© Strojniški vestnik (43) št. 7-8, str. 299-308, 1997
Tiskano v Sloveniji. Vse pravice pridržane.
UDK 629.114.6:621.3
629.331.5

SV:0039-2480(97)4-R3

© Journal of Mechanical Engineering (43) No. 7-8, pp. 299-308, 1997
Printed in Slovenia. All rights reserved.
UDC 629.114.6:621.3
629.331.5

Razvoj menjalnika za električni avtomobil
Development of New Electric Car Gearbox

STANISLAV PEHAN - JOŽE FLAŠKER - IZIDOR RUPNIK

Pogonski prenos avtomobila mora biti prilagojen lastnostim motorja in zahtevam, ki jih mora izpolnjevati vozilo. Sodoben bencinski pogonski agregat osebnega avtomobila zahteva vsaj torno sklopko, 5-stopenjski menjalnik in razdelilno gonilo. Če za pogonski agregat avtomobila uporabimo 3-fazni elektromotor, ki ima po pravilu manjšo moč, vendar višji največji moment in popolnoma drugo obliko momentne značilnosti, potem sklepamo, da je običajni avtomobilski prenos treba prirediti tem novim karakteristikam pogonskega agregata. Najprimernejša rešitev tega problema je konstrukcija novega prenosa; brez sklopke in velikega števila prestav. Preprost dvostopenjski planetni menjalnik z integriranim razdelilnim gonilom v veliki meri ustreza lastnostim električnega pogoskega agregata, električnemu avtomobilu pa daje optimalne vozne karakteristike.

Ključne besede: avomobili električni, menjalniki, razvoj, gonila planetna, transmisije

Transmission between the engine and driving wheels depends on the characteristics of the power engine and on the requirement, which the vehicle must meet. Modern cars with internal combustion engine require at least a friction clutch, 5-speed gearbox and differential drive. This paper deals with a substitute power engine, the electric 3-phase AC induction motor, which has a lower power rating, but higher maximum torque and completely different torque characteristic. Vehicles fitted with the electric engine require adaption of the transmission line. The best solution is a completely new transmission line without the clutch and only with 2-speed gearbox. A simple planetary gear drive, integrated with the differential drive, fulfils a majority of demands related to the vehicle drive behaviour and easily meets the requirements of the electric engine.

Key words: electric vehicle, gearboxes, development, planetary drive, transmission line

0 UVOD

Pojavlja se vedno večja želja, da bi avtomobile opremili s takšnim motorjem, ki bi čim manj onesnaževal okolje.

Za to rešitev se ponuja elektromotor, ki ima obilo prednosti, saj ima primerno momentno karakteristiko, je tih in poleg toplote v okolico ne oddaja nezaželenih plinov. Žal pa se zdi, da na nek preprost, cenen in hiter način v vozilo nikakor ne moremo uskladiščiti dovolj energije, ki bi zadoščala, da bi vozilo prepeljalo vsaj 400 km ali več, kar se pač dandanes od avtomobila pričakuje. Toda razvoj gre vseeno v smer vgradnje elektromotorja v vozila. Očitno je treba računati s tržno nišo, ki jo predstavljajo vozila v mestnem prometu, kjer je emisija še kako nezaželena.

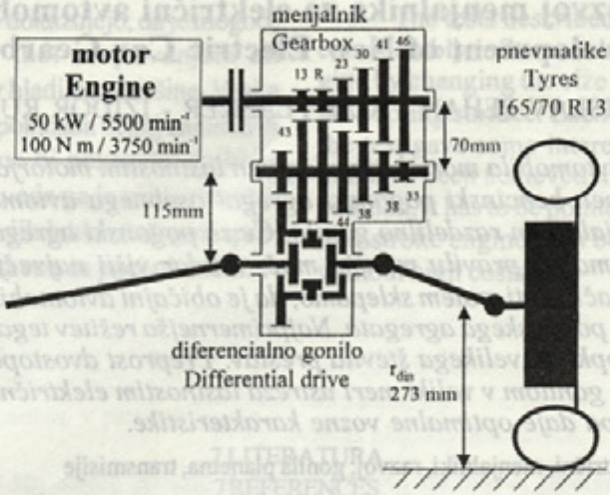
Če projektantu vozila uspe premagati problem shranjevanja energije in krmiljenja elektromotorja, nastane problem v transmisijski verigi, ki je v svoji najpreprostejši verziji taka, kakršno prikazuje slika 1. Snovanje električnega avtomobila se prične običajno tako, da se preprosto odstrani motor z notranjim zgorevanjem, na njegovo mesto pa postavi elektromotor. Toda električni avtomobil sploh ne potrebuje sklopke, niti ne potrebuje množice prestav. Te višje prestave so še posebej nepotrebne, saj vozilo zaradi manjše moči ne dosega hitrosti niti 100 km/h.

0 INTRODUCTION

The electric car is a device which can probably help in solving the environmental pollution crisis.

The electric engine has many advantages. The engine has more adequate torque characteristics, it runs quietly, and electric engines do not emit poisoned gases in the surroundings. Unfortunately, as yet it is impossible to accumulate in a simple manner enough energy in the vehicle, which could provide sufficient power for the car to run at least 400 km, which is today's expectation. Nevertheless, a significant effort is spent on the development of electric vehicle. Obviously there is a quite significant niche in the market, which is represented by the city vehicles in which it is desirable to minimise the pollution level.

Under the assumption that designers can solve the problems related to the energy storage batteries and to electric engine control, the last remaining problem is appropriate design of the transmission line. In the simplest manner, the transmission line is shown in figure 1. The reconstruction of the ordinary vehicle with the internal combustion engine starts by simply changing the engines. Instead of the internal combustion engine a new electric engine is fitted to the vehicle. However, the electric vehicle does not need the clutch and so many gearbox speeds. Higher gearbox speeds are obviously unusable, because of the low power level of the electric motor, which can hardly drive the car to speeds of over 100 km/h.



Sl. 1. *Enostaven pogonski prenos osebnega vozila z motorjem z notranjim zgorevanjem, Škoda felicia 50 kW*

Fig. 1. *Simple transmission line of passenger car with the internal combustion engine, Škoda felicia 50 kW*

Zaželeno je, da bi bil električni avtomobil vozilo, pri kakterem prestavne ročice in pedala sklopke ne bi bilo. Prestava menjalnika naj bi se samodejno sprožila na električni impulz vozilskega računalnika. Sistem za oddvajanje motorja od prenosa ni potreben. Prestava nazaj v menjalniku ni potrebna, saj se pri vožnji nazaj zavrti elektromotor v drugo smer.

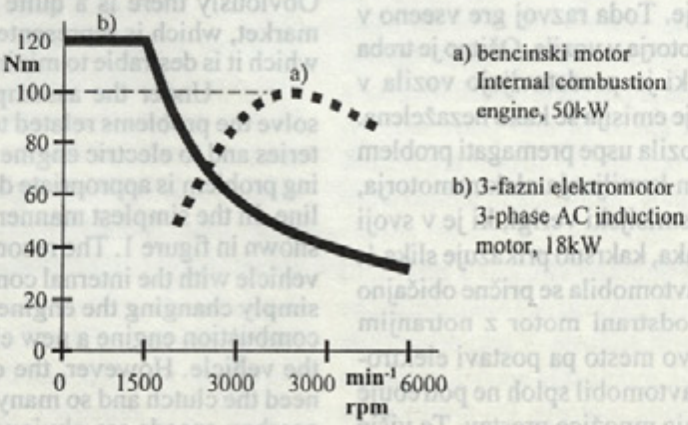
1 VOZNE KARAKTERISTIKE ELEKTRIČNEGA AVTOMOBILA

Električni avtomobili, ki so napravljeni tako, da so snovalci le nadomestili bencinski pogonski agregat z elektromotorjem, vozijo po Mariboru že nekaj let. Takšen avtomobil smo vzeli za osnovo, za katero bi zasnovali čimbolj primeren prenos. Pomemben vhodni podatek za načrtovanje prenosa je zunanja momentna karakteristika pogonskega motorja in prav ta parameter se pri 3-faznih elektromotorjih z močjo 18 kW bistveno razlikuje od 50 kW močnega bencinskega motorja (sl. 2).

It is apparent that a completely new gearbox needs to be designed for the electric vehicle. A perfect electric car does not need the gearbox control stick and the clutch pedal. A suitable gearbox speed could be activated by an electric signal from the onboard computer. Backward gearbox speed is unnecessary, because the electric motor can simply change the rotation direction.

1 DRIVING CHARACTERISTICS OF THE ELECTRIC CAR

A couple of electric cars are already running in Maribor city and in the neighbouring area. These cars are produced only by replacing the engines and they served as a basis for new transmission development. The most important input property of the electric engine is the maximum torque characteristic. These characteristics are very different for an electric 3-phase AC induction motor (18 kW) and internal combustion engine (50 kW). The difference is shown in figure 2.

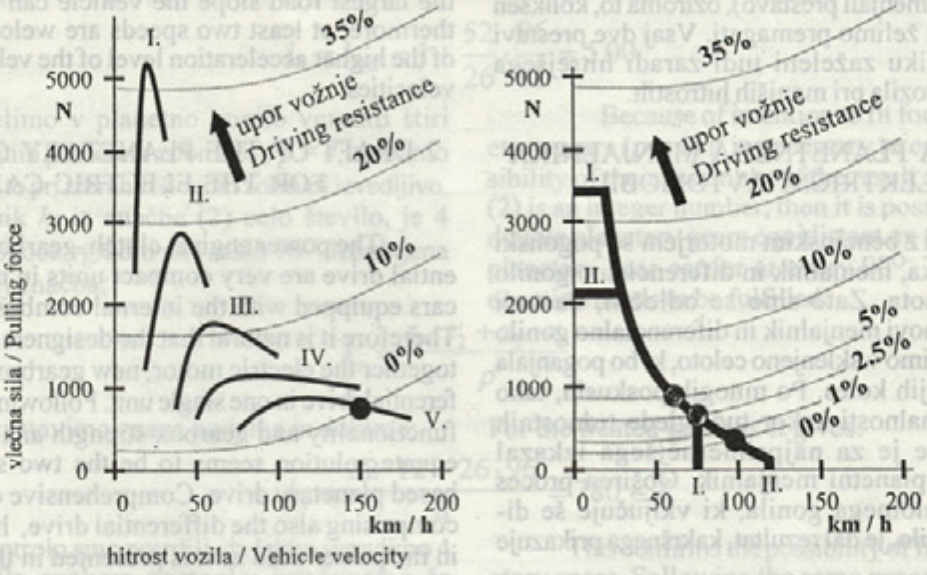


Sl. 2. *Momentna karakteristika*

Fig. 2. *Torque characteristic*

Vozni diagrami bencinskega motorja (sl. 3a) kažejo, da vozilo na vodoravni poti doseže največjo hitrost okoli 150 km/h, v prvi prestavi pa premaga 35-odstotni klanec.

Diagrams of pulling forces of the four-stroke internal combustion engine (Fig. 3a), show that vehicle equipped with this engine can reach about 150 km/h on a perfectly horizontal road and in the first speed it can handle a 35-percent slope.



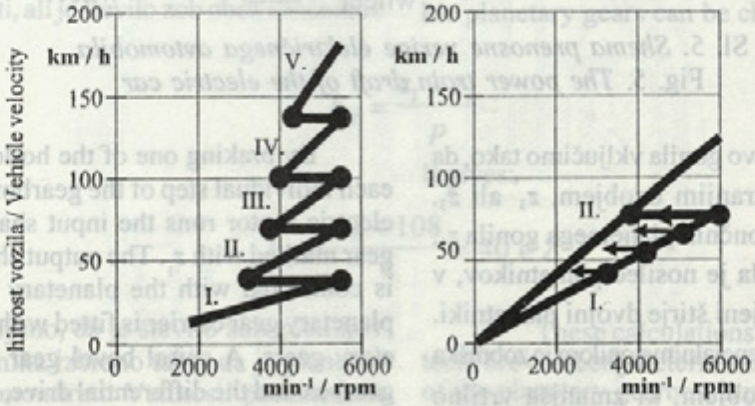
a) motor z notranjim zgorevanjem
Internal combustion engine, 50 kW

b) trifazni elektromotor
3-phase AC induction engine , 18 kW

Sl. 3 . *Vozni diagram vozila Škoda felicia*
Fig. 3. *Diagram of pulling forces of the Škoda felicia vehicle*

Vozni diagrami vozila Škoda felicia, ki so napravljeni z upoštevanjem električnega pogonskega agregata, (sl. 3b), kažejo na prvi pogled slabše vozne lastnosti. Vozilo doseže kvečjemu hitrost 100 km/h in premaga le 23-odstotno strmino, kar pa je za mestno vožnjo čisto dovolj. Predpostavljeni sta dve prestavi menjalnika, saj se motor lahko vrti le do 6000 min^{-1}. Če bi se elektromotor lahko vrtel hitreje, na primer do 9000 min^{-1}, potem menjalnika sploh ne bi potrebovali.

Diagrams of the pulling forces of the vehicle equipped with the 3-phase AC induction engine (Fig.3b), show weaker driving characteristics. The top velocity of the vehicle hardly reaches 100 km/h and the steepest slope is 23-percent. However, those properties meet all the requirements characteristic for city traffic. Because the top speed of the AC induction engine is only around 6000 rpm, it is necessary to design a two-speed gearbox. However, in the case of higher engine speeds in the range of 9000 rpm, the gearbox would become unnecessary.



a) motor z notranjim zgorevanjem
Internal combustion engine, 50 kW

b) trifazni elektromotor
3-phase AD induction engine, 18 kW

Sl. 4 *Hitrostni diagram Škode felicie*
Fig. 4. *Speed diagram of the Škoda felicia*

Analiza tako imenovanega hitrostnega diagrama (sl. 4), pokaže, da razmerje prestav pri pogonu vozila z elektromotorjem, nima nobenega vpliva na vozne lastnosti vozila. Edini dejavnik, ki vpliva na velikost posamezne prestave v menjalniku, je udobnost vožnje (kdaj bi menjali prestavo), oziroma to, kolikšen največji vzpon želimo premagati. Vsaj dve prestavi sta v menjalniku zaželeni tudi zaradi hitrejšega pospeševanja vozila pri manjših hitrostih.

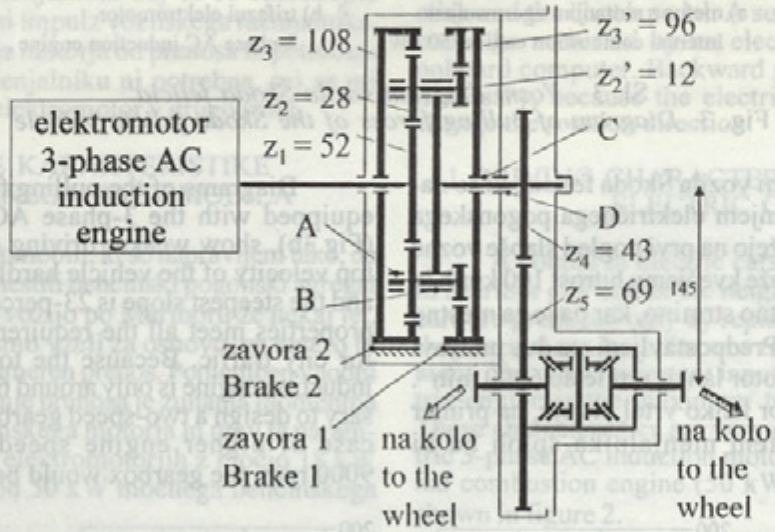
The analysis of the speed diagram shown in figure 4 lead to the conclusion that the partitioning of the gearbox speeds has no influence on the driving performance of the vehicle. The gear ratio of each individual speed influences the driving comfort and the largest road slope the vehicle can handle. Furthermore, at least two speeds are welcome in view of the higher acceleration level of the vehicle at lower velocities.

2 ZASNOVA PLANETNEGA MENJALNIKA ZA ELEKTRIČNI AVTOMOBIL

V vozilu z bencinskim motorjem so pogonski agregat, sklopka, menjalnik in diferencialno gonilo kompaktna enota. Zato smo se odločili, da tudi elektromotor, novi menjalnik in diferencialno gonilo prav tako združimo v sklenjeno celoto, ki bo poganjala polgredi prednjih koles. Po mnogih poskusih, tako glede funkcionalnosti kakor tudi glede trdnostnih značilnosti, se je za najprimernejšega izkazal dvostopenjski planetni menjalnik. Obširen proces optimiranja celotnega gonila, ki vključuje še diferencialno gonilo, je dal rezultat, kakršnega prikazuje slika 5.

2 DRAFT OF THE PLANETARY GEARBOX FOR THE ELECTRIC CAR

The power engine, clutch, gearbox and differential drive are very compact units in most modern cars equipped with the internal combustion engine. Therefore it is natural that the designers tend to unite together the electric motor, new gearbox and the differential drive in one single unit. Following from many functionality and gearbox strength analyses the adequate solution seems to be the two step gearbox based planetary drive. Comprehensive calculations, comprising also the differential drive, have resulted in the power train as it is presented in the figure 5.



Sl. 5. *Shema prenosne verige električnega avtomobila*
Fig. 5. *The power train draft of the electric car*

Posamezno prestavo gonila vključimo tako, da zavremo sončnik z notranjim ozobjem, z_3' ali z_3. Elektromotor poganja sončnik planetnega gonila z_1. Odgonski element gonila je nosilec planetnikov, v katerem so prosto vležajeni štirje dvojni planetniki. Med menjalnikom in diferencialnim gonilom je zobniška dvojica s poševnim ozobjem, ki zmanjša vrtilno frekvenco za faktor 1,6. Če zavremo zavoro 1, se zaustavi sončnik z_3', potem se moč v gonilu prenaša prek zobnikov z_1, z_2, z_2' in z_3'. Prestava gonila tipa 2AI se v tem primeru izračuna po enačbi:

By braking one of the hollow gears, z_3 or z_3', each individual step of the gearbox is activated. The electric motor runs the input shaft and the central gear marked with z_1. The output shaft of the gearbox is connected with the planetary gear carrier. The planetary gear carrier is fitted with four double planetary gears. A spiral bevel gear pair connects the gearbox and the differential drive, which reduces the rotation speed by the ratio 1.6. Brake 1 stops the hollow gear z_3' and consequently the power is transformed through gears z_1, z_2, z_2' and z_3'. In this case a planetary drive of type 2AI is formed, where the speed ratio can be calculated by the following equation:

$$i_I = i_{1/S} = 1 + \frac{z_2}{z_1} \cdot \frac{z_3'}{z_2'} \quad (1).$$

Ko v enačbo (1) vstavimo podatke o številu zob zobnikov, dobimo prvo prestavo menjalnika:

From eq. (1) the first ratio of the treated gearbox follows as:

$$i_I = i_{1/S} = 1 + \frac{52}{26} \cdot \frac{96}{12} = 5{,}00$$

Ker želimo v planetno gonilo vgraditi štiri garniture dvojnih planetnih zobnikov, $p = 4$, moramo preveriti, ali je to pri izbranih številih zob tudi izvedljivo. Če bo količnik k_I iz enačbe (2) celo število, je 4 planetnike mogoče vgraditi na vsakih 90°. Izpolnjena mora biti torej enačba:

Because of intention to fit four double planetary gears ($p = 4$) it is necessary to control the possibility of their assembly. If the result of the equation (2) is an integer number, then it is possible to fit four double planetary gears equidistant on the pitch of the planetary gear carrier at every 90°. The following equation needs to be fulfilled:

$$k_I = \frac{z_1 \cdot z_2' + z_2 \cdot z_3'}{p} \quad (2).$$

V enačbo (2) vstavimo znane podatke in dobimo:

For the treated gearbox it gives:

$$k_I = \frac{52 \cdot 12 + 26 \cdot 96}{4} = 780 \in Z$$

S to kontrolo smo potrdili, da lahko vgradimo 4 planetnike. Po enakem postopku izračunamo še prestavo planetnega gonila pri aktivirani zavori 2 (zavora 1 je sproščena).

This confirms the possibility of fitting four planetary gears. Following the same procedure the gearbox ratio can be calculated for the case of active brake 2 (brake 1 is relaxed).

Če zavremo zavoro 2, se zaustavi sončnik z_3, potem se moč v gonilu prenaša prek zobnikov z_1, z_2 in z_3. Prestava gonila tipa 1AI je neodvisna od števila zob planetnikov in se v tem primeru izračuna po enačbi:

Brake 2 stops the hollow gear z_3 and consequently the power is transformed through gears z_1, z_2 and z_3. In this case a planetary drive type 1AI is formed, where the ratio can be calculated by the following equation:

$$i_{II} = i_{1/S} = 1 + \frac{z_3}{z_1} \quad (3).$$

Ko v enačbo (3) vstavimo podatke o številu zob zobnikov, dobimo drugo prestavo menjalnika:

For the treated gearbox the second ratio of the gearbox is equal:

$$i_{II} = i_{1/S} = 1 + \frac{108}{52} = 3{,}08$$

V gonilo so vgrajeni štirje dvojni planetni zobniki, zato je treba preveriti, ali je število zob obeh sončnikov primerno izbrano:

For this case the possibility of fitting four double planetary gears can be checked with:

$$k_{II} = \frac{z_1 + z_3}{p} \quad (4).$$

Tako dobimo:

It gives:

$$k_{II} = \frac{52 + 108}{4} = 40 \in Z$$

S tem je potrjeno, da je število zob zobnikov planetnega menjalnika izbrano tako, da je planetne zobnike mogoče zmontirati. Velikosti posameznih prestav menjalnika so prilagojene voznim značilnostim vozila. Kriterij sta bila največja strmina (23 %) in največja hitrost vozila (100 km/h). Pravilna izbira prestavnih področij je potrjena z voznim diagramom, (sl. 3).

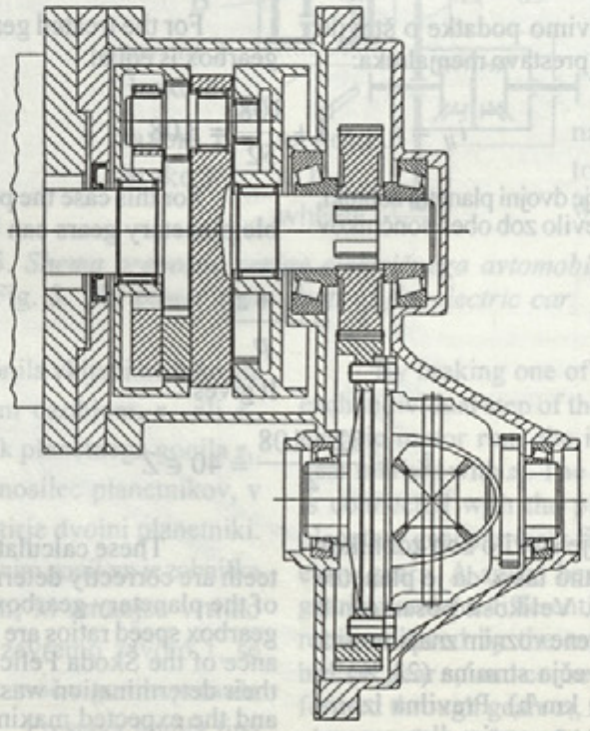
These calculations tell that the numbers of teeth are correctly determined and that the assembly of the planetary gearbox is possible. The individual gearbox speed ratios are fitted to the driving performance of the Škoda Felicia vehicle. The criterion for their determination was the top velocity (100 km/h) and the expected maximum slope (23 %). The diagrams in figure 3 confirm the correctness of the criterion.

Trdnostni izračun glavnih elementov planetnega gonila je bil odločilen kriterij za izbiro modula zobnikov, širine zobnikov, primernih ležajev, maziva itn. Zobniki menjalnika so dimenzionirani na trajno trdnost, (preglednica 1). Označbe zobnikov, ki so omenjeni v preglednici 1, so pojasnjene na sliki 5. Za povezavo planetnega menjalnika in diferencialnega gonila je dodana zobniška dvojica, ki ima modul $m = 2{,}5$ mm, prestavo $i = 69/43 = 1{,}605$. V preglednici 2 so rezultati preračunov ležajev. Doba trajanja ležajev povsem zadošča za vozilski menjalnik, za katerega se pričakuje, da zdrži vsaj 300 000 km z 10-odstotno verjetnostjo poškodbe. Na sliki 6 je prikazana zasnova planetnega menjalnika, ki rabi za izdelavo sestavne risbe in za razdelavo konstrukcijske in tehnološke dokumentacije. Izgube v ozobju planetnega menjalnika so kvečjemu 330 W (2%).

The decisive criterion for gear module, face width, bearings and lubricants determination is the strength calculation. The gears are designed to endure a theoretically unlimited service life, see Table 1. The markings used in the Table 1 are described in the figure 5. A special gear pair connects the gearbox with the differential drive (module: $m = 2.5$ mm, ratio: $i = 69/43 = 1.605$). The results of the bearing calculations are collected in Table 2. The service life of the bearings is appropriate for the vehicle gearbox, where the vehicle-covered distance of around 300 000 km with 10% failure probability is expected. In figure 6 the draft of the gearbox is illustrated, which is the basis for the design and manufacturing documentation. Power loss due to gear engagement is estimated to be 330 W (2%).

Preglednica 1. *Trdnostni izračun zobnikov planetnega menjalnika*
Table 1. *Strength calculations of the planetary gearing*

gonilo drive	zobnik gear	σ_H MPa	σ_{HP} MPa	S_H	σ_F MPa	σ_{FP} MPa	S_F
1AI	z_1	736	1732	2,35	195	686	3,44
	z_2	736	1732	2,35	195	480	2,45
	z_3	381	708	1,86	217	414	2,66
2AI	z_1	736	1732	2,35	195	686	3,44
	z_2'	1449	1732	1,19	408	480	1,65
	z_3'	512	1298	2,52	400	614	2,14
menjalnik/ diferencial	z_4	1070	1499	1,40	643	686	1,50
gearbox / differential	z_5	1070	1499	1,40	582	686	1,64



Sl. 6. *Zasnova planetnega menjalnika*
Fig. 6. *Draft of the planetary gearbox*

Preglednica 2. *Trdnostni izračun nekaterih ležajev planetnega menjalnika*
Table 2. *Strength calculations of some bearings of the planetary gearbox*

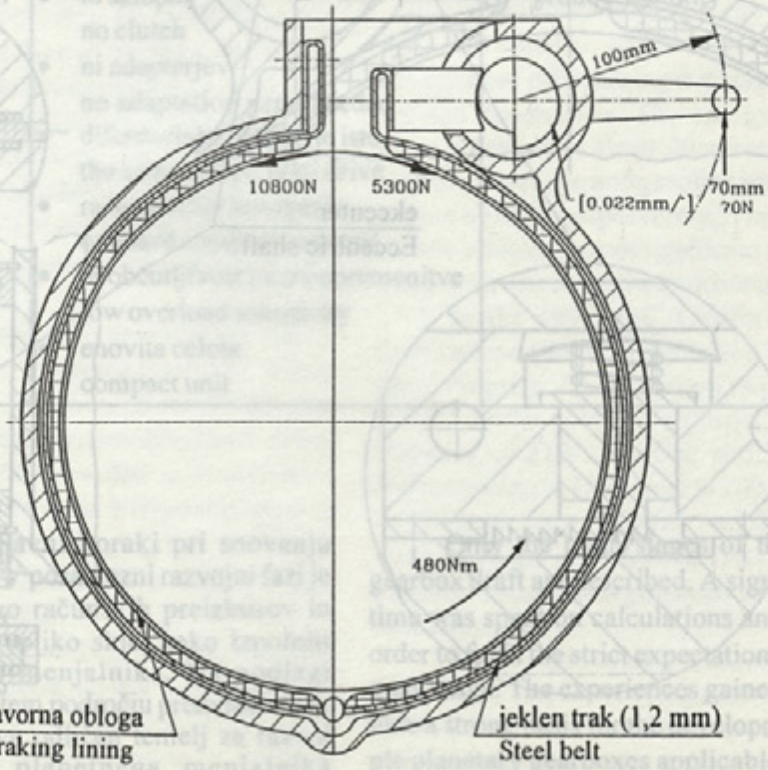
ležaj bearing	vrsta ležaja bearing type	C kN	F_r kN	F_a kN	f_n	f_L	L_h h
A	kroglični ležaj ball roller bearing	14	834	/	0,2832	2,08	57200
B	iglični ležaj needle roller bearing	14	709	/	0,2832	2,08	93200
C	stožčasti ležaj tapered roller bearing	72	7725	2889	0,4881	2,08	9300
D	stožčasti ležaj tapered roller bearing	57	7725	2889	0,4881	2,08	4600

Pri snovanju menjalnika je bila prav posebna pozornost posvečena zavorama 1 in 2, ki morata izmenoma zaustaviti sončnika z_3' in z_3, tako da ju v aktiviranem stanju zavore povežeta z okrovom menjalnika. Prvi preračuni so bili opravljeni za lamelne zavore, ki pa so se zaradi načina aktiviranja in relativno velike vzdolžne izmere pokazale za neprimerne. Tračne zavore imajo boljše karakteritike. Oporni moment zavore 1 je 480 Nm, oporni moment zavore 2 pa 250 Nm.

Smer delovanja opornega momenta na tračni zavori se med vožnjo avtomobila spreminja, odvisno pač od režima. Zaradi enostavnosti konstrukcije se odločimo, da sta obe zavori enakih dimenzij. Slika 7 prikazuje zasnovo tračne zavore, na kateri so vpisani karakteristični podatki. Zavorni trak je debel 0,5 mm, širok pa 40 mm. Potrebni zavorni moment se ustvari s trenjem med površinsko kaljenim kolutom (sončnikom) in brezazbestno zavorno oblogo, ki je prilepljena na jekleni trak.

Special care is devoted to brakes 1 and 2, which alternatively need to stop hollow gears z_3' and z_3. The active brake connects the hollow gear with the gearbox housing. First calculations were conducted for the disc brakes, which are not appropriate for this purpose due to large axial dimensions. Belt brakes have much better characteristics. Capacity (torque) of the brakes 1 and 2 is 480 Nm and 250 Nm, respectively.

During the driving, the directions of the braking torque changes. The direction depends on the driving regime. For the sake of simplicity it was decided that both brakes would have the same dimensions. Fig. 7 shows the draft of the belt brake. The braking belt is 0.5 mm thick and 40 mm wide. The necessary braking torque is generated by friction between the surface hardened disk (hollow gear) and the braking lining that is glued on the steel belt.



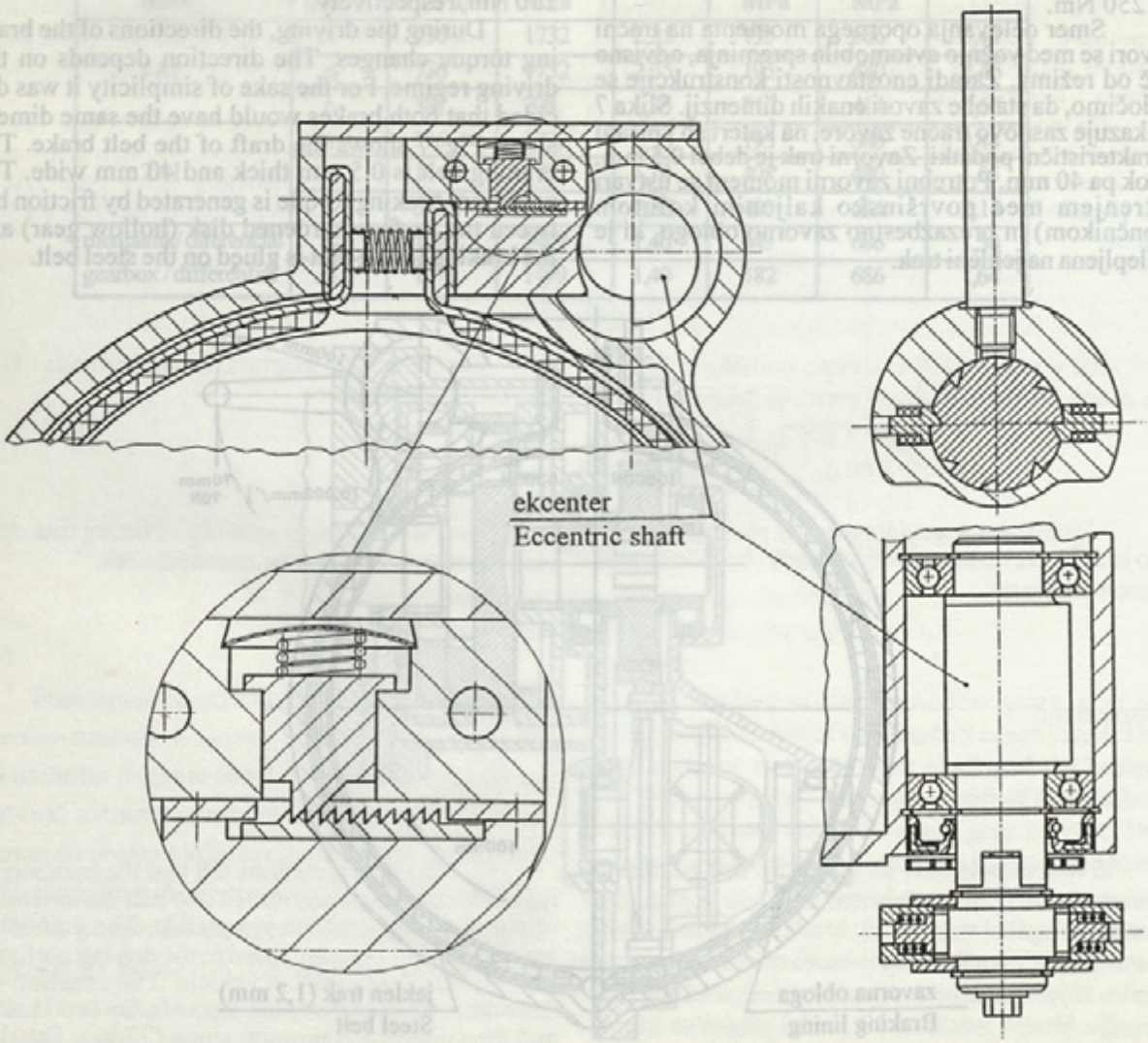
Sl. 7. *Tračna zavora v planetnem menjalniku*
Fig. 7. *Belt brake in the planetary gearbox*

Koeficient trenja brezazbestne zavorne obloge, ki drsi po kaljeni jekleni površini, vse skupaj pa je v oljni kopeli, znaša okoli 0,12 [1]. Dopustni ploščinski tlak za tovrstne sklopke ali zavore znaša okoli 1000 kPa. Če zavoro aktiviramo pri momentu motorja ali vztrajnosti pod 40 Nm, potem največja površinska obremenitev zavorne obloge ne presega dopustnega površinskega tlaka. Ko pa je zavora aktivirana; to pomeni, da površini medsebojno že relativno mirujeta, moramo tlak dvigniti na okoli 3000 kPa, da lahko prenašamo oporni moment planetnega gonila v celoti, 480 Nm.

Tračno zavoro aktiviramo z ekscentrom in ročico, ki imata skupno prestavo okoli 70, kar pomeni, da namesto sile 5300 N, ki deluje na trak, potrebujemo le še silo 75 N, ki deluje na 100 mm dolgi ročici. Sila za aktiviranje zavore povzročimo z elektromagnetom, ki mora premakniti ročico za ločni gib okoli 70 mm. Nekateri detajli mehanizma za aktiviranje zavore in mehanizma za zagotavljanje zračnosti so prikazani na sliki 8. Predpostavljena zračnost med zavornim kolutom 180 mm in brezazbestno zavorno oblogo na dvodelnem jeklenem traku znaša od 0,2 do največ 0,4 mm.

The friction coefficient of the braking lining, which is lubricated with oil, is about 0.12 [1]. The allowable surface pressure for those kinds of clutch or brakes is about 1000 kPa. It is supposed that the brake is activated by only one third of the maximum engine torque (40 Nm). At this torque the allowable lining pressure is reached. In order to reach the full capacity of the brake (480 Nm) the pressure is raised to 3000 kPa. This value is still acceptable because the surfaces have stopped relatively during the rising pressure.

The belt brake is activated by the eccentric pin and a beam, which produce a force ratio of around 75. This means that to generate the force of 5300 N on the braking belt only 70 N is needed at the distance of 100 mm along the beam. The active force is produced by an electric coil, which is needed to move the beam by around 70 mm. Some details of brake mechanism and a system for necessary gap assurance are shown in figure 8. Between the brake disc and lining glued on the two parts of the belt there is a gap ranging between 0.2 up to 0.4 mm.



Sl. 8. *Zasnova sistema za aktiviranje in kompenzacijo obrabe zavorne obloge*
Fig. 8. *Draft of the brake activation mechanism and the mechanism for gap providing*

3 SKLEP

Planetni menjalnik, ki je opisan v tem članku, v celoti zadovoljuje pričakovanja tako glede voznih lastnosti kakor tudi glede kompaktnosti konstrukcije. Primerjava pogonskega prenosa električnega avtomobila, ki je prevzeta iz vozila z bencinskim motorjem, in prenosa s planetnim menjalnikom, je prikazana v preglednici 3.

3 CONCLUSION

The described planetary gearbox completely satisfies the driving performance expectations and is the most suitable from the engineering aspect. Table 3 shows the comparison between the conventional transmission line (used in the vehicle equipped with the internal combustion engine) and the transmission line with the new planetary gearbox.

Preglednica 3. *Primernost uporabe planetnega menjalnika v električnem avtomobilu*
Table 3. *Advantages of using the planetary gearbox in the electric car*

	prednosti advantages	pomanjkljivosti disadvantages
običajen prenos (5-stopni menjalnik) conventional transmission line (5-speed gearbox)	• preizkušen in lahko dosegljiv well tested and easy reachable • nizka cena low price	• nepotrebne prestave unnecessary speeds • nepotrebna sklopka unnecessary clutch • zahtevno pretikanje uncomfortable speed change • prekratka doba trajanja short service life
prenos z 2-stopnim planetnim menjalnikom transmission line with 2-speed planetary gearbox	• nepotrebna prestavna ročica gearbox control stick unnecessary • samo dve prestavi two speeds only • ni sklopke no clutch • ni adapterjev no adaptation parts needed • diferencialno gonilo je isto the same differential drive • računalniško krmiljenje onboard computer control • neobčutljivost na preobremenitve low overload sensitivity • enovita celota compact unit	• novi razvoj new development • višja cena zaradi manjše serije higher price because of small product quantity

Opisani so le glavni koraki pri snovanju planetnega menjalnika. V posamezni razvojni fazi je bilo opravljenih mnogo računskih preizkusov in narisanih je bilo zelo veliko skic, kako izpolniti pričakovane funkcije menjalnika. Na podlagi dolgoletnih izkušenj na tem področju predvidevamo, da je prikazana zasnova odličen temelj za razvoj popolnoma novega planetnega menjalnika električnega avtomobila. Ta zamisel je primerna tudi za komunalna vozila.

Only the main stages of the new planetary gearbox draft are described. A significant amount of time was spent on calculations and draft making in order to fulfil the strict expectations at each development stage. The experiences gained in this field provide a strong basis for the development of other simple planetary gearboxes applicable for similar electric cars. This concept is especially well suited for community service vehicles.

5 LITERATURA
5 REFERENCES

[1] Shigley, J. E.: Mechanical engineering design. The University of Michigan, 1990.

Naslov avtorjev: doc. dr. Stanislav Pehan, dipl. inž.
prof. dr. Jože Flašker, dipl. inž.
Izidor Rupnik, dipl. inž.
Fakulteta za strojništvo
Univerze v Mariboru
Smetanova 17
2000 Maribor

Authors' Address: Doc. Dr. Stanislav Pehan, Dipl. Ing.
Prof. Dr. Jože Flašker, Dipl. Ing.
Izidor Rupnik, Dipl. Ing.
Faculty of Mechanical Engineering
University of Maribor
Smetanova 17
2000 Maribor, Slovenia

Prejeto:
Received: 17.4.1997

Sprejeto:
Acceptted: 29.8.1997