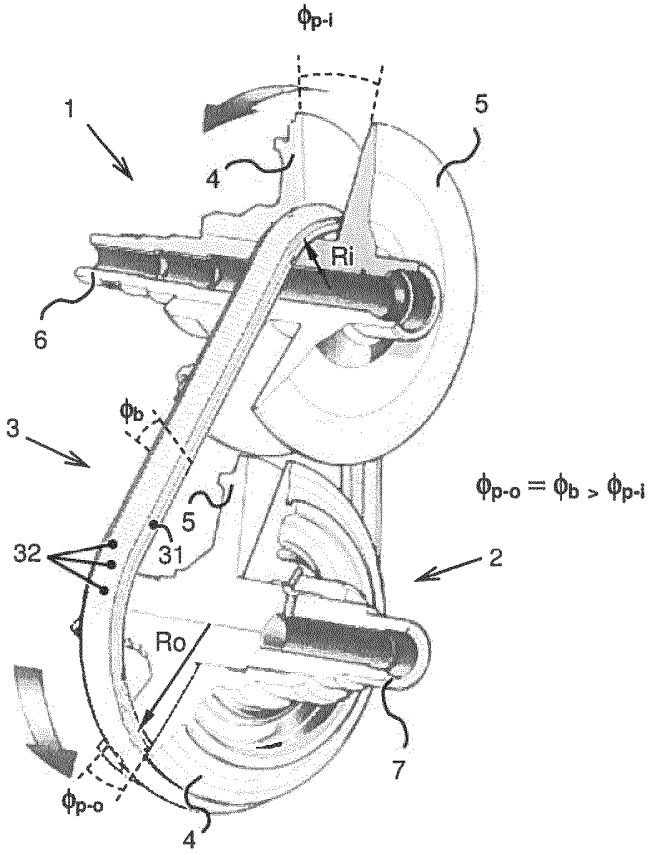

ϕp-i
1
5
4
Ri
6
ϕb
3
ϕp-o = ϕb > ϕp-i
5
32
31
2
Ro
7
ϕp-o
4

FIG. 5

SAMENWERKINGSVERDRAG (PCT)

RAPPORT BETREFFENDE NIEUWHEIDSONDERZOEK VAN INTERNATIONAAL TYPE

IDENTIFICATIE VAN DE NATIONALE AANVRAGE	KENMERK VAN DE AANVRAGER OF VAN DE GEMACHTIGDE
	R.353362/PD
Nederlands aanvraag nr. **1040573**	Indieningsdatum **24-12-2013**
	Ingeroepen voorrangsdatum
Aanvrager (Naam) **ROBERT BOSCH GMBH**	
Datum van het verzoek voor een onderzoek van internationaal type **01-03-2014**	Door de Instantie voor Internationaal Onderzoek aan het verzoek voor een onderzoek van internationaal type toegekend nr. **SN61528**

I. CLASSIFICATIE VAN HET ONDERWERP (bij toepassing van verschillende classificaties, alle classificatiesymbolen opgeven)

Volgens de internationale classificatie (IPC)

F16G5/16;F16H55/56

II. ONDERZOCHTE GEBIEDEN VAN DE TECHNIEK

Onderzochte minimumdocumentatie

Classificatiesysteem	Classificatiesymbolen
IPC	**F16G;F16H**

Onderzochte andere documentatie dan de minimum documentatie, voor zover dergelijke documenten in de onderzochte gebieden zijn opgenomen

III.	☐	**GEEN ONDERZOEK MOGELIJK VOOR BEPAALDE CONCLUSIES**	(opmerkingen op aanvullingsblad)
IV.	☐	**GEBREK AAN EENHEID VAN UITVINDING**	(opmerkingen op aanvullingsblad)

ONDERZOEKSRAPPORT BETREFFENDE HET RESULTAAT VAN HET ONDERZOEK NAAR DE STAND VAN DE TECHNIEK VAN HET INTERNATIONALE TYPE

Nummer van het verzoek om een onderzoek naar de stand van de techniek
NL 1040573

A. CLASSIFICATIE VAN HET ONDERWERP
INV. F16G5/16 F16H55/56
ADD.

Volgens de Internationale Classificatie van octrooien (IPC) of zowel volgens de nationale classificatie als volgens de IPC.

B. ONDERZOCHTE GEBIEDEN VAN DE TECHNIEK

Onderzochte miminum documentatie (classificatie gevolgd door classificatiesymbolen)
F16G F16H

Onderzochte andere documentatie dan de mimimum documentatie, voor dergelijke documenten, voor zover dergelijke documenten in de onderzochte gebieden zijn opgenomen

Tijdens het onderzoek geraadpleegde elektronische gegevensbestanden (naam van de gegevensbestanden en, waar uitvoerbaar, gebruikte trefwoorden)
EPO-Internal

C. VAN BELANG GEACHTE DOCUMENTEN

Categorie °	Geciteerde documenten, eventueel met aanduiding van speciaal van belang zijnde passages	Van belang voor conclusie nr.
X	JP 2002 130408 A (NISSAN MOTOR) 9 mei 2002 (2002-05-09) * samenvatting; figuren 1-8 *	1-7
X	EP 1 158 210 A1 (DOORNES TRANSMISSIE BV [NL]) 28 november 2001 (2001-11-28) * alineas [0015], [0029]; figuren 1-6 *	1-7

☐ Verdere documenten worden vermeld in het vervolg van vak C.

☒ Leden van dezelfde octrooifamilie zijn vermeld in een bijlage

° Speciale categorieën van aangehaalde documenten

"A" niet tot de categorie X of Y behorende literatuur die de stand van de techniek beschrijft

"D" in de octrooiaanvrage vermeld

"E" eerdere octrooi(aanvrage), gepubliceerd op of na de indieningsdatum, waarin dezelfde uitvinding wordt beschreven

"L" om andere redenen vermelde literatuur

"O" niet-schriftelijke stand van de techniek

"P" tussen de voorrangsdatum en de indieningsdatum gepubliceerde literatuur

"T" na de indieningsdatum of de voorrangsdatum gepubliceerde literatuur die niet bezwarend is voor de octrooiaanvrage, maar wordt vermeld ter verheldering van de theorie of het principe dat ten grondslag ligt aan de uitvinding

"X" de conclusie wordt als niet nieuw of niet inventief beschouwd ten opzichte van deze literatuur

"Y" de conclusie wordt als niet inventief beschouwd ten opzichte van de combinatie van deze literatuur met andere geciteerde literatuur van dezelfde categorie, waarbij de combinatie voor de vakman voor de hand liggend wordt geacht

"&" lid van dezelfde octrooifamilie of overeenkomstige octrooipublicatie

Datum waarop het onderzoek naar de stand van de techniek van internationaal type werd voltooid	Verzenddatum van het rapport van het onderzoek naar de stand van de techniek van internationaal type
31 oktober 2014	

Naam en adres van de instantie	De bevoegde ambtenaar
European Patent Office, P.B. 5818 Patentlaan 2 NL - 2280 HV Rijswijk Tel. (+31-70) 340-2040, Fax: (+31-70) 340-3016	Simens, Mark Phil

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ONDERZOEKSRAPPORT BETREFFENDE HET RESULTAAT VAN HET ONDERZOEK NAAR DE STAND VAN DE TECHNIEK VAN HET INTERNATIONALE TYPE

Informatie over leden van dezelfde octrooifamilie

Nummer van het verzoek om een onderzoek naar de stand van de techniek
NL 1040573

In het rapport genoemd octrooigeschrift	Datum van publicatie	Overeenkomend(e) geschrift(en)	Datum van publicatie
JP 2002130408 A	09-05-2002	JP 3675329 B2	27-07-2005
		JP 2002130408 A	09-05-2002
EP 1158210 A1	28-11-2001	DE 60102159 D1	01-04-2004
		DE 60102159 T2	27-01-2005
		EP 1158210 A1	28-11-2001
		EP 1299659 A1	09-04-2003
		JP 4798929 B2	19-10-2011
		JP 2003535289 A	25-11-2003
		US 2003144097 A1	31-07-2003
		WO 0192763 A1	06-12-2001

WRITTEN OPINION

File No.	Filing date *(day/month/year)*	Priority date *(day/month/year)*	Application No.
SN61528	24.12.2013		NL1040573

International Patent Classification (IPC)
INV. F16G5/16 F16H55/56

Applicant
ROBERT BOSCH GMBH

This opinion contains indications relating to the following items:

- ☒ Box No. I Basis of the opinion
- ☐ Box No. II Priority
- ☐ Box No. III Non-establishment of opinion with regard to novelty, inventive step and industrial applicability
- ☐ Box No. IV Lack of unity of invention
- ☒ Box No. V Reasoned statement with regard to novelty, inventive step or industrial applicability; citations and explanations supporting such statement
- ☐ Box No. VI Certain documents cited
- ☒ Box No. VII Certain defects in the application
- ☒ Box No. VIII Certain observations on the application

	Examiner
	Simens, Mark Phil

Application number

WRITTEN OPINION

NL1040573

Box No. I Basis of this opinion

1. This opinion has been established on the basis of the latest set of claims filed before the start of the search.

2. With regard to any **nucleotide and/or amino acid sequence** disclosed in the application and necessary to the claimed invention, this opinion has been established on the basis of:

   a. type of material:

   ☐ a sequence listing

   ☐ table(s) related to the sequence listing

   b. format of material:

   ☐ on paper

   ☐ in electronic form

   c. time of filing/furnishing:

   ☐ contained in the application as filed.

   ☐ filed together with the application in electronic form.

   ☐ furnished subsequently for the purposes of search.

3. ☐ In addition, in the case that more than one version or copy of a sequence listing and/or table relating thereto has been filed or furnished, the required statements that the information in the subsequent or additional copies is identical to that in the application as filed or does not go beyond the application as filed, as appropriate, were furnished.

4. Additional comments:

Box No. V Reasoned statement with regard to novelty, inventive step or industrial applicability; citations and explanations supporting such statement

1. Statement

Novelty	Yes:	Claims	
	No:	Claims	1-7
Inventive step	Yes:	Claims	
	No:	Claims	1-7
Industrial applicability	Yes:	Claims	1-7
	No:	Claims	

2. Citations and explanations

see separate sheet

WRITTEN OPINION

Box No. VII Certain defects in the application

see separate sheet

Box No. VIII Certain observations on the application

see separate sheet

Re Item V

Reasoned statement with regard to novelty, inventive step or industrial applicability; citations and explanations supporting such statement

1 Prior art

Reference is made to the following documents:

D1 JP 2002 130408 A (NISSAN MOTOR) 9 mei 2002 (2002-05-09)

D2 EP 1 158 210 A1 (DOORNES TRANSMISSIE BV [NL]) 28 november 2001 (2001-11-28)

2 Novelty objections

The present application does not meet the criteria of patentability, because the subject-matter of claim 1 is not new.

2.1 Claim 1

D1 discloses

Continu variabele overbrenging met een ingangspoelie (see figure 1, reference sign 20), een uitgangspoelie (see figure 1, reference sign 30) en een drijfriem (see figure 1, reference sign 40), welke ingangs- en uitgangspoelies (20 and 30) elk zijn voorzien van twee, in hoofdzaak conische schijven (see figure 1, reference signs 21,23,31,33), die onderling onder een poeliehoek (see figure 6) zijn geplaatst en die in contact staan met contactvlakken aan weerszijden van de drijfriem (see figure 1), welke contactvlakken aan weerszijden van de drijfriem onderling onder een bandhoek (see figure 8) zijn geplaatst, waarin het contact tussen enerzijds de drijfriem (40) en anderzijds de ingangspoelie (20) of de uitgangspoelie (30) tussen een kleinste en een grootste radiale positie ten opzichte van de respectievelijke poelieschijven kan variëren (see abstract), waarbij tenminste ter plaatse van de kleinste radiale positie van de drijfriem (40) ten opzichte van de respectievelijke poelieschijven, de poeliehoek van de ingangspoelie kleiner is dan de poeliehoek van de uitgangspoelie (see abstract and figure 6).

D2 also discloses all the features of claim 1 (see figures, and description, paragraph [0029], lines 35-40).

2.2 Dependent claims

Dependent claims 2-7 do not contain any features which, in combination with the features of any claim to which they refer, meet the requirements in respect of novelty and/or inventive step either because the additional features of these claims are also disclosed for similar assemblies (see documents D1-D2 and the corresponding

passages cited in the search report) or because they relate to simple constructional details that the skilled person would normally contemplate without the exercise of any inventive activity.

Re Item VII

Certain defects in the application

3

3.1

The relevant background art disclosed in D1 and D2 is not mentioned in the description, nor are these documents identified therein.

Re Item VIII

Certain observations on the application

4

4.1

It is clear from the description on page 3, lines 8 and 9 that the features of claim 5 is essential to the definition of the invention.

Since independent claim 1 does not contain this feature it does not meet the requirement of clarity that any independent claim must contain all the technical features essential to the definition of the invention.