Redesigning the cycloidal drive for innovative applications in machines for smart construction yards

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Abstract

Purpose – This study aims to carry out an investigation of design approaches that should be used for the design of unconventional, innovative transmission system for construction yards to privilege a smooth behaviour efficiency, and the use of innovative production techniques. Results are quite surprising, as with a proper method it is possible to demonstrate that a cycloidal drive with Wolfrom topology should be an interesting solution for the proposed application.

Design/methodology/approach – With a functional approach, also considering materials and specifications related to the investigated application, it is possible to demonstrate that possible optimal solutions should be quite different respect to the ones that can be suggested with a conventional approach. In particular for proposed applications constraints related to encumbrances, the choice of new material has led to the innovative unconventional choice of a Wolfrom cycloidal speed reducer.

Findings – Provided solution is innovative respect current state of the art for machine currently used in construction yards: in terms of adopted transmission layout; in terms of chosen materials, resulting in an innovative solution.

Research limitations/implications – Current research has strong implications on the adoption of polimeric materials for the construction of reliable transmission for harsh industrial environment as the proposed case study (concrete mixer for construction yard).

Originality/value – Proposed transmission system is absolutely original and innovative respect current state of art also considering proposed materials and consequently production methods. This is an example of transmission designed to be built with polymeric materials by optimizing chosen topology respect to chosen material.

Keywords Mechatronics, Cycloidal drive, Wolfrom drive, Harmonic drive, Construction yards, Cycloidal, Polymeric gears, Smart construction yards

Paper type Case study

1. Introduction

Machines used in construction yards are characterized by very demanding specifications in terms of transmission ratio, reduced production cost and reliability in harsh environmental conditions with limited maintenance.

Especially for concrete mixers (probably the most important application on which is focussed this work), an important objective is the need to optimize the technical solutions according to the cost specifications required by mass production. Moreover, these devices are moved through different construction yards, often with limited available spaces, e.g. in case of urban requalification and restoration programs. The use, the construction and the disposal of yard equipment's involve an environmental impact incompatible with the overall development trends, which demands to reduce the environmental footprint by improving system efficiency and extending its useful life. Furthermore, the widespread diffusion of energy storage systems and renewable sources involves the possibility of self-sustained yards where energy is autonomously generated, stored and managed. This is a quite feasible scenario mainly for three reasons. First, there is an increased diffusion of distributed power sources and local grids fed by renewable energy sources (Carrasco *et al.*, 2006). Second, there is an availability of large volumes of low-cost high performances static storage systems deriving from "second life application" of automotive accumulators (Tong *et al.*, 2013). Eventually, there

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is an increasing diffusion of fast wireless recharge technologies (Pugi et al., 2017a) also designed for hard environments (Allotta et al., 2017) which can drastically simplify power distribution in construction yards. Concerning mixing machines, they are present in any construction site and are deemed to be very important to obtain optimal and workable concrete (Maiti and Agarwal, 2009). Accordingly, several studies have been performed about the mixing methods and the related impact on the quality of the concrete, Maiti and Agarwal (2009). Further studies have been performed to estimate and analyse the forces within the mixing drum (Valigi and Gasperini, 2007, Valigi et al., 2015, 2016, 2019). Indeed, such forces considerably affect the mechanical stresses on both mixer structure and motor. The availability of this long standing and comprehensive set of studies nowadays allows the production of efficient mixers that ensure good mixing performances. However, this is not the sole performance that should be considered for the commercial success of this kind of products. Indeed, the technical solutions adopted for the embedded systems that allow the functioning of the drum (or any other alternative mixing device), need to be carefully selected and/or designed to reach the economical target required by the market.

In such a context, the proposed investigation activity is focussed on the identification of new efficient and compact reducers for the drum of the concrete mixer. Furthermore, shape and encumbrances of the proposed solution should be optimized for a simple and fast integration in different machines. In such a sense, a simple cylindrical shape with coaxial aligned input and output shafts it is quite ideal for an easy integration in a generic machine. This activity has been carried out in collaboration with an important firm whose main business sector refers to the design and production of machines for construction yards.

An additional aspect of the proposed investigation is represented by the possibility to exploit innovative low cost plastic/compound materials that could contribute in reducing the costs of the apparatus, as well as in reducing the need of expensive lubricants currently used for conventional gears (Biernacki, 2014; Berger, 2015).

For what concern lubrication the idea is to adopt a self-lubricating couple of polymers: wear rate is influenced by orientation of reinforcement fibres, which should be oriented axially respect to polymer gears. In addition, many nanomaterials exist which should be dispersed in the polymer matrix to further improve friction efficiency. Currently, authors are privileging these dry self-lubricating solutions since many machines are used often discontinuously in harsh conditions but for a relatively low number of hours (few thousands). Therefore, the idea is to design a cheap polymer-based transmission, which requires low maintenance, and that can operate in a wide range of environmental and loading conditions, respect to conventional lubricant.

In addition, the resistance to vibrations is quite important for the construction of compact/integrated units, like in wheel motors often proposed for electric vehicles (Pugi et al., 2017b).

According to the above-introduced objectives, the paper presents the performed research activity and the achieved results. More precisely, Section 2 reports the current state of the art in terms of available drive solutions for construction machines. Section 3 shows an excerpt of the elicited design requirements, while different drive solutions are evaluated in Section 4, against the extracted requirements. Section 5 presents a preliminary investigation about the adoption of different materials with a focus on polymeric ones. Eventually, Section 6 describes the design process of the new cycloidal reducer, while Section 7 summarizes the conclusions of this work.

2. Innovative high reduction gearboxes

In this section, currently available speed reducer technologies are briefly reviewed. Moreover, their criticalities are also summarized, which motivate the investigation of more cost-effective and compact solutions.

Different gearbox types could theoretically be exploited in the machines for construction yards. Among them, a single worm drive stage is a quite trivial example, but it is not applicable due to quite low efficiency and relatively high costs (Dudás, 2005). Therefore, the attention is focused here on planetary gearboxes (PG), cycloidal drive (CD) and harmonic drive (HD).

2.1 Planetary gearboxes

PG are often used in automation and the typical layout of a single stage is represented in Figure 1.

However, reduction rates achievable with a single stage are not enough for the considered application [9] (Kwon et al., 2014). A possible solution to further increase the reduction ratio is represented by the possibility of increasing the number of teeth of the planetary, obtaining the configuration, in terms of primitives, described in the scheme of Figure 2. Assuming the carrier as input shaft and the planetary as output, this case

Figure 1 Planetary gearbox, typical layout for a single stage reducer

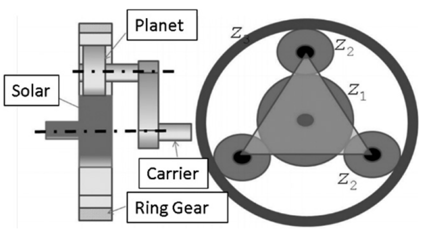
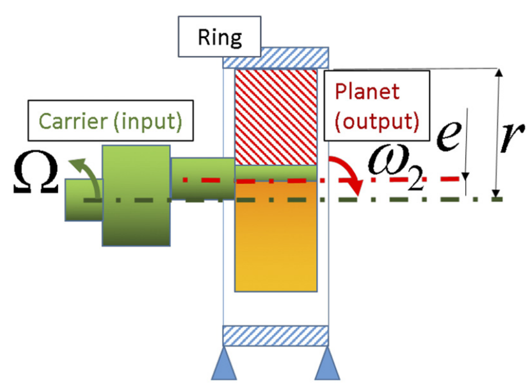


Figure 2 motion primitives of a cycloidal gearbox (studied as particular case of epicyclodal gearing)



Note: Studied as particular case of epicyclodal gearing

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of PG reducer provides very high reduction ratios when the eccentricity e is smaller than the primitive radius of the external ring gear r. This can be easily verified by using the Willis formula (1):

$$\tau_0 = \frac{r-e}{r} = \frac{-\Omega}{\omega_2 - \Omega} \Rightarrow \omega_2 = -\frac{\Omega e}{r-e} = -\frac{\Omega(z_3 - z_2)}{z_2} \tag{1}$$

The solution introduced in the scheme of Figure 2 provides a clearly advantage in terms of possible reduction ratios, but it has at least three main drawbacks. The first one concerns the difference between the number of teeth of the ring (z_3 in Figure 1) and the one of the planetary (z_2 in Figure 1), which has to be minimized; therefore, the optimal solution in terms of achieved reduction ratio corresponds to the lowest possible difference between z_3 and z_2 (e.g. one tooth).

Unfortunately, for standard involute gears the difference between z_3 and z_2 cannot be lower of three-four teeth in order to avoid the well-known secondary profile interference, as described in many handbooks (Henriot, 2013).

The second drawback concerns the shaft, which is rotating respect to the fixed frame with an eccentricity e. Consequently, a homokinetic joint/transmission system is needed to compensate the variable misalignment between the planetary rotating axis and the fixed frame's one.

Eventually, the last drawback concerns unbalanced inertial forces transferred to the machine frame, which unavoidably appear for a high reduction ratio, since only one planetary can be introduced. However, this last issue can be trivially solved by introducing some counterweights on the shaft.

2.2 Cycloidal drive

CDs are acknowledged to provide high speed ratio in a single stage (Gorla *et al.*, 2008; Blagojevic *et al.*, 2014). By adopting a cycloidal profile, it is possible to generate a planetary-coupled planetary gear and a ring profile, which corresponds to the optimal condition (2). This configuration assures the possibility of obtaining the maximum reduction ratio respect to the number of lobes z_2 of the planetary:

$$z_3 - z_2 = 1 \tag{2}$$

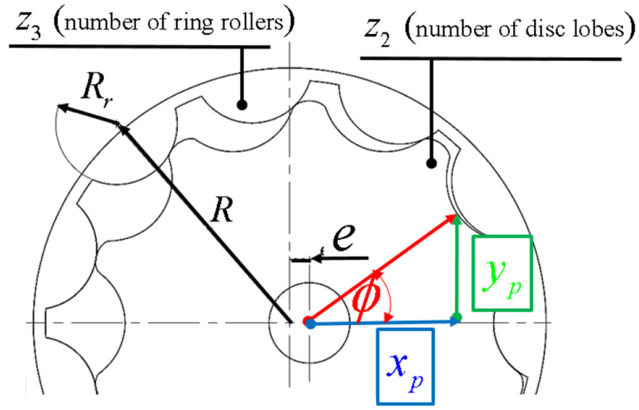
In particular, according to literature (Shin and Kwon, 2006), the cycloidal profile of the planetary can be parametrically generated according equations (3), being z_2 the number of the lobes of cycloidal disc and z_3 the number of rollers [assuming valid the relation (2)] as visible in the scheme of Figure 3.

$$\begin{aligned} x_p &= R\cos(\phi) - R_r\cos(\phi+\theta) - e\cos(z_3\phi) \\ y_p &= -R\sin(\phi) + R_r\sin(\phi+\theta) + e\sin(z_3\phi) \\ \theta &= \tan^{-1}\left(\frac{\sin(z_2\phi)}{\cos(z_2\phi) - \dfrac{R}{ez_3}}\right) \end{aligned} \tag{3}$$

More specifically, referring to equations (3) and to the scheme of Figure 3, the following symbols are adopted:

- ϕ polar coordinate of the planetary disc profile;
- $x_p(\phi), y_p(\phi)$ corresponding Cartesian coordinates of the planetary disc profile

Figure 3 Simplified scheme to explain the generation of cycloidal profiles



- e eccentricity;
- R, R_r correspond respectively to the distance of rollers respect to the centre of the ring and the roller radius; and
- θ contact angle between roller and lobe.

To reduce friction losses, the better configuration foresees pivoted rollers that are free to rotate respect to their centre.

The planetary disc is rotating with an eccentricity e respect to the fixed frame: to transmit the motion to an external shaft, a homokinetic joint/transmission is needed. One of the most adopted solution is the "pin in hole", whose configuration is represented in Figure 4.

The example represented in Figure 4 is referred to a 1/100 reduction stage, and it is interesting to observe that the disc lobes are quite small, consequently involving relatively precise and high-cost mechanical machining.

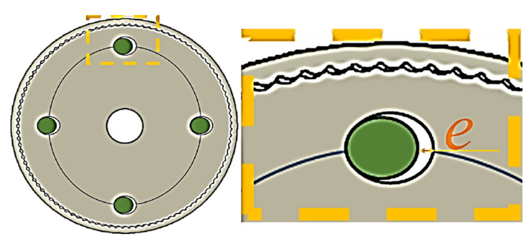
Two counter posed planetary discs should be used to produce a self-balanced version of cycloidal gearbox.

2.3 Harmonic drive

In the HD (Fathi *et al.*, 2001; Chen *et al.*, 2014), the three functional troubles of the kinematic scheme of Figure 2 are solved using a flexible planetary, the so called "Flex Spline" (Taghirad and Belanger, 1998) with z_2 teeth coupled to a fixed ring with $z3$ teeth.

The deformation of the flex spline is caused by the particular shape of the carrier (called Wave exciter) without the need of any eccentricity e, since the obtained gear transmission satisfies the relation (2), expressed in terms of number of teeth, thanks to the deformability of the flex spline.

Figure 4 Simplified scheme of pin-hole homokinetic joint

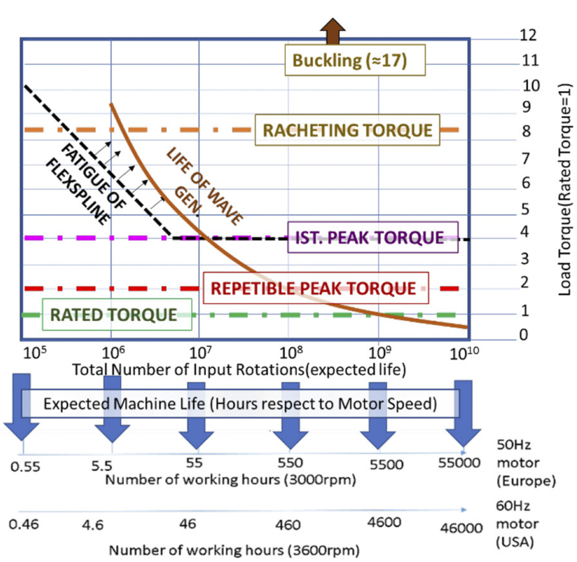


In particular, the correct coupling without interferences between the fixed ring's teeth and the flexible spline is guaranteed by the deformable behaviour of the last one. Flex spline's teeth are engaged and disengaged thanks to the radial deformation of the spline, so avoiding the secondary overcut/interference troubles that are unavoidable for rigid involute gears where z_2 and z_3 are near to be equal. Therefore, with an optimal design (Taghirad and Belanger, 1998; Harmonic Drive GMBH, 2019) it is possible to directly use involute gears without further correction, by adopting an optimal deformation law of the flex-spline.

Since the flex spline is coaxial to the fixed ring and the eccentricity is null, there is no need of homokinetic joints on the output shaft, which is rigidly connected to the end section of the flex spline. In addition, as the flex spline is mounted with null eccentricity, no unbalance has to be compensated.

Unfortunately, the flex spline is quite sensitive to overloads, since it carries teeth whose functioning is similar to gear and it undergoes high cyclic deformations. According to both, literature and technical documentation from manufacturers (Harmonic Drive GMBH, 2019), flex spline and wave generator are highly sensitive to mechanical fatigue. Therefore, the system is typically sized for a finite operational life. Figure 5 shows a tabulated relationship (Harmonic Drive GMBH, 2019) between loads and failure of the machine in terms of working cycles: even considering an applied load equal to the nominal/rated torque of the gearbox, the life of the gearbox is limited to about 5500 cycles (50 Hz motor input frequency) or 4400 (60 Hz motor frequency). Even a relatively small overload factor equal to 2 produces a reduction of the expected life of the transmission to one tenth. Indeed, it is possible to infer that HD has to be carefully protected from possible overloads to assure the prescribed durability. As HD exhibits modest reliability

Figure 5 Overload vs expected life of an Harmonic Drive according Manufacturer tech. doc



Source: Data in figure are referred to criteria proposed by Harmonic Drive GMBH, 2019) modified in order to take count of different input motor speed

respect to overloads, the designer is compelled to adopt cautious sizing, but also this possible choice involves negative drawbacks. In fact, efficiency of HD is very sensitive to various parameters and in particular transmission efficiency is drastically reduced when the transmitted torques are low respect to the rated one (partial loading), (Gorla *et al.*, 2008). More specifically, efficiency is very high for full loading conditions (about 90%), while for a partial load (for instance 10% of the nominal one) efficiency decreases to 40%–50%. This behaviour is explained considering that the deformation of flex-spline involves losses that are quite insensitive respect to the value of transmitted torque. As losses are almost constant, efficiency decreases for partial loads that corresponds to lower transmitted power. For the cited reasons, flex spline is a structural demanding component that has to be carefully designed and verified. Moreover, since the loads applied to the transmission of a concrete mixer are affected by uncertainties and high variability, the adoption of HD for this kind of applications must be avoided.

2.4 Considerations about the state of the art

According to the limits observed in the potential alternative gearbox solutions surveyed in this section, it is possible to assert that an accurate design process needs to be followed to find cost-effective and compact solutions.

Indeed, while currently adopted solutions are considered non-sufficiently efficient by the firm, other alternatives are not yet mature or are usually exploited in completely different applications with peculiar design requirements.

Therefore, as indicated by systematic design procedures (Pahl *et al.*, 2007; Fiorineschi *et al.*, 2016), it is important to extract a comprehensive set of design requirements. The gathered information is used to correctly and systematically define the design task, identify solutions and select the preferred ones.

Accordingly, the following sections are focused on the information required to perform the mentioned assessment, as well as to provide a first preliminary concept of a potentially applicable solution.

3. Eliciting design specifications

By referring to literature checklists (Pahl *et al.*, 2007; Pugh, 1991)., the following set of design requirements has been extracted by means of multiple interview sessions with the firm staff. These requirements are general, and they have been defined according to the machines for which the new solution is relevant.

- *Geometry*: a cylindrical or conical shape of the whole transmission system is highly desirable to ease the integration within the components of the different machines. Moreover, a certain degree of modularity is required to make the solution adoptable in different machines.
- *Kinematics*: reduction ratio 1/100 is required, while 1/50 is acceptable only if considering the use of more expensive four pole motors.
- *Forces*: transmitting desired torques with high overload capability. In this study a load corresponding to output Torques between 150 and 350 Nm are supposed. A safety

factor of about 2 respect to nominal mean values should be considered realistic.

- *Materials*: employed materials (especially for external parts) should be compatible with harsh environmental and operational conditions (water, dust, sand, overloads, impacts). Even temperature is a quite variable operational parameter since a construction yard should be exposed to extremely cold (−40C°) or hot (50°C–60°C) weather conditions.
- *Production/Assembly*: the number of components should be reduced (in relation to the current firm solution).
- *Energy*: gearbox efficiency almost equal respect to existing solutions.
- *Use/Operation*: vibration and acoustic noise level should be equal or inferior respect current solutions.
- *Costs*: Concerning the speed reducer, an indicative cost constraint is important for mass production (no more than €30, for a production of about 10.000 pieces/year).

4. Evaluating different drive solutions against extracted requirements

Curved linear motors are not currently sufficiently mature to be considered here a valid alternative to existing solutions, due to the lack of comprehensive information about their behaviour and costs. Similarly, for conventional direct drive, it is possible to assert that the lack of comprehensive information about its application on concrete mixers implies the need for comprehensive investigations, which are not affordable in a short-term scenario.

Therefore, PG, HD and CD constitute the potentially valid alternative to be investigated in this paper. However, as shown in Section 2, PG are not capable to provide the needed ratio in a single stage, leading to an undesired increase of component count. According to this further consideration, it is possible to assert that HD and CD are potential alternatives, which however are normally used in robotics, i.e. with very different specifications.

Table 1 compares HD and CD with the current gearbox solution, against a set of main design requirements extracted from the list presented in Section 3. Authors, in collaboration with the firm's staff, have performed this preliminary assessment.

As shown in Table 1, HD can be embodied in very compact architectures. However, it is not well suited for the application. Indeed, overloads are not admitted, and high precision tolerances are required, thus avoiding cheap manufacturing procedures.

Table 1 CD and HD Compared with current gearbox solution

Specification	HD	CD
High reduction ratio vs size	++	+
High Overload Capability (min 200% better 500%)	−	++
Simple and cheap manufacturing	− −	−
Efficiency	−	=
Vibration and Acoustic Noise Reduction	=	=

Notes: ("+ +" stands for "better", "+" stands for "quite better", "=" stands for "equal", "- "stands for "quite worse", and "- -" stands for "worse")

Differently, although CD embodiments can be quite bigger than HD, their overload capacity is quite good. However, the high reduction ratio required by this application involves the execution of precise geometries that should be relatively difficult or expensive to be obtained for a low-cost industrial product. This is especially true for HD, whose working principle requires to deal with the interaction between rigid and deformable toothed parts.

Moreover, HD efficiency strongly depends on the load conditions and this feature represents a non-negligible drawback for the considered application. On the contrary, CD has an efficiency more stable under time-dependent loads.

Therefore, to face the recalled critical aspects, the basic and "conceptual" kinematic shown in Figure 2, from which both HD and CD have been derived, has to be modified.

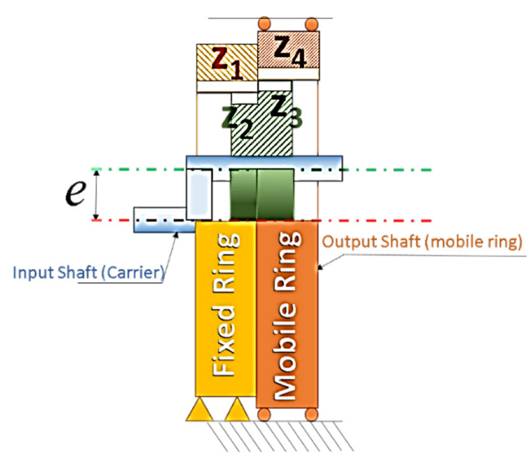
In particular, Figure 6 shows the simplified scheme of the drive, which has been adopted to further refine the design of the adopted reduction stage. More specifically, a carrier is used to move a planetary that has two toothed surfaces (z_2 and z_3 are the number of teeth), which, respectively, engage a fixed ring with internal teeth (z_1 teeth) and a rotating mobile one (z_4 teeth).

The rotating ring is also connected to the output shaft of the transmission system. Therefore, the resulting proposed solution should be considered a particular case of double stage transmission (Lin *et al.*, 2014; Mihailidis *et al.*, 2015).

Respect to the scheme of Figure 2, the proposed one avoids the need of a homokinetic joint to transmit the motion to the output shaft, since the rotation axis of the mobile ring is fixed respect to the frame of the machine. In this sense, the proposed solution is a compound cycloidal reducer that resemble a Wolfrom scheme (Mihailidis *et al.*, 2015). However, in reference to the latter solution there are two main differences:

1. The number of gears is quite smaller and with a proper design (as it is demonstrated in following sections) the machine can be produced and assembled with relatively poor tolerances.
2. The planetary is unbalanced: this drawback, it's not very important for the application to concrete mixers, since rotation speed are relatively low and the unbalanced

Figure 6 Reference double stage kinematic scheme



masses of the transmissions are near to negligible respect to the physiological unbalance introduced by the multi-phase materials (solid and liquids) that have to be shuffled in the concrete mixer.

The kinematic behaviour of the transmission shown in Figure 6 can be studied using the Willis formula (4):

$$\tau_0 = \frac{z_1 z_3}{z_2 z_4} = \frac{-\Omega}{\omega_4 - \Omega} \Rightarrow \frac{\omega_4}{\Omega} = \frac{(z_1 z_3 - z_2 z_4)}{z_1 z_3} \tag{4}$$

With the proposed scheme it's possible to obtain a very high reduction ratio, if condition (5) is approximately verified:
$\frac{z_1 z_3}{z_2 z_4} \approx 1$(5) Moreover, condition (5) implies that to obtain a high reduction ratio, the number of teeth of rings and planetary should be almost the same (6):

$$z_1 \approx z_2 \approx z_3 \approx z_4 \tag{6}$$

Under these conditions, involute gears cannot be adopted due to the troubles related to interference/undercut of profiles. The adoption of flexible gearing such as in the case of single stage HD is very difficult or almost impossible for the much more complex scheme of Figure 6. Finally, it is worth of noting that all the troubles of sizing respect to variable loads that negatively affects the reliability of the single stage HD, are even more difficult to be solved for a more complex double stage solution. For the rapid prototyping of a double stage transmission system, the sizing of rigid components, such as the ones corresponding to a reducer with cycloidal gears, is much faster respect to a solution with flexible elements. This is a relevant advantage of cycloidal vs flexible/harmonic solution, especially considering the possibility of an easy scaling of the future prototype respect to different size of machines.

Adopting cycloidal gears, minimum difference between the number of teeth (cycloidal lobes and rollers) between the engaging gears can be reduced to one as summarized below:

$$\begin{aligned} z_2 &= z_1 - 1; \\ z_3 &= z_2 + 1 = z_1; \\ z_4 &= z_3 + 1 = z_1 + 1; \end{aligned} \tag{7}$$

By substituting conditions (7) in the calculation of the transmission ratio (4), it is possible to obtain an expression in which the reduction managed by the gearbox is a function (8) of the number of teeth of just one gear (the first one):

$$\frac{\omega_4}{\Omega} = \frac{(z_1 z_3 - z_2 z_4)}{z_1 z_3} = \frac{1}{z_1^2} \tag{8}$$

From (8) and (7) it is possible to automatically calculate the set of gears that assure the desired transmission ratio, as described in Table 2.

Table 2 Set of gears able to obtain desired transmission ratio

	$\frac{\omega_4}{\Omega}$	z_1	z_2	z_3	z_4
symbolic	$\frac{1}{z_1^2}$	z_1	$z_1 - 1$	z_1	$z_1 + 1$
numeric	1/100	10(rollers)	9(lobes)	10(lobes)	11(rollers)

It is worth to notice that by adopting the proposed solution, the gears are composed by a small number of lobes (about 10).

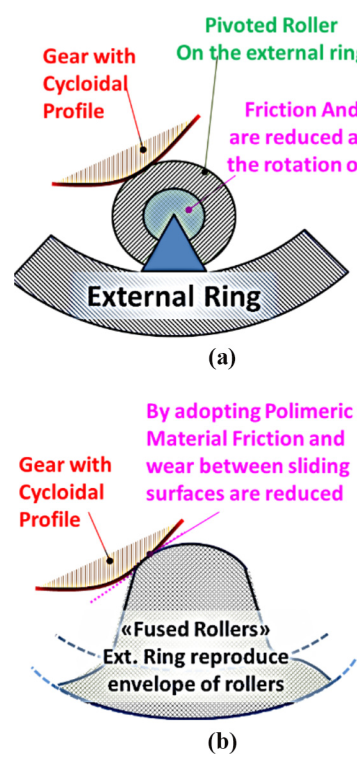
Therefore, respect to a single stage cycloidal reducer with similar radial encumbrances, the size of rollers and lobes is about ten times larger. This is a clear advantage for the manufacturing process, since a larger scale of profiles and rollers generally reduces performance sensitivity against production tolerances. This should be clearly an advantage also for the application of unconventional materials and production techniques.

5. On the usage of polymeric-compound materials

There is an increasing interest in using polymeric materials for transmission systems, whose employment is supported by the improvements of high density structural polymers (duPont,2019), which have excellent structural and self-lubrication properties able to improve resistance to wear and to chemical degradation.

Low friction and good wear resistances of polymeric materials are of fundamental importance also to introduce some significant design and assembly simplifications of the cycloidal transmission systems. As visible in Figure 7(a), pivoted rollers on rings are adopted to reduce losses and wear of cycloidal profiles. Adopting polymeric materials, friction between sliding profiles is reduced allowing the elimination of pivoted rollers and the adoption of rings with "fused rollers" as visible in Figure 7(b): the ring is a monolithic component whose profile is shaped with a profile that reproduces the envelope of roller surfaces. This solution visible in Figure 7(b) can be adopted also in metallic cycloidal drives, but

Figure 7 Conventional pivoted rollers (a) vs "fused rollers" construction of the ring (b)



corresponding drawbacks like higher wear and friction losses are better compensated by the superior tribological properties of polymeric/compound materials.

The usage of polymeric materials for cycloidal gears involves several further advantages if compared to conventional involute ones (Biernacki,2014): first, in cycloidal drives the material is mainly stressed by compression loads, while in involute gears the tooth is subjected mainly to flexural and shear stresses. This is a clear advantage for the usage of polymeric materials in cycloidal drives with fused rollers since mechanical properties of polymeric materials are often optimized respect to compression stresses.

Moreover, polymeric materials have a Young Modulus E that is very low (few thousands of MPa respect to the typical value of about 200,000 MPa of steel). According to Hertz theory (Sneddon, 1965), maximum contact pressure p_{max} between two cylindrical bodies (the lobe of the cycloid and the roller) is described by:

$$p_{\max} = \sqrt{\frac{E_* F}{R_* L}} \qquad (9)$$
$$where\,\frac{1}{E_*} = \frac{1-\upsilon_1^2}{E_1} + \frac{1-\upsilon_2^2}{E_2}; \qquad \frac{1}{R_*} = \frac{1}{R_1} + \frac{1}{R_2}$$

R_i curv. radius of the i − th component
E_i Young Modulus of the i − th component
$\frac{F}{L}$ force over unit lenght transmitted in the contact

Indeed, the lower stiffness of polymeric materials respect to steel implies a strong reduction of the maximum pressures to which the contact patch is subjected. Considering the same geometries and loading conditions, peak stresses for profiles manufactured with polymeric materials should be reduced to one about 1/3–1/5 respect to corresponding steel ones (the outcomes depend on the properties of the considered polymer).

From a physical point of view, these results can be easily explained: a lower Young modulus implies higher deformations and consequently larger contact patches on which transmitted forces can be distributed, so implying lower pressures and stresses.

Moreover, in CD the number of lobes that are in contact with rollers is relatively high. Therefore, a reduced stiffness of the material should cause a uniform distribution of loads over a larger number of teeth with a further improvement of the loading capabilities of the transmission.

Eventually, it is worth to notice that a reduced stiffness of the material can also reduce negative effects introduced by manufacturing tolerances. A manufacturing tolerance of cycloidal gears can be considered as an equivalent imposed deformation disturbance between the engaging profiles.

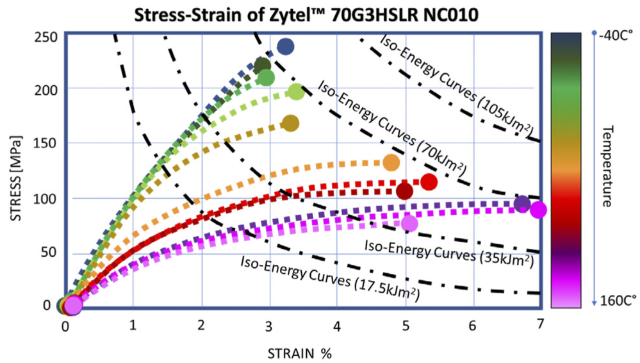
According to the definition of Young Modulus E (10), stress caused by an imposed deformation is proportional to E:

$$E = \frac{\sigma}{\varepsilon} = \frac{stress}{deformation} \qquad (10)$$

Consequently, in presence of assembly misalignments, induced stresses are lower when the adopted materials are softer.

In Figure 8, it is visible the strain-stress behaviour of a compound polymeric material, i.e. a nylon polymer (Dupont Zytel^{TM}) enforced with glass fibres. This is an example of a typical compound polymeric material that is often adopted for the construction of cams and gears (duPont, 2019).

Figure 8 Stress–strain curves of Nylon enforced with fibre glass performed with data taken from manufacturer datasheet



Source: duPont, (2019)

The material has a typical thermo-plastic behaviour, since is relatively "harder" at low temperatures (in the range 0, −40 C° tensile strength is 200–240 MPa but no troubles of fragility are recorded), and softer at higher ones (at 60 C° tensile strength is reduced to about 130 MPa).

This behaviour considering the Hertz contact model (9) is self-stabilizing: in fact, the ratio of maximum contact pressure at minimum and maximum temperature is equal to the root square of the ratio of equivalent elasticity modulus E^* (11).

$$\frac{p_{\max\_(-40C^\circ)}}{p_{\max\_(60C^\circ)}} = \frac{\sqrt{\frac{E_{*\_(-40C^\circ)}}{R_*}\frac{F}{L}}}{\sqrt{\frac{E_{*\_(60C^\circ)}}{R_*}\frac{F}{L}}} = \sqrt{\frac{E_{*\_(-40C^\circ)}}{E_{*\_(60C^\circ)}}} \approx 1.6 - 1.7 \qquad (11)$$

The ratio between max tensile stress that the material it can sustain at different temperatures according to Figure 8 is not much different (1.8/1.7 according approximations).

Consequently, a softening behaviour involves a reduction of contact pressures, that largely compensate the decrease of the material tensile strength. This feature is relevant for the usage of the material on components that are heavily loaded by compression contact loads, such as gears with cycloidal profiles. Another interesting property of these materials regards resilience: temperature causes polymer softening but its resilience is much less affected; this is a very important property for the construction of gears which should be subjected to dynamic loads and to mechanical fatigue. Also, specific weight of these polymeric materials is typically a degree of magnitude lower respect to steel (from 0.7 to 1.5 Kg/dm3 depending on the chosen material). Consequently, an increase of encumbrances of the components due to relatively modest mechanical properties of the material respect to steel, produces limited effects in terms of weight, especially for moderately compression loaded components.

6. Preliminary design of the cycloidal gearbox

To verify the feasibility of a cycloidal solution, authors performed a systematic conceptual design process based on problem-solution co-evolution(Fiorineschi *et al.*, 2016), to generate a set of preliminary concepts and to select the preferred one, according to the design requirements (Pugh, 1991).

Once a first preferred concept was identified (Figure 9), more detailed embodiment evaluations have been performed to evaluate the stress conditions of the critical elements of the CD gearbox.

From Figure 9 it is clearly advisable the possibility of a compact integration of the proposed solution within a wide variety of possible machines (in this example a concrete mixer to which most of the performed design activities are referred). More precisely, a flexible multi-body FEM model has been realized for this purpose.

For design purpose, a maximum transmitted torque of about 350 Nm is considered.

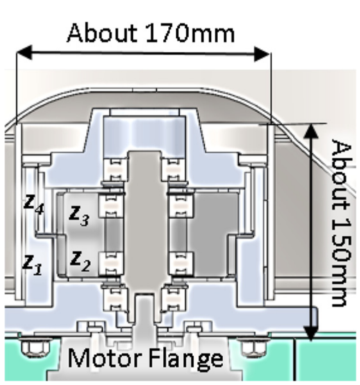
For the transmission, authors considered the double stage CD gearbox configuration, briefly described in Figure 6 and Table 2.

To perform preliminary FEM simulations and sizing of the system, the nylon enforced with fiberglass, previously introduced in Figure 8, has been selected as a candidate material.

According to the dimensions of the machines in which the solution will be adopted, it was supposed to realize a cylindrically shaped transmission system with encumbrances corresponding to an external diameter of no more than 170 mm and an equivalent length that is almost the same (about 150 mm). In this way, as visible in the schemes of Figure 9, the proposed transmission system should be relatively simple to be integrated in all the structures of the relevant machines.

Considering the design of cycloidal profiles, and according to equation (3), the external radius *R* of each cycloidal ring is constrained by the imposed radial encumbrances of the reducer. In addition, the number of teeth and rollers is known (Table 2 for values) due to the desired transmission ratio (1/100). Consequently, the design of profiles and rollers has to be optimized by considering the two remaining parameters, i.e. the eccentricity e and the roller radius R_r.

Figure 9 Preliminary design, example of integration of the gearbox in an existing concrete mixer machine



6.1 Flexible multi-body model

To identify the optimal profiles of lobes, and to perform an accurate sizing of the critical parts of the reducer, authors adopted a flexible multibody model developed using Comsol Multiphisics™, which is briefly described in the scheme of Figure 10. The proposed CD is decomposed in two single stage systems that are simulated using a simpler and faster planar finite elements model. The coupling between the two models is managed in terms of equivalent load and imposed motions mutually exchanged in the following way: the first stage calculates motion of both carrier and planetary gear that are imposed to the second stage; the second stage is then able to calculate the corresponding loads that are then applied to the first stage. As boundary/loading conditions, for the first stage it is imposed the motion of the carrier, while for the second one, load torque applied to mobile ring is known from design specifications. The advantage of this simplified approach is to drastically reduce the computational time associated to the resolution of a single cycloidal stage. Moreover, also the calculation of contact conditions (performed with augmented Lagrangian approach) is drastically simplified.

Therefore, computational resources needed to solve the systems are affordable, allowing the execution of multiple simulations to find near to optimal profiles of gears and rollers.

Another interesting aspect concerns the plastic material modelling and the effect in terms of computational load. To simplify calculations, a constant value of the material Young modulus has been considered, which is calculated by linearizing the stress strain curve for very small deformations (0.01) at a constant temperature (20 C°). Young modulus calculated in this conditions (10,000 MPa) is a bit overestimated respect to the one corresponding to maximum loading conditions in the contact patch (which is around 1,000–2,000 MPa). This simplification introduces limited errors which are appreciable only for stress-strain condition far higher respect to the linearization interval.

This simplification leads to conservative results in terms of estimated contact pressure, since a softer material involves [according to (9)] larger contact patches and lower contact pressures.

However, calculated contact patches are relatively larger (Figure 11, whose results are referred to a temperature of about 20 C°) than the ones calculated for steel gears. Therefore, also for finite element calculations, size and number of elements that have to be used are more relaxed than the corresponding case with steel elements.

All these practical considerations are supported by know-how and best simulation practice that are often available in finite elements handbooks for engineers (Kardestuncer and Norrie, 1987, Bu Liu and Quek, 2013). Figure 12 shows an example of the adopted mesh, which is relatively rough and simple.

Thanks to the availability of a model that should be executed in a relatively fast way, the authors performed multiple simulations to evaluate the effects derived by the adoption of different design factors. In particular, the following variable parameters have been considered:

- number of teeth lobes (9-10 for the two stages);
- eccentricity of the carrier *e;*
- roller Radius *Rr;*

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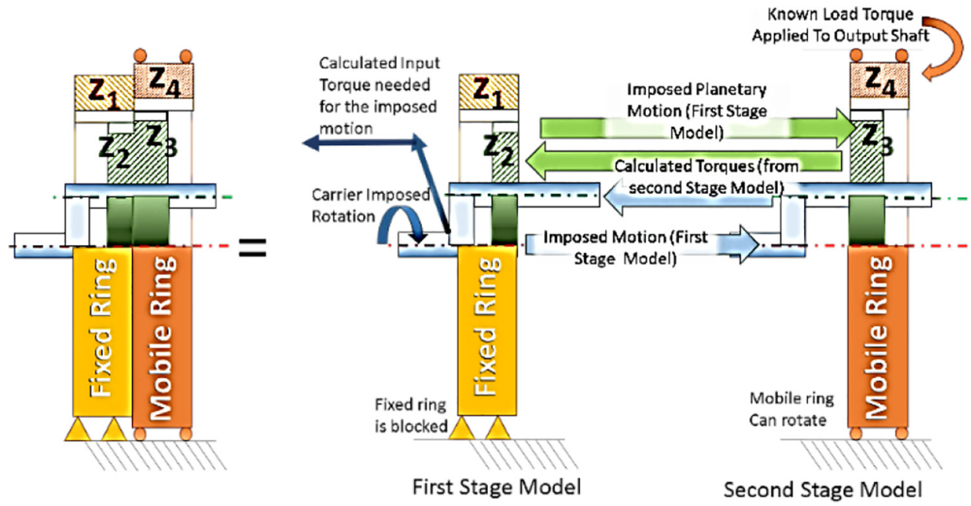
Figure 10 Simulation of double stage cycloidal gearbox is decomposed in two planar models of single stage system exchanging loads and imposed motions



- temperature/Young modulus: exploring extreme operational temperatures authors also investigate sensitivity respect to variations of the material Young Modulus;
- concerning friction, authors preferred to introduce only one value (0.1) which was considered quite conservative respect to the expected one (0.05–0.08);
- for both cycloidal profiles, a maximum axial length equal to 30 mm is considered, as also axial encumbrances are limited by specifications.

Some results are shown in Figures 13, 14 and 15, where the behaviour of the system is shown, in terms of maximum contact pressure, equivalent Von Mises Stress, and finally dissipated power on contact surfaces. The first two indexes (contact pressure and equivalent stress) are quite indicative of the structural loads to which the teeth surfaces are subjected.

Surface pressures are minimized with roller diameters around 10–15 mm, while equivalent Von Mises stresses are minimized for a roller radius of about 15–20 mm.

Consequently, authors decided that a roller radius of about 15 mm can be an acceptable compromise.

Moreover, as visible in Figure 15, the level of dissipated power is quite similar in the two stages, then it is possible to infer that also the wear rates of both stages are quite similar.

Concerning the eccentricity e, lower values (4 mm) involve higher level of dissipated power, while higher ones (6 mm) produces more demanding solicitations in terms of equivalent Von Mises stress. Therefore, an intermediate value of eccentricity equal to 5 mm is considered an acceptable compromise.

Therefore, a near to optimal solutions corresponds for both stages to roller radius of 15 mm and an eccentricity of about 5 mm.

Main features of the proposed solution are summarized in Table 3: higher stresses are associated to low temperatures when material stiffness and resistance are higher. At higher temperatures (100 C°), material properties drastically decrease; however, also maximum pressures and stresses are reduced as a direct consequence of material softening.

Chosen Wolfrom layout and polymeric materials also involve an interesting property: proposed transmission is quite insensitive to geometric errors on roller and gear profiles.

This property has important consequences: poor production tolerances can be accepted; also overall performances of the system are relatively less sensitive to light profile modifications due to wear.

In this work, this topic is only preliminarily investigated in a simple way by obtaining the results visible in Figure 16. Finite elements analysis of proposed transmission described in Table 3 are repeated considering the application of known offset to roller profiles. In this way, by tolerating value of roller radius respect to its nominal values, is quite easy to introduce a known error in contact geometry. Results of Figure 16 are repeated for the highest operating temperature considered in Table 3: as clearance/backslash between roller and gear profile increases, the number of contact points, and consequently of engaging rollers and lobes, decreases. For this reason, maximum stress on the most loaded lobe increases. On the other hand, by increasing the radius of rollers is possible to cause a slight interference between profiles, in this way contact is distributed over a large number of lobes but higher pressures on contact patches produce an increase of dissipated power with negative consequences not only on efficiency but also in terms of wear. Probably a further increase of interference should also produce a drastic increase of maximum stresses.

Encouraging results of Figure 16 (tolerated errors of about one tenth of mm) are justified by the following considerations:

- With Wolfrom topology, cycloidal gears have a reduced number of lobes (about one tenth respect to conventional cycloidal drives). So, lobes are much bigger respect to the ones of conventional cycloidal transmissions. With bigger lobes, the same absolute errors on profiles have minor impact on kinematics.
- Polymeric materials are less resistant respect to steel alloys. Properties of a good steel are about ten times higher

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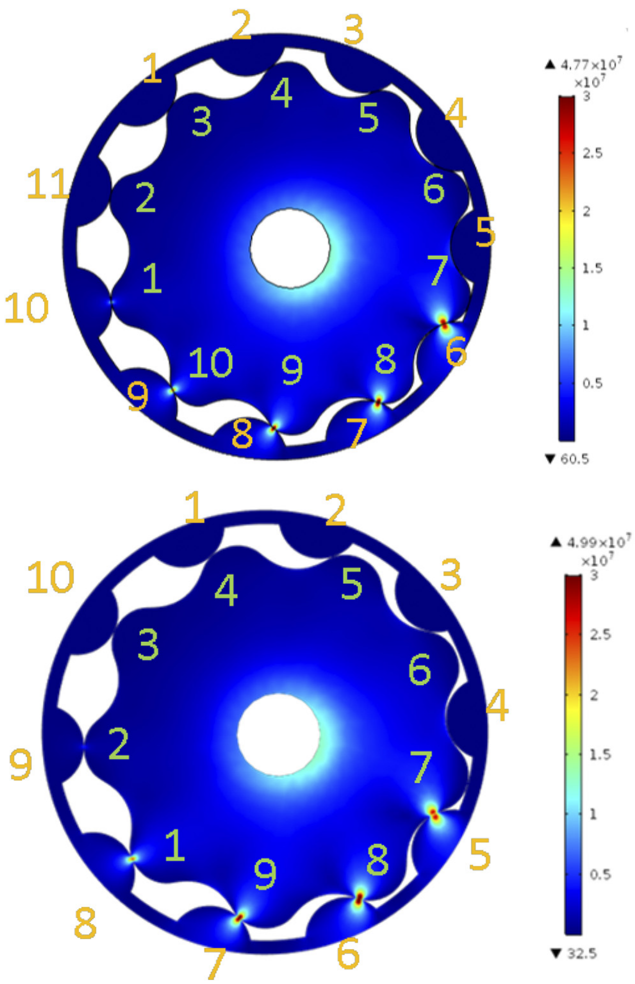
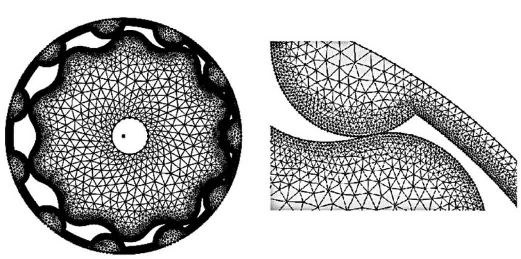
Figure 11 Simulation of stress distribution of both stages (nominal chosen configuration) in which is quite evident the simple geometry and the relatively large contact patches



Note: Due to low stiffness of plastics respect to steel

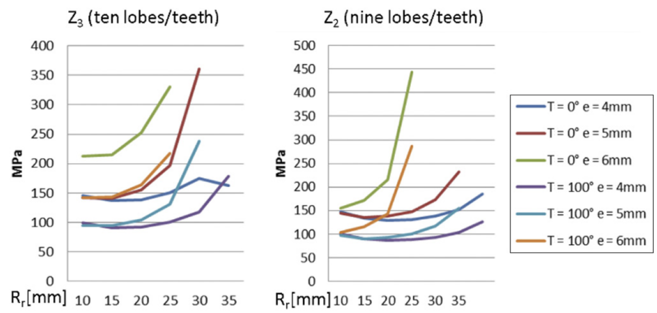
Figure 12 Example of refined mesh adopted for simulation of contact between fused roller of the ring and cycloidal profile over the planetary



Note: Stress values are in Pa or N/m²

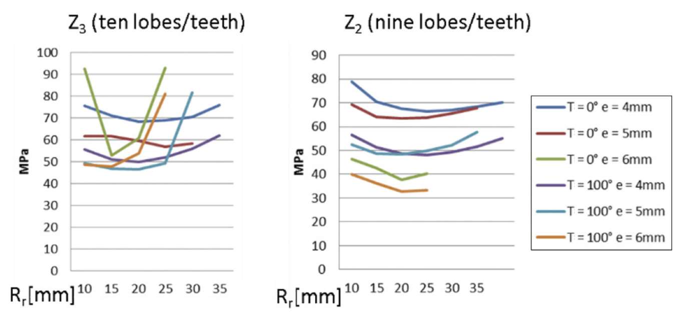
respect to the chosen polymeric material. The transmissible loads for a gear increase with a cubic power law of its dimensions. Therefore, a polymeric gear is about two times bigger than a steel one and the absolute

Figure 13 Parametric simulation of simulated maximum contact pressures respect to the number of teeth, the roller radius R_r the eccentricity e and different extreme operating temperatures



Note: Axial depth 30mm

Figure 14 Parametric simulation of simulated maximum Von Mises eq. stress respect to the number of teeth, the roller radius R_r the eccentricity e and different extreme operating temperatures



Note: Axial depth 30mm

precision on geometry can be relaxed consequently. This increment of the gear dimensions due to material properties is compensated by adopting a Wolfrom topology, which helps to maintain acceptable encumbrances of the transmission.

- Young modulus of polymeric materials is about one hundredth respect to steel, so the same geometric error should produce very limited changes in contact forces between gears. This is especially true for simulations of Figure 16, performed with high temperatures and consequently with low values of Young modulus.
- A small increase of backslash produces a reduction of the contact area between rollers and cycloidal gear. This reduction is localized on the lobes where the contact pressure angles are lower. Thus, backslash mainly affects contact patches that less contribute to an efficient transmission of the torque.

In accordance with the above considerations, it should be concluded that one of the major innovations of the proposed solution is the adoption of a transmission topology (Wolfrom) able to exploit the properties of the chosen polymeric materials at their best. In this way, the proposed transmission is relatively simple, cheap and robust.

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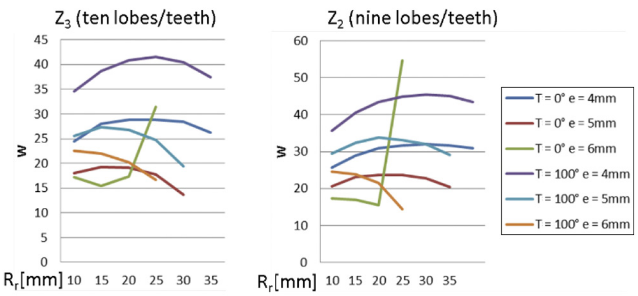
Figure 15 Dissipated power on the two stages respect to the number of teeth, the roller radius R_r the eccentricity e and different extreme operating temperatures



Note: Axial depth 30mm

Table 3 Brief description of the proposed solution

	T = 0°		T = 100°	
	Z = 10	Z = 9	Z = 10	Z = 9
Max Pressure (Max Pressure limit: About 260–270 MPa at 0 C°, About 140 MPa at 100 C°)	141 [MPa]	136 [MPa]	95.0 [MPa]	90.6 [MPa]
Max eq. Von Mises Stress (Tensile Strength limit: About 210 MPa at 0 C°, About 110 Mpa at 100 C°)	61.7 [MPa]	64.2 [MPa]	46.8 [MPa]	48.8 [MPa]
Mean lost Power	19.2 [W]	23.1 [W]	27.3 [W]	32.3 [W]

Note: *for both stages the excentricity "e" is equal to 5 mm and the roller radius is equal to 15 mm

Efficiency η (12) of the proposed solution can be calculated by comparing the desired output power with the input one, which is estimated by considering the sum of the losses due to contact sliding frictional forces and to internal friction of bearings:

$$\eta = \frac{T_4\omega_4}{T_4\omega_4 + \sum_{i=1}^{n} F_i \omega_{r\_i} \rho_i^{d_i}/2} + W_{dis} \tag{12}$$

Where the following symbols are adopted:

- T_{out}, ω_{out}, Output torque and speed.
- W_{dis}, power dissipated between sliding surfaces in contact (it was assumed equal to the mean of the values calculated at T=0° and T=100°).
- F_i,ω_{r_i}, ρ_i, d_i, these are the parameters for the calculation of losses in the i-th, bearing according to commonly adopted formulations (Zienkiewicz and Taylor, 2005), respectively, the normal force, the rotation speed, the friction factor, the diameter.

By introducing numerical values in equation (13) (and/or directly implementing the calculation through the Comsol FEM model), the resulting estimated efficiency is approximately equal to 0.78.

Figure 16 preliminary sensitivity analysis respect to tolerated geometry of rollers

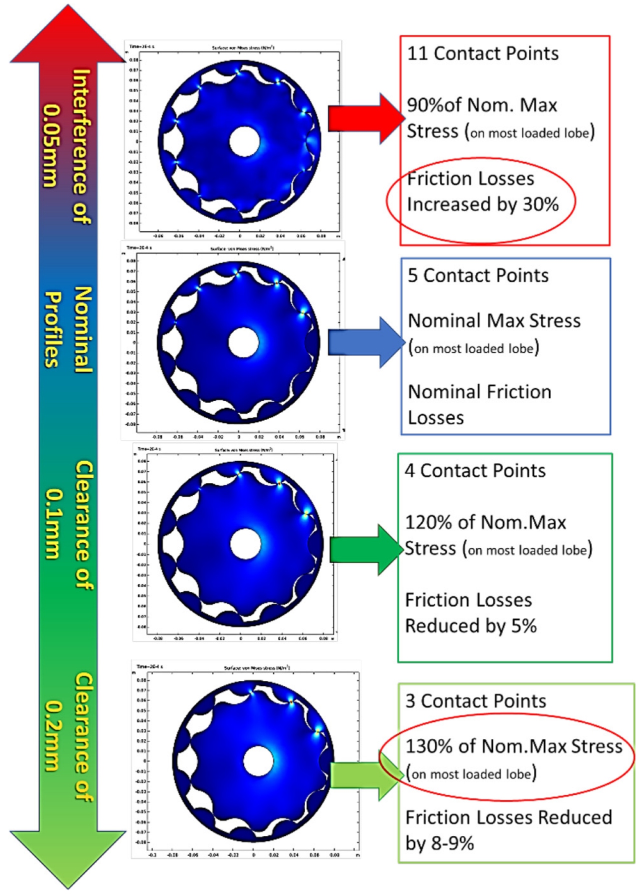


Table 4 Gear drive currently adopted by the firm vs the proposed CD

Traditional gear drive	Proposed CD
24 components	18 components
3 shafts	2 shafts
6 centerings	2 centerings
2 different toothed wheels + 2 different pinions on the shafts	2 different rollers + 2 different wheels with lobes

$$\eta \approx \frac{1100[W](\text{Net} \quad \text{Output} \quad \text{Power})}{1100[W] + 253[W] + 51[W]} \approx 0.78 \tag{13}$$

In particular, in equation (13) losses due to friction on bearings and contact losses are considered. It's noticeable the limited value of contact losses (51 W) which is mainly due to two reasons:

1 Friction factor with polymeric material is reduced.
2 Relative speed between lobes and rollers are relatively low.

However, most of the losses are due to the friction of the bearing elements, which must support the high normal forces produced by cycloidal profiles.

For what concern losses on bearing a relatively higher value of losses is also due the rotational speed of bearings that support

Table 5 Comparing the new proposed solution with standard CD and HD

Specification	HD	Classical CD	Proposed CD
High reduction ratio vs Size	++	+	++
High Overload Capability (min 200% better 500%)	–	++	++
Simple and cheap manufacturing	– –	–	=
Efficiency	–	=	=
Vibration and Acoustic Noise Reduction	=	=	+
Note: (the reference datum is the current gearbox adopted by the firm)			

the fastest shaft: so friction torques are relatively low, but speed is relatively high, so contribution to losses of bearings is important.

Efficiency estimation (13) should be further refined after programmed experimental activities on prototypes. It should be a part of a further optimization process for the final industrialization of the product. According to the approximated relationship adopted by Gorla *et al.* (2008), radial compression forces of a cycloidal disc are inversely proportional to eccentricity e. However, in the current design, the value of eccentricity is limited by the structural properties of materials (higher eccentricities should lead to higher surface pressures). Nonetheless, it is not excluded that different design criteria, in which higher values of e are introduced, could lead to solutions with a slightly worse contact quality (then reducing efficiency), but with reduced bearings loads and consequently with non-negligible design advantages.

6.2 Considerations about manufacturing and costs

Table 4 benchmarks the drive currently adopted by the firm against the solution presented in the previous section. The traditional gear drive is constituted by three reduction stages whose toothed wheels are made of a polymeric material like the one considered for the proposed CD. As shown in the recalled table, the Wolfrom configuration allows a strong reduction of the number of components with respect the gear drive (of about 25%). Furthermore, it is worth to notice that the number of shafts decreases from 3 to 2 allowing a positive impact on the centring constraints in terms of dimensional and geometrical tolerances. Concerning manufacturing, the process the authors considered to produce rollers and lobes is the same currently used for the toothed wheels, i.e. the injection moulding. However, since the sensitivity of kinematics to the potential errors affecting rollers and lobes is lesser than that of the traditional toothed wheels, the minor precision required to produce these components has a positive impact on costs. Therefore, the performed preliminary benchmark shows that the proposed CD is lesser complex and more convenient, in terms of costs, than the gear drive currently adopted by the firm.

6.3 Conclusions and future developments

Aim of this work was to investigate an innovative cycloidal drive able to improve some important aspects of current concrete mixers. Starting from a set of design requirements it was possible to compare different solutions. An innovative CD design was required to efficiently fulfil needs and specifications of the industrial partner of this project. Therefore, by following a systematic design process, a preliminary low-fidelity (Fiorineschi and Rotini, 2019) concept of the solution has been designed and preliminarily evaluated from the feasibility point of view. More precisely, by referring to the design specifications described in Section 3 and the preliminary design outcomes, it has been possible to assess the solution shown in Section 6 in terms of requirements satisfaction (as previously introduced in Table 2). Table 5 compares the scores reached by the new solutions with those of HD and classical CD.

As shown in Table 5, the new proposed CD seems to be potentially embodied by more compact architectures, if compared with the classical CD. The overload capability is of the same magnitude order of the classical CD solution, while the use of polymeric materials allows to ease the manufacturing process (e.g. cycloidal discs obtained by injection moulding). Moreover, the polymeric material also allows a non-negligible noise reduction.

Notwithstanding the encouraging outcomes from this work, it is still necessary to verify the actual potentialities of the proposed concept. To support these further steps and to collect further detailed information, also correct prototyping strategies (Lauff *et al.*, 2019) should be identified and tuned with the design task, by correctly identifying the best match between fidelity and roles of prototypes (Carfagni *et al.*, 2018). Results from future experimental activities are expected to allow the evaluation of parameters like wear (e.g. by comparing the performances with classical gears), reliability and manufacturing/assembly aspects. Additionally, fatigue tests are required, together with comprehensive estimations about the manufacturing time required for this kind of gears, since it is a critical parameter for the actual feasibility of the product (Tung *et al.*, 2020a, 2020b). Indeed, the currently available data about the proposed solution were not sufficient to perform these evaluations. To that purpose, and to verify a wide range of different loading and operating conditions, authors are also considering the application of Hardware In the Loop Testing techniques. The latter have been successfully applied in previous experiences concerning the robust design of actuators for safety relevant applications in the oil and gas sector (Pugi *et al.*, 2017c, 2016).

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