

ANALYSIS OF APPLICABILITY OF A MECHANICAL GEAR DRIVE COMPOSED BY FACE GEARS IN A SMALL FARM TRACTOR

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Abstract. *Small size tractors are extensively used in family farming properties in a ample variety of activities. Such tractors usually present manually shifted gear transmission systems, with or without intermediate reducers. The designers of these transmission systems have to determine a range of work velocities which is adequate to the major part of applications. However, some specific tasks or cultures demand very low velocities, while others require an overdrive, which the transmission system is not able to provide. The mechanical drive system based on face gears proposed by Bianchi and Batista (2008) presents features that made it adequate to execute small velocity changes during the tractor operation, in an automatic form (without operator intervention). This paper presents a discussion of the applicability of this new mechanical gear drive concept as an auxiliary device to the original transmission of a small farm tractor. The goals are the amplification of the range of work velocities and the automatic and instantaneous variation in torque and velocity during operation. The requirements that this device should attend to be assembled on the tractor are explained, mainly with respect to the maximum system dimensions and the torque-velocity desired variation. The analytical dimensioning process of the main mechanical components is described, for which two algorithms were implemented: i) teeth proportion determination, based on AGMA standards and ii) determination of motion strokes (gears spacing and pinion nutation). An assembling drawing with only the main components is presented, with main dimensions and material and manufacturing process definitions of the principal components. Also the aspects related to costs of implementation are discussed, as well as the real benefits reached with the use of this device in the tractor operation.*

Keywords: *mechanical transmission, tractor drive, face gears.*

1. INTRODUCTION

Usually known as mechanical components suitable only for small load applications, face gear drives have been subject of recent and ample research and development (Litvin et. al., 1994, 2000, 2002 and 2005; Mao, 2007; Guingand et. al., 2005; Zanzi and Pedrero, 2005; Barone et. al., 2004 and Zhang and Wu, 1997). For example, Dudley (1994), in the last edition of his classical book, reported that “equipment to grind face gears is not available”. However, in the same year, Litvin et. al. (1994) present a paper showing the application of hardened and precision ground face gears in an helicopter transmission of more than 2000 kW. Figure 1 shows this mechanical gear system. A new manufacturing process for cutting and grinding face gears with worm tools enabled to obtain components for very high torque and velocity applications (Litvin et. al., 2000).

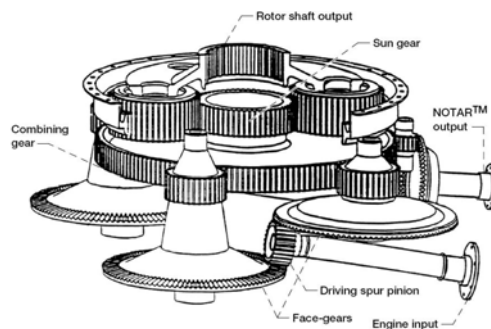


Figure 1. Helicopter transmission with face gears (Litvin et. al. 1994).

Face gears can be applied in crossed or intersected axes drives, or even in transmissions with an offset in the axes. Spur or helical pinions can be used and ratios of velocities in a range of 3.5 or more can be achieved in only one step. The advantages related to face gears are: *i)* good distribution of load between teeth, *ii)* reduced level of noise, *iii)* no sensitivity to pinion axial displacement, in case of a manufacturing or assembling error, *iv)* low thrust loads on transmission parts, *v)* lower manufacturing costs in relation to bevel gears. On the other hand, ordinary face gears are strongly sensible to angular misalignment between pinion and gear axes. In other words, a relatively small angular error moves the contact region to the edge of the teeth, increasing contact pressure level and causing decrease in lifetime.

Another drawback is that the manufacturing process of generation by a shaper tool does not permit work over hardened materials. However, a set of strategies, related to design and manufacturing, nowadays permits to overcome these disadvantages. Some of the strategies are the application of micro modifications in teeth geometry (crowning is an example) and the use of a shaper tool with 2 or 3 teeth more than the real pinion. The final result is the concentration of contact pressure in the center of face and pinion teeth, maintaining the stress level in admissible values.

Despite the advantages in costs and performance of face gear drives, these transmission systems ordinarily have fixed ratio of velocities. As an example, Litvin et. al. (2005) presented two different systems applying face gears and helical pinions. Both drives have aligned input and output axes.

Because of the impossibility of gear ratio shifting, face gears have limited applicability in vehicle transmissions. In other words, for application of a mechanical drive based on face gears in the automotive or agricultural fields, it is necessary to implement a system that presents different ratio of velocities and, additionally, a simple way of shifting the working gears.

The recent studies made over a face gear transmission with 3 ratios of velocities, implemented by Bianchi and Batista (2008), show that it is possible to achieve the desired design and operation constraints of the automotive and agricultural fields. This paper briefly presents the concept of the proposed face gear drive and, sequentially, presents the analysis method adopted to design a real gear drive for implementation in a small tractor.

2. DESCRIPTION OF THE FACIAL GEAR DRIVE WITH THREE RATIOS OF VELOCITIES

The concept idea of a stepped transmission for agricultural purposes, developed by Bianchi and Batista (2008), is shown in Fig. 2. The system here presented has three relations of gears and is composed by two identical and opposed input and output gears, with a set of intermediate pinions. Both gear and pinion are divided in toothed bands and this number of bands corresponds to the number of ratios of velocities.

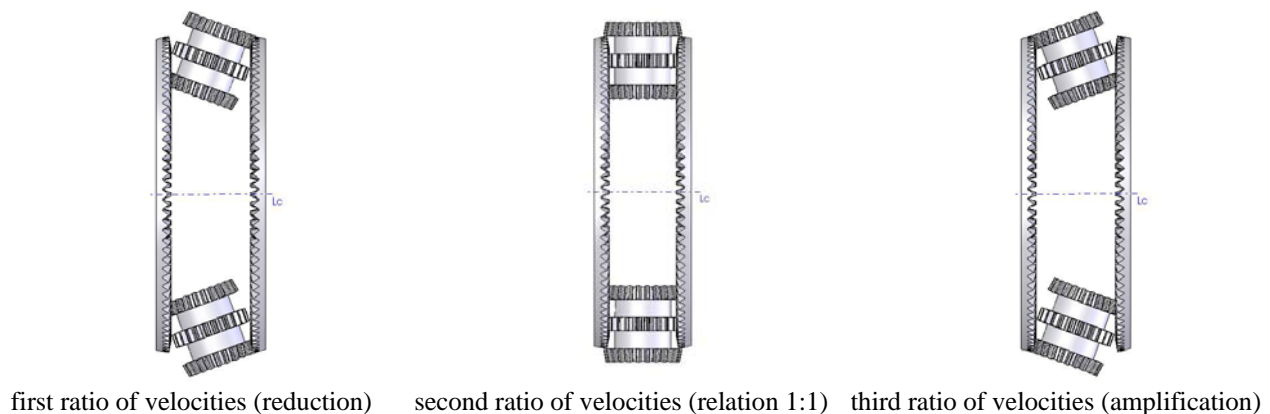


Figure 2 – Principal parts that compose the face gear drive with three ratios of velocities.

Four movements are necessary for the system operation and gear shifting: *i*) angular rotation of input face gear; *ii*) nutation of intermediate pinions; *iii*) relative axial displacement between input and output face gears and *iv*) axial displacement of the pinions (half of the relative gears displacement).

A relative axial displacement of the gears means that at least one of these gears should slide over a splined shaft; while an axial displacement of the pinions means that these pinions should be fixed over a carrier part.

A prototype was constructed and tested by the authors to prove the functionality of the proposed system. However, a transmission for field application have not been designed yet, being the subject of the present article.

Nevertheless, all the preliminary studies and tests realized show that, beyond the characteristics related to the face gears, the proposed system presents the following features:

- high load capacity with small teeth sizes (great gear diameter permits the transmission of high torque values)
- small variation in the ratio of velocities
- relative simple mechanisms for gear shifting
- great moment of inertia

Because of the importance of each one of these aspects, a more detailed explanation will be made in next sections.

2.1. Load capacity of the proposed face gear transmission

Dudley (1994) relates that: “The formulas for determining the dimensions of a pinion to run with a face gear are not different from those for the dimensions of a pinion to run with a mating gear on parallel axes. The pressure angles and pitches used are similar to spur-gear (or helical gear) practice”.

Although the intermediate pinions do not contribute to the final value of ratio of velocities, the power flow passes through them. Because of the small primitive diameter, the relative thin teeth profile and the extremely high angular velocity, the pinions represent the weaker components in the transmission.

An important aspect is that the face gear tooth has a considerable variation in shape along the radial direction. In a long teeth, at the outer gear diameter, there is the encounter between opposed tooth faces, forming a pointed tip. On the other hand, the inner diameter tends to present the undercut problem, which means that the contact meshing lines concur to a small region and the real work tooth face narrows. Both problems represent a limiting radius that delimits the gear tooth length. The recommendation is the final tooth length to be somewhat shorter than the difference between outer and inner limiting radii.

In face of these limitations, the tooth width of each pinion band is determined by the corresponding face gear band radial length. So the dimensioning process for each pinion band has the tooth width as one of the input data. So, for a given force (or torque) value to be transmitted, the tooth thickness is inversely proportional to its width.

Logically, in the design process, the overall permissible dimensions for the system also influences on the tooth thickness. So, if a great gear diameter is allowed, the teeth could be smaller. Some important benefits can be achieved by using small teeth (and, consequently, small pinion diameter): *i*) the possibility of implementation of a greater number of working bands and, consequently, ratios of velocities, *ii*) small relative axial displacement between input and output face gears for velocity shifting, *iii*) less total length in the pinion (more compact system) and *iv*) better load distribution in the pinions.

This last aspect is related to the fact that mechanical gear drives composed of a set of gears working together and sharing the load, because of manufacturing and assembling inaccuracy, shall present components with some flexibility. As the load capacity of the proposed system is dependent on the number of intermediate pinions, the use of small flexible teeth represents a more efficient load sharing. On the other hand, the relative high teeth deflection have two drawbacks: *i*) necessity of adoption of strength and hardened materials in the pinions (small teeth supporting high loads) and *ii*) elevation of dynamical effects in high operation velocities. This last aspect can be minimized by a small modification in teeth involute profile, more specifically, a tip relief (Mao, 2007).

Finally, geometrical and kinematics aspects influence on gears and pinions strength and life. A very important parameter is the pressure angle of the pinion. Although a spur pinion has a constant pressure angle as one of its kinematics and manufacturing parameters, on the contrary, if this pinion works with a face gear, the pressure angle of the couple varies in the gear radial direction. As a general rule, a great value of pressure angle in the pinion results in a more strength transmission because of: *i*) the pinion presents more robust teeth and *ii*) the face gear tooth length, given by the limiting inner and outer radii, is greater. Figure 3 shows three transmissions with the same gear and pinion teeth number and size. Drawings (a), (b) and (c) present a piece of a face gear in contact with pinions of, respectively, 14.5°, 20° e 25° pressure angle; while drawings (d), (e) and (f) show a detail of the shape and length of the respective face gear tooth. Observing the differences between the gears shown in Fig.3, it can be stated that, considering only kinematics and strength aspects, the choice of a large value of pressure angle is favorable.

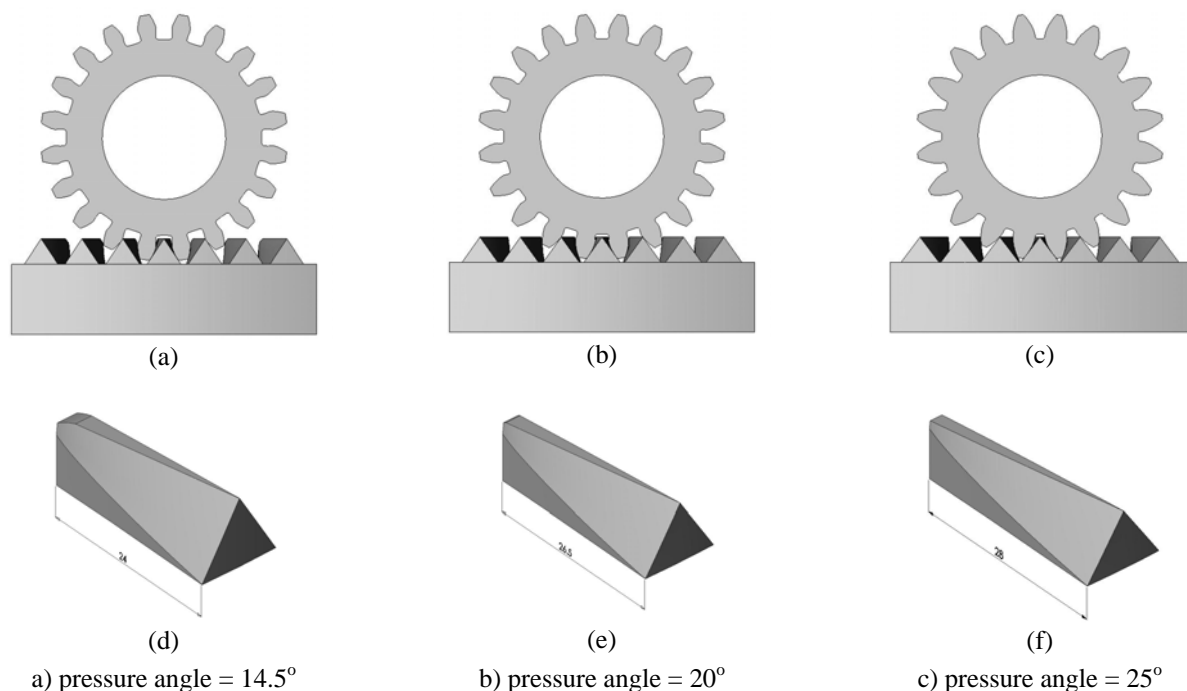


Figure 3 – Influence of the pinion angle of pressure on the teeth shape and strength.

The determination of the final gears proportions, material and hardening process and manufacturing quality degree involves the manipulation of great quantity of information. An algorithm based on AGMA/ISO standards was implemented to help to determine the final teeth proportions, permitting to compare different alternatives of material, hardening and manufacturing processes. This algorithm takes into consideration velocity, loads and geometry factors that influence on the final system performance.

Although an algorithm gives flexibility to the dimensioning process, the final proportions of a mechanical drive system can vary considerably by changing input data. For example, if there is enough space, low working speeds and tolerance to inertial load, one could prefer to adopt greater gears, composed of materials of low strength with simple or none hardening process.

2.2. Ratios of velocities

The prototype developed has gears working bands of 80, 88 and 96 number of teeth, which result in relations of velocity of -1.2, -1 and -0.833. The capacity of obtaining greater ratios of velocities is related to the possibility of adoption of small teeth. Anyway, the alteration in the velocity would not be expressively greater.

To compare this values with current relations of velocities of the mobility field, hydraulic torque converters usually have relations between 2 and 3. The alteration on the velocity from one gear to another in a manually shifted transmission is in the order of 40 to 60%.

So, the proposed system is not adequate for replacing existing transmissions, but to help these gear drives to work with more flexibility. In an specific gear relation of the transmission (1st gear, for example), the proposed system could offer a small variation to obtain an increment in torque or in the final velocity.

This possibility is only profitable if the proposed system could work in an automatic form. If the gear relation shift should be done by the user, maybe a better solution would be adopting a transmission with more gear sets.

2.3. Mechanism for gear shifting

The tests implemented over the prototype showed that, if the axial and nutation movements of the pinions are very fast, there is no necessity of synchronizing components. However, fast movements require damping devices at the end of the course, to avoid impact loads and noise. Anyway, the mechanism for gear shifting adopted in the prototype is extremely simple and could be extended for real applications.

Small electrical or pneumatic linear actuators, associated with levering arms in the case of nutation movement, could offer the necessary velocity. In the prototype, two pneumatic actuators were adopted.

2.4. Inertial load

An important characteristic of the proposed system is the high inertia of input and output gears. Frequently, mechanical gear drives are used not only to decrease the working load to the motor axis, but also to reduce the inertial load during transient. In this aspect, the contribution of the reducer is very important, because the inertia of the moving components, when referenced to the motor axis, is decreased in a square relation with the ratio of velocities. So, mechanical systems that should present a high dynamic performance, as servo controlled mechanisms for example, frequently are equipped with reducers for decreasing the inertial load referenced to the motor axis.

However, in some mechanical systems, the inertia of the components is well accepted, because of its capacity of reduce noise and impact load or motor amplitudes. Classical examples are the flywheels adopted in press machines and in crankshaft of combustion engines.

This aspect of the proposed gear drive limits its application field to the family of machines and equipments where the inertia is not an important item, or in the systems where high inertia values in the components are desirably.

3. STUDY OF A MANUALLY OPERATED GEAR DRIVE USED IN A SMALL TRACTOR

Small tractors for family farms frequently are equipped with manual or synchro shift transmissions with 4 or 6 ratio of velocities divided in high and low stages of operation. Figure 4 shows a representative diagram of a mechanical drive used in a small tractor with 6 ratios of velocities. Additional and usual sub-systems were omitted for simplicity, as examples, the output gears and axis for front wheels and PTO (Power Take Off) actuation.

The analysis process for determination of gears sizes and materials considers two important aspects: *i*) the power provided by the engine and *ii*) the traction capacity of the rear tires. The transmission system shown in Fig. 4 has an alteration in the ratio of velocities, from 1st to 2nd pair of working gears, of more than 57 %. On the other side of work range, from the 5th to 6th ratios of velocities, there is a change of more than 68 %. So, inside the working band corresponding to one of the gears ratios, the operation point in the engine torque and power curves is changed in face of a load fluctuation. In other words, the working band corresponding to each step of the drive is too large, making difficult to operate around the optimal fuel consumption point. Considering only this aspect, it would be desirable to

have the total working range divided in more steps (8, 10 or more ratio of velocities). However, it would be negative in terms of operational and costs aspects.

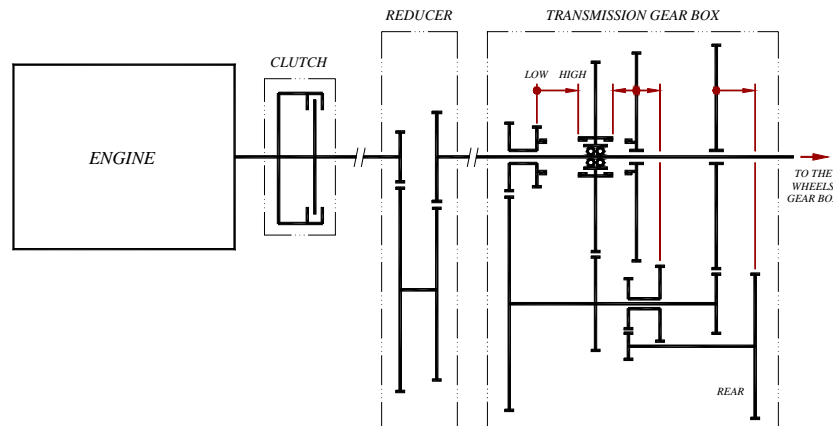


Figure 4 – Mechanical transmission with 6 ratios of velocities of a tractor.

The installation of an auxiliary transmission based on face gears, with three relation of velocities values, could extend the working range of the tractor. The possibility of automatic operation of this auxiliary device, would not impact in the user driving comfort. So, for an specific ratio of velocities determined by the tractor operator, a control system would detect the necessity of higher torque or velocity and send de signals necessary for the actuators responsibly for the gear shifting. Describe the control algorithm is not the objective of the present work, but mainly to design the mechanical components of the proposed transmission.

4. ANALYSIS OF THE PINIONS AND GEARS

The project of the facial gear drive took into consideration some important aspects:

- a) the tractor structure and the power flow composed by engine, transmission, reducers and power-train had to be evaluated. The system shall be designed to be assembled in a part of the tractor that does not requires alteration on existing tractor components and fixtures.
- b) the gears radii should be as longer as it was possible, permitting to adopt small teeth thickness in all the working bands of the transmission. The choice of great gears radii implies in high moments of inertia. So, the system should be assembled in a part of the power flow (from engine to the wheels) in that this inertial load is tolerable, or even recommendable.
- c) pinion material and hardening process should be of high quality. The adoption of great radius in the input gear implies in high tangential velocities and high number of cycles of load in the pinions. Moreover, this procedure permits to adopt small teeth in the transmission.

After a simple evaluation of the tractor power flow, remained clear that the best place to this auxiliary transmission device to be installed is between the clutch and the reducer. In this part of the tractor there is enough space (there is a splined axis transmitting power from clutch to the reducer) and the inertial load added to the system is not relevant, but, on the contrary, desirably for softening the alternating movement of the motor. Also, the flexibility of the clutch could prevent the transmission system against torque peaks of the motor.

Table 1 shows the first input data for the dimensioning process, which are the maximum power and torque provided by the motor. In Tab.1 is shown the reducer width value, which permits to estimate the overall dimensions of the device, as well as the gears diameter.

Table 1. First input data for the dimensioning process

Power - maximum	22 kW – 3000 rpm
Torque - maximum	70 Nm – 2250 rpm
Reducer width	300 mm

Based on the reducer width, a gear diameter of 250 mm for the intermediate ratio of velocities was adopted as reference value for the analysis. These input data permitted to obtain the force to be transmitted by the face gear and pinion teeth in the intermediate working band.

The next step was the definition of the teeth proportions for the pinion in the intermediate working band. The procedure, based on AGMA standards, took into consideration the input data shown in Tab. 2.

Table 2. Input data for definition of pinion teeth modulus in the intermediate working band.

power	22 kW
angular velocity	input gear: 314.2 rad/s (3000 rpm) pinion: 1963.5 rad/s (18750 rpm)
angle of pressure	25°
material	ASTM 4340 – grade 1 – quenched and tempered (350 HB)
pinion number of teeth	20
number of pinions	5 (3 for analysis process)
lapping process	no
AGMA quality number	6
teeth crowning	no
load characteristic	intense shocks (agricultural application)
engine characteristic	intense shocks
reliability	0.995
estimated temperature	120°C
cycle load number	1E12

For these material and operational conditions, a width of 20 mm and a module of 2 mm for the intermediate pinion working band is sufficient and adequate. Defined de module and the gear radius, the number of teeth in this face gear is 125.

Some considerations should be made about the input data shown in Tab.2: *i)* the number of teeth of pinion was previously determined, trying to avoid undercut but with a small diameter because of the high operational velocity; *ii)* pinion velocity was obtained by a looping process, because it is a module function; *iii)* the power is divided by the number of intermediate pinions, but it was considered that, because of manufacturing errors, only three pinions should be responsible for the entire load; *iv)* the pinion width was compared, in the looping process of analysis, with the difference between limiting inner and outer radii of the gear.

The same procedure was adopted for external and internal bands. Because of manufacturing aspects, the pinion module and width was maintained for all the working bands.

An important geometrical feature is the angle between pinion and gear axis, which is function of the necessary axial and nutation movements in the pinions. Fig. 5 shows a pinion with 20 teeth and three working bands of width $F = 20$ mm. Fig. 5(a) shows pinion and face gear in mesh in the intermediate band; Fig. 5(b) shows the necessary translation displacement for gear shifting and Fig. 5(c) presents the pinion position after the nutation movement for external band operation. The pinion nutation and axial displacement values are: a) $\theta = 22.5^\circ$ and $\delta = 6.75$ mm.

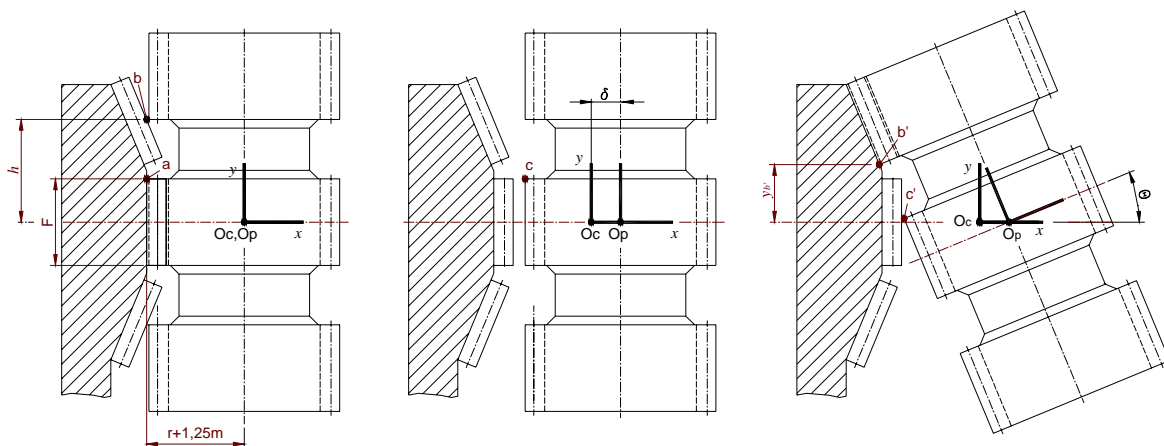


Figure 5 – Axial and nutation displacements.

Pinion axial and angular displacements are correlated, so for different nutation values, a different axial displacement is necessary. A simple algorithm was implemented to determine the minimum distance h from external and internal band to the pinion origin, as well as the axial displacement δ (see Fig 5(a)), for a given value of nutation angle. These geometrical parameters are determined so that the point \mathbf{b} , when rotated to position \mathbf{b}' , has a vertical coordinate greater than point \mathbf{a} . On the other hand, point \mathbf{c} , when rotated to \mathbf{c}' , should not collide with the tip of the teeth that compose the intermediate working band of the gear.

Fig. 6 shows the result for the pinion nutation and axial displacement values of $\theta = 15^\circ$ and $\delta = 6.5$ mm.

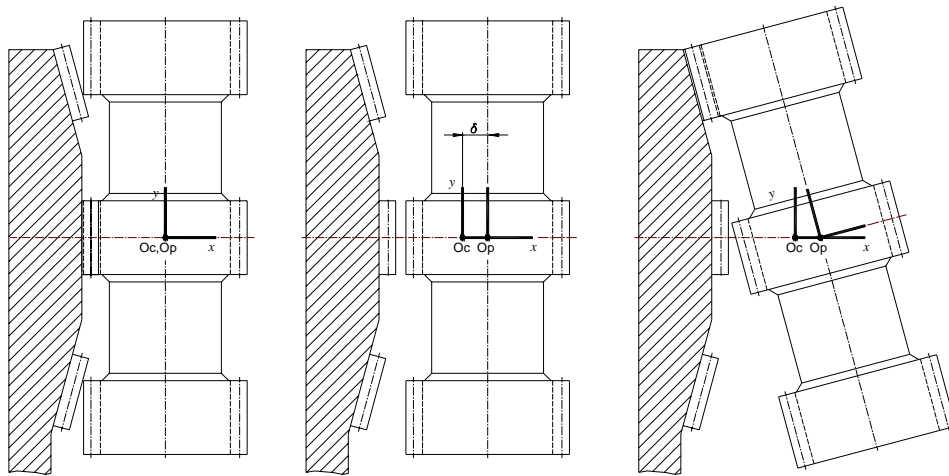


Figure 6 – Pinion and gear geometry for $\theta = 15^\circ$ and $\delta = 6.5$ mm.

The examples shown in Fig. 5 and Fig. 6 demonstrate that smaller values of nutation angle result in a longer pinion and in a large transmission system. However, a very large nutation angle results in a relative large axial displacement of pinion. In face of these aspects, the geometry defined by Fig. 5 was adopted for the proposed transmission system, which represents an angle between axes of, respectively, 67.5° and 112.5° for the outer and inner working bands.

After the definition of axes angles, it was possible to define the teeth proportions for these bands. The number of teeth obtained was 100 for inner and 140 for the external band. The final aspect of gears and pinions is shown in Fig. 7.



Figure 7 – Final aspect of pinions and gears working bands.

The final values of ratio of velocities are presented in Tab.3. The negative signal represents a velocity inversion. An example may be used to prove if these values are adequate: the original tractor transmission has the first ratio of velocities of $m_G = 9.62$. In association with the proposed system, this value could be changed to 13.468 (step 1), 9.62 (step 2) and 6.87 (step 3). This final ratio represents a intermediate value in relation to the second ratio of the current transmission system, which is of 5.55.

Table 3. Values obtained for ratios of velocities.

step	input : output teeth numbers	Ratio
1	100 : 140	-1.4
2	125 : 125	-1
3	140 : 100	-0.714

5. FINAL ASPECT OF THE PROPOSED SYSTEM

Figure 8 shows, in an exploded view, the internal components of the proposed system for application in a small tractor. Additional components are presented in this figure: the pinion carrier ring, the pinion fork holders and the nutation rods. A spur gear is assembled in the input face gear axis only for velocity inversion.

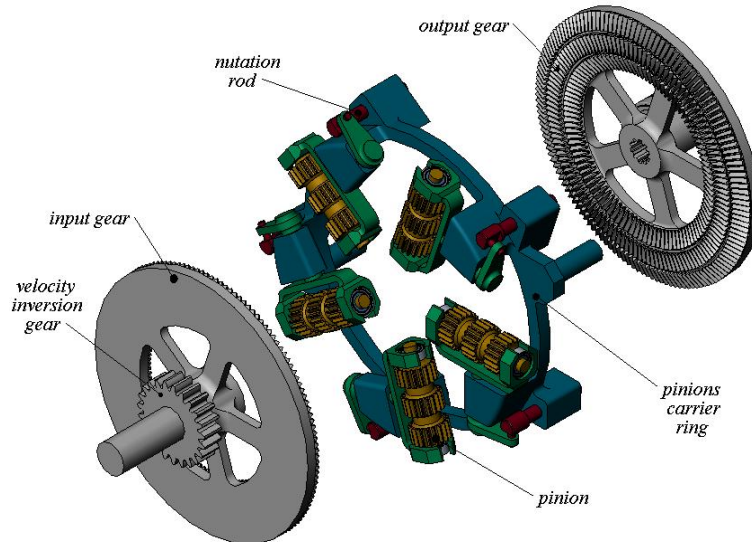


Figure 8 – Internal components of the proposed system for application in a small tractor.

The pinion carrier ring embraces the output gear and its axial displacement is made by solenoids inserted in the external box. Solenoids also will be responsible for the axial displacement of the output gear, as well as the axial displacement of the nutation rods, which perform the axial rotation of the pinions for gear shifting.

The system presents all the initially expected features: high load capacity in a relative compact frame, but the face gears have a high inertia. The robustness associated with face gears and the relative conservative dimensioning process adopted, may be considered indications of a reliable device. However, a product prototype shall be constructed, assembled in the tractor and hardly tested to confirm all the design, security and operational aspects.

6. CONCLUSION

The mechanical gear drive concept based on face gears presented by Bianchi and Batista (2008) was extended for a real application in a small tractor. Initial studies rapidly pointed to the use of this drive as an accessory device for an already existing mechanical transmission system of a tractor. Such implementation could extend the operational range of this tractor, permitting a more flexible and efficient use in different work conditions.

Design process took into consideration the power and torque engine curves, as well as the fixed ratios of velocities of the existing transmission system. Another important aspects were considered, as an example, the available space for accommodation of the device in the tractor, which impacts directly on the face gears diameter and in the overall load capacity of the system. For pinions and gears analysis, two auxiliary routines were implemented: the first permitted to visualize different alternatives of sets of axial and nutation displacements of the pinions and, after that, to choose the alternative that provided a more compact system. The second routine, based on AGMA standards, permitted to obtain the pinion teeth module and width. This width should be equal or less than the difference between outer and inner limiting radii of the face gear, which correspond respectively to the regions in that undercut and opposed face encounter happens in the face teeth geometry.

After this analysis process, the solid models of the components were obtained by the procedure already described in Bianchi and Batista (2008). The next step was designing the accessories components, necessary for the implementation of all the movements related to system operation. The final aspect of the system is of a compact mechanical gear drive, of a relatively high load capacity.

All the components that compose the proposed system have simple shapes and present ordinary manufacturing processes. The generation process of face gears with worm tools is comparatively simple and presents low costs. All the materials and hardening processes specified are of regular application in mechanical systems. The mechanisms for system operation are very simple and effective. Because of all these aspects, it is assumed that the system presents a compatible cost for agricultural, mechanical and mobility applications. However, a final evaluation over a real product prototype should be done to prove this market aspect.

5. ACKNOWLEDGEMENTS

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