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Research Article

MODELING OF SELECTED DESIGN CHARACTERISTICS OF CAM AND HYPOCYCLOIDAL DRIVES OF HIGH-PRESSURE FUEL PUMPS

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ABSTRACT

The advancement of diesel fuel injection systems is forced by the increasingly stringent exhaust emissions limits. Modern injection systems must generate fuel pressure in excess of 200 MPa, which very often leads to premature wear of the mating components. Generating high pressures results in much force load acting on the pumps. In the case of frequently applied cam-driven pumps, high loads result in significant forces caused by friction. The friction force loads the piston perpendicularly to its axis, which rules out the application of super hard but fragile materials such as ceramics. One of the possible solutions of this problem is the application of unconventional drive systems, the example of which is the hypocycloidal drive. The paper presents the results of simulations comparing selected parameters of the piston motion of the cam-driven pumps and the pumps fitted with hypocycloidal drive. The use of the CAD software also enabled identification of the force acting on the piston (force perpendicular to the piston axis) in the cam-driven pump and its lack in the pump fitted with hypocycloidal drive.

Keywords: common rail, hypocycloidal drives, high-pressure fuel pumps.

INTRODUCTION

In order for modern diesel engines to meet the increasingly stringent exhaust emissions requirements, advanced and precise injection systems are applied. They can perform a direct injection to the combustion chamber or can divide the injection into fuel doses (up to 10) [1]. Modern injection systems can generate very high pressures – currently applied injection pumps pressurize the fuel up to 3000 bars, which loads the powertrain with a force exceeding 6 kN. Currently manufactured pumps use a cam system to drive the pump components [2]. This mechanism, as shown in Figure 1, is composed of an eccentric cam lobe on which a cam ring is fitted. The rotary motion of the shaft acts on the pusher and the piston to pressurize the fuel.

The above-presented solution is very popular because of low production costs achievable through basic technological processes. The application of a cam system does have its downsides – while generating pressure, the friction between the cam ring an the base of the piston increases leading to contact tensions in the area of the cam lobe and the pusher. This phenomenon has been presented on the example of a Bosch CP1 fuel pump (Figure 2).

The effect of such an action is the beveling of the piston inside the cylinder leading to premature wear of the mating surfaces of precision pairs. The occurrence of the force perpendicular to the longitudinal axis of the piston rules out the application of super hard materials (ceramics) that are extremely fragile [4]. The paper treating on the malfunctions of commonly applied high-pressure cam-based pumps proves that further advancement aiming at obtaining higher pressures is difficult because increasing the pressure results

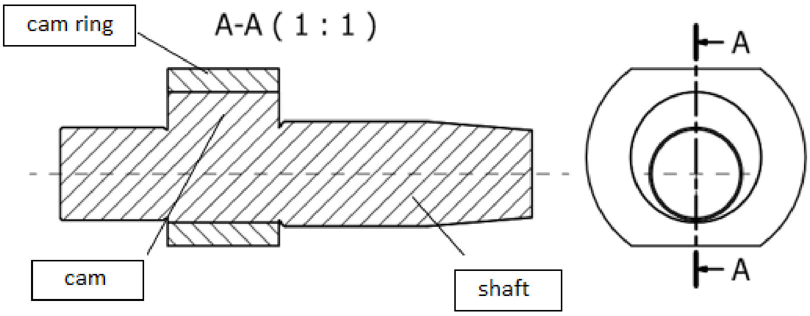

Fig. 1. Schematics of the cam drive used in a dual section common rail pump manufactured by Continental

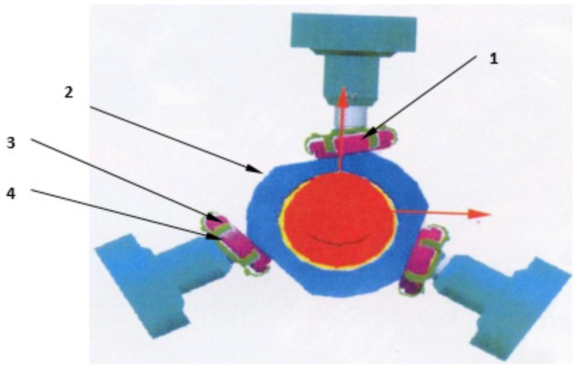

Fig. 2. Cross section of the Bosch CP1 fuel pump in the operating position [3]:
1 – piston base, 2 – cam lobe, 3 – plate, 4 – strainer

in increased friction forces accelerating the tribological processes. Additionally, fuels available worldwide differ in their rheological properties. When comparing the admissible parameters of diesel fuel available in the European Union and the US (Table 1) we may observe a divergence in the values of the admissible lubricity i.e. the most crucial parameter under the considerations of this work.

A mechanism that appears not to suffer from this disadvantage is a hypocycloidal transmission. A hypocycloidal transmission is composed of two wheels, the larger of which has inner teeth while the smaller has outer teeth. The torque is supplied to the smaller wheel generating rotary motion while the larger wheel can rotate around the axis. The smaller wheel moves on the circumference of the larger one and a selected point on the radius of the smaller wheel moves on a hypocycloidal curve. In order to create such a drive, a hypocycloidal transmission was applied of the wheel radius ratio of $R/r = 2$. Such a selection of wheels allows a resultant rectilinear motion, as shown in Figure 3.

THE RESEARCH

The application of a hypocycloidal transmission to drive a fuel pump is a subject of a patent claim of the authors (no. P.418961). This solution brings a variety of benefits – it eliminates the lateral forces described in the introductory part of the paper. An advantage of this solution is high displacement of the working component, which equals to the pitch diameter of the large toothed

Table 1. Comparison of the admissible parameters of diesel fuel – Europe vs. USA [5, 6]

Property	Unit	EUROPE	USA
Density	kg/m3	820 … 845	813 … 863
Viscosity	c.St. (40 °C)	2.0 … 4.5	2.1 … 3.2
Dist. 95% vol rec.	°C	< 360	324 … 344
Total Aromatic Cont.	%	n.a.	16 … 46
Cetane No.		> 51	44 … 57
Sulfur	mg/kg	< 350	23 … 416
Water	mg/kg	< 200	42 … 96
Total Contamination	mg/kg	< 24	0.8 … 3.1
Lubricity	µm (HFRR 60C)	< 460	351 … 648
Alcohol	% vol.	n.a.	< 0.1

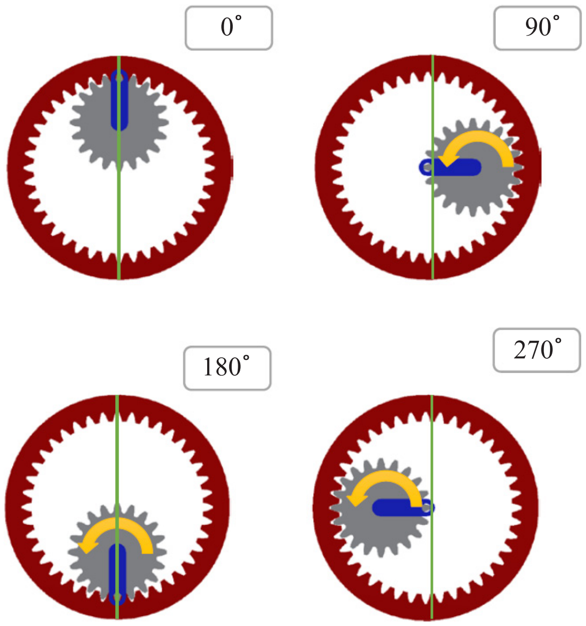

Fig. 3. Principle of operation of a hypocycloidal transmission [7]

wheel. In the analyzed example of the tooth module of 1 mm and the number of teeth 40, the displacement of the working component is 40 mm. A characteristic feature of the hypocycloidal drive applied in the model is the precisely determined gear ratio of the wheels that must equal 2 [8]. What is more, in order for the mechanism to convert the rotary motion to a rectilinear one, it is necessary to apply an even number of teeth. The application of a hypocycloidal mechanism also leads to an increased resistance of the pump to heavy fuels – in the presented pump design, the pumping section has been separated from the driving section, assuming its lubrication is to be performed by the engine or transmission oil. Due to the significant piston displacement, hence the length of the pumping section, it is possible to apply a double labyrinth seal along with the return channels of the leaking fuel to the overflow duct of the injectors. This will minimize the acceptable fuel leakage to the lubricant. The positive features of the pump include the lack of a return spring. Pump solutions based on a cam system for each section have a return spring that ensures a constant pressure of the pusher on the cam. Aside from an obvious increase in the weight of the system, the spring is a component that consumes energy and only some of it is returned in the form of heat. Additionally, due to a large number of cycles, return springs are susceptible to fatigue related damage [9].

In the further part of the paper, selected features of the model of the pump fitted with hypocycloidal drive were compared with the model of a conventional pump. The model of a pump fitted with the hypocycloidal drive has been shown in Figure 4.

As an example of a high-pressure pump utilizing a cam drive, Continental (former Siemens VDO) pump was selected. The view of the pump has been shown in Figure 5. This pump is commonly applied in the engines of the Volkswagen group – most frequently fitted in the 1.6 and 2.0 dm^3 units. Compared to the model of the pump fitted with the hypocycloidal drive, the pump by Continental has a displacement of merely 7.8 mm.

The pump has two opposite placed pumping sections. Such a solution reduces pressure pulsations in the system and allows proper fuel flow for

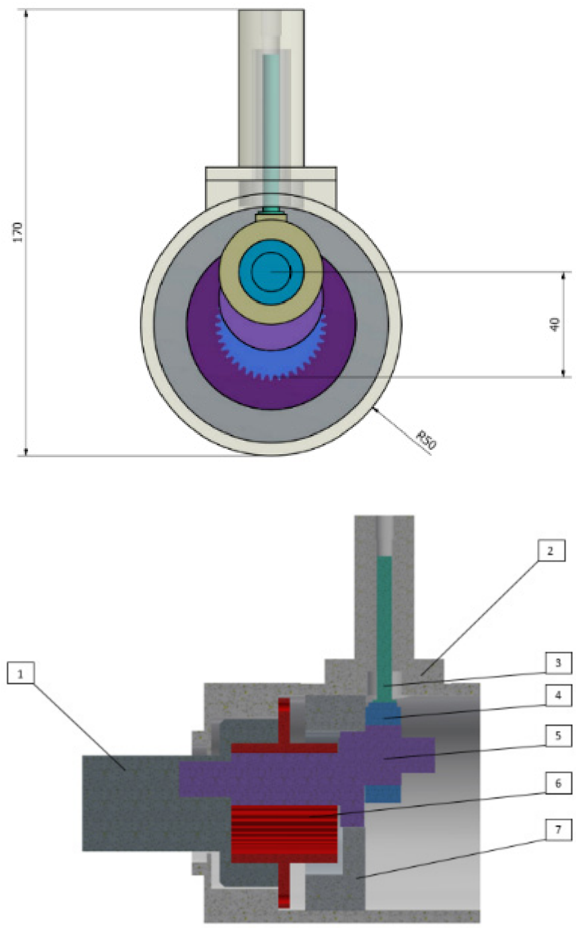

Fig. 4. Concept of the model of a pump based on the hypocycloidal drive:
1 – driving shaft, 2 – body, 3 – piston, 4 – ring mating with the piston, 5 – indirect shaft with a toothed wheel, 6 – inner tooth wheel, 7 – bearing support

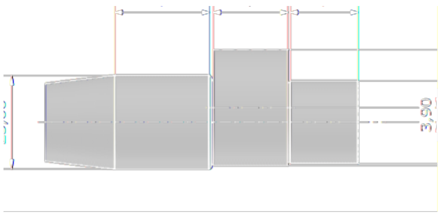

Fig. 5. Dual section pump by Continental

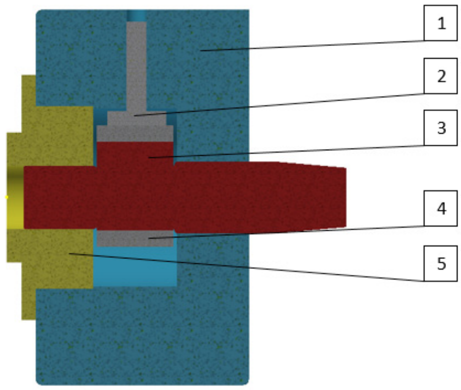

Fig. 7. Model of a pump allowing for the assumed simplifications
1 – body, 2 – piston, 3 – driving cam shaft, 4 – cam sleeve, 5 – body cover

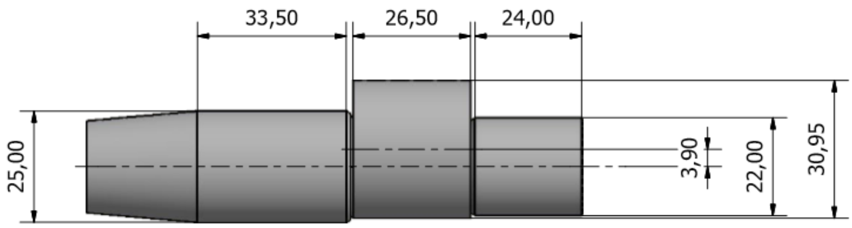

Fig. 6. Dimensions of the driving shaft with an eccentric cam

high-powered engines. The driving component of the reciprocal mechanism is a camshaft whose dimensions have been presented in Figure 6.

The tri-dimensional model of the pumps has been developed in the Autodesk Inventor Professional 2016 environment fitted with a dynamic simulation module. Allowing for the requirements of the simulation software, a simplification of the model was necessary maintaining its actual dimensions. The simplifications included the application of only one section composed of the piston and the base of the pusher with no guides, and the cylinder directly in the body, as shown in Figure 7.

In order to compare the parameters of the discussed types of drives, dynamic simulations of both solutions were carried out according to the assumptions presented in Table 2.

The value of the force acting on the piston is mainly generated by the fuel pressure that pushes the front part of the piston (of the diameter of 6.3 mm in both cases). Due to the pressure changing with the angle of the drive rotation, the value of the force acting on the shaft changes as well. The value of the fuel pressure in the pumping section depends on the working phase of the section of the pump. In both pump types we may distinguish the following cycles:

- filling cycle - the pumping section is filled with fuel supplied by the feeding pump (pressure of approx. 7 bar),
- pumping cycle - a very abrupt increase in the pressure takes place resulting from low

Table 2. The parameters and their values used in the dynamic simulations of the solution under analysis

Parameter	Value
Speed of the driving shaft	1500 rpm
Generated pressure	1500 bar
Feed pressure	8 bar
Section filling	100 %
Coefficient of friction of the slide pairs	μ=0.05
Efficiency of the gearset	η=0.98
Piston diameter	6.3 mm
Initial piston location	Top dead center
Maximum force acting on the piston	5300 N
Direction of the force	Parallel to the axis
Number of steps in the simulation	1000

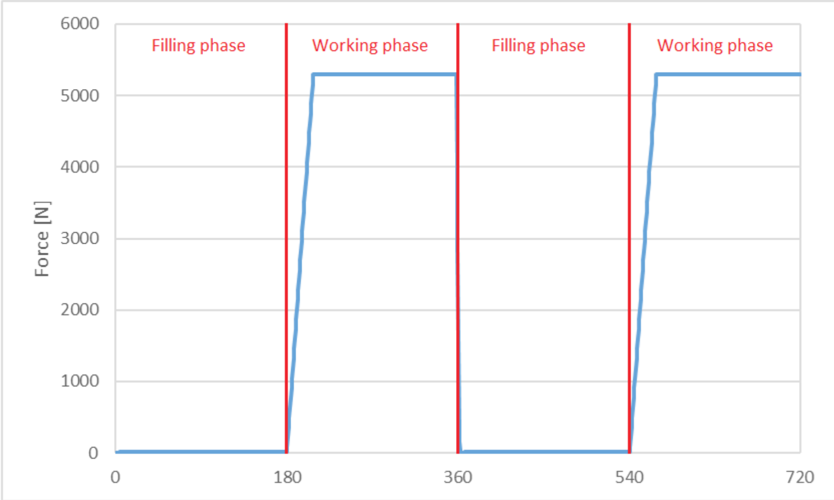

Fig. 8. Curves of the force acting on the piston allowing for the pump operating phases

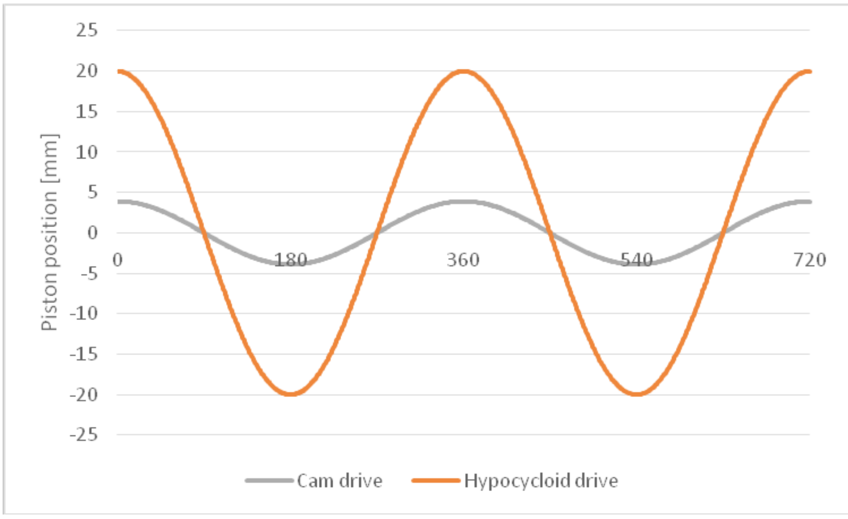

Fig. 9. Comparison of the piston position in reference to the angle of the shaft rotation in the cam-driven pumps and the pumps fitted with the hypocycloidal drive

compressibility of the fuel along with the fuel pumping with a constant pressure (opening valve opening pressure - 1500 bar)

The curve of the force acting on the piston, allowing for the pump working phases has been shown in Figure 8.

SIMULATION RESULTS

The basic parameter describing the motion is the change of the position in time. In the analysis of the piston position of both the pumps fitted with the hypocycloidal and the cam-driven pumps, the time can be substituted with angular position of the camshaft. This is possible owing to the cyclical nature of the pump operation. The results of the simulation presenting the change in the position of the piston against the Z axis for both drive systems has been presented in Figure 9.

Analyzing both curves, we may observe that they resemble a sine. Both the maximum and minimum values are obtained for the same angular positions of the driving shaft. In each case, the starting position of the piston was the top dead center. Yet, for each type of the drive it was at a different distance from the axis of the driving shaft. As we could see in the previous figure, the position of the piston at the same point of rotation of the driving shaft is different for each of the pumps. For the hypocycloidal drive, the piston displacement from the top dead center (0 °SA) to the bottom dead center (180 °SA) is 40 mm. For the conventional driving system, the piston displace-

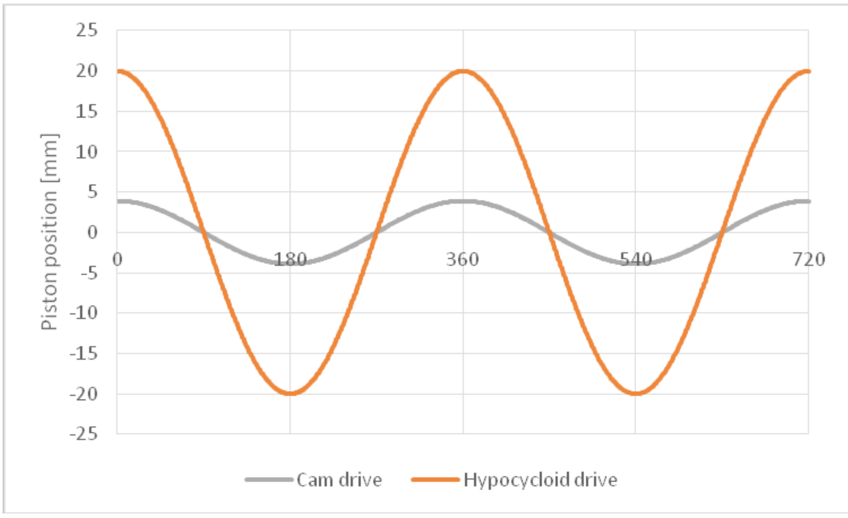

Fig. 10. Comparison of the piston velocity against the angle of the drive shaft rotation of the pumps fitted with the hypocycloidal drive and the cam-driven pumps

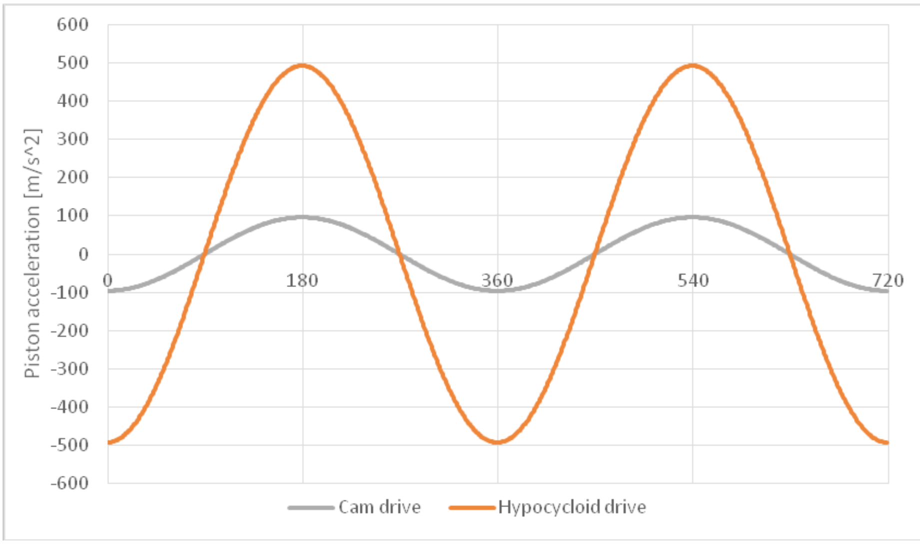

Fig. 11. Comparison of the piston acceleration in reference to the angle of rotation of the driving shafts of the pumps fitted with the hypocycloidal drive and the cam-driven pumps

ment is 7.8 mm. This means that each of the pump drives is characterized by different linear velocity of the piston. This was confirmed by the simulation, whose result has been shown in Figure 10.

Velocity is a vector value, hence we need to acknowledge that if it assumes negative values the sense of the vector is towards the bottom dead center and consequently, when assuming positive values – the top dead center. The initial value of the velocities for both types of drive equals zero, which results from the piston located at the top dead center at the start of the simulation. Additionally, the simulation allows the maximum piston velocities. Therefore, the maximum piston velocity for the hypocycloidal drive is 3.14 m/s i.e. it is several times higher compared to the piston velocity in the cam drive system (0.61 m/s).

Another parameter characterizing the motion of the piston that was obtained during the dynamic simulation is the acceleration. The second derivative of distance covered in time allows obtaining the change in the acceleration. In the analyzed case, this was the change of the acceleration as a function of angle of rotation of the driving shaft. The curves presenting the change in the piston acceleration for the pumps fitted with the hypocycloidal drive and the cam-driven pumps have been shown in Figure 11.

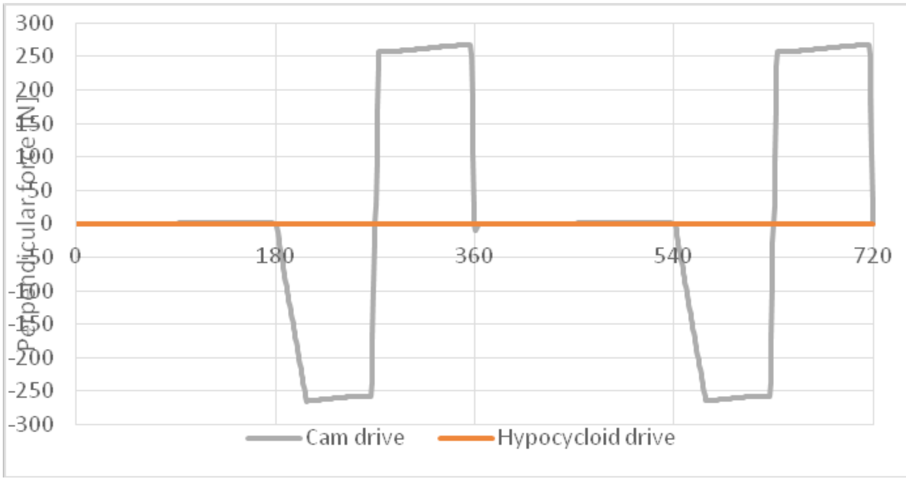

Fig. 12. Comparison of the force acting on the piston perpendicularly to its axis in the cam-driven pump and the pump fitted with the hypocycloidal drive

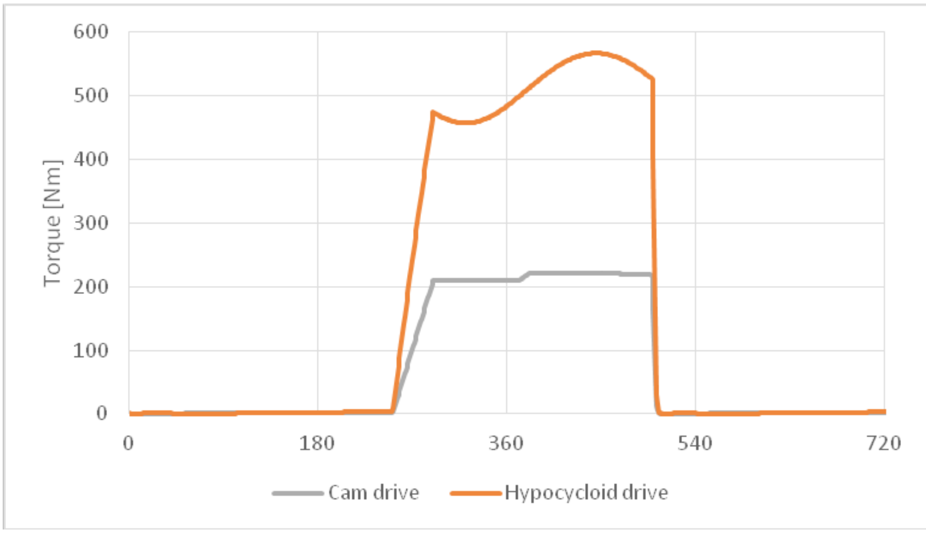

Fig. 13. Comparison of the torques on the driving shaft of the cam-driven pump and the pump fitted with the hypocycloidal drive

The maximum values of piston acceleration were determined using the obtained curves of this parameter. For the cam-driven pump, the maximum piston acceleration is 96.23 m/s^2, and for the pump based on the hypocycloidal drive, the acceleration is 493 m/s^2.

The dynamic simulation also allowed validating the correctness of the preliminary assumptions related to the occurrence of force perpendicular to the axis of the piston. Figure 12 presents the distribution of the radial force acting on the piston of the cam-driven pump and the pump fitted with the hypocycloidal drive.

In the curves, we can observe that the radial force occurs only in the cam-driven pump. The appearance of this force is related to the friction between he base of the piston and the cam lobe. This force assumes a maximum value of 267 N. A very disadvantageous phenomenon in terms of operation is the change of its sense that takes place at 270 ° S.A.

The last series of data (figure 13.) obtained thanks to the dynamic simulation presents the torque curve on the driving shaft. Comparing the two types of drives, the value of torque may be one of the key elements confirming the superiority of one system over the other.

The obtained torque results for both cases vary widely, particularly in terms of the differences in the values of the obtained torques. For the pump with the conventional drive, the torque obtained in the pumping phase is almost constant. It reaches a steady value of approx. 210 Nm for 180° S.A. to 270° SA, then it grows to 220 Nm after 270° S.A. and remains on this level until the end of the pumping phase. For the pump fitted with the hypocycloidal drive, the value of the torque in the pumping phase varies. At the start of the pumping, similarly to the cam-driven pump, we may observe an abrupt increment of torque related to the compressibility of fuel. Then the value of the torque decreases slightly, to grow again and reach its maximum for the position when the shaft has fully rotated. The maximum torque of the pump fitted with the hypocycloidal drive is 567. The value of the torque changes, most likely, due to the change of the distance of the piston fitting point against the axis of the driving shaft, which is characteristic of hypocycloidal drives.

CONCLUSIONS

The analyzed models of pumps using modern hypocycloidal and conventional cam drives have their pros and cons. A significant difference in the design of these systems is the piston displacement. For the pump fitted with the hypocycloidal drive, the piston displacement is much higher and amounts to 40 mm. For the cam-driven pump, it amounts to 7.8 mm. The proportional difference of the obtained values was also observed during the simulation of the piston displacement. Also, on the graph showing the piston velocities, we may observe that the values obtained by the pump fitted with the hypocycloidal drive are more than five times higher than the value obtained when a conventional pump system is used. Thanks to this knowledge, there will be no need to perform two separate simulations in order to determine the piston velocity, which will save much time. The same relation can be observed when we analyze the piston acceleration.

The performance of a dynamic simulation also confirmed the occurrence of a force perpendicular to the axis of the piston, resulting from the friction between the base of the piston and the cam lobe. This force does not occur in the pump fitted with the hypocycloidal drive. The results of the analysis show that further research works should focus on the factor influencing the value of this force. From the results of the performed analysis, we may infer that the value of this force depends on the value of the force acting on the piston (pressure) and the coefficient of friction. Given the differences in the lubricating properties of fuels, the value of this force (due to a change in the coefficient of friction) may change after refueling. In order for diesel engines to meet the environmental requirements, a further increase in the fuel injection pressure will be necessary. In the future, however, this may turn out very difficult because of the limitations regarding the material strength and the difficulty in predicting the radial force leading to beveling of the piston. The hypocycloidal drive has a big advantage in this respect. In this type of drive, the radial force does not occur at all, as confirmed in the simulation.

Due to higher piston displacement, the pump fitted with the hypocycloidal drive is characterized by the active volume of the pumping section of 4.99 cm^3, while the active volume of the pumping section of a cam-driven pump is more than five times lower and amounts to 0.97 cm^3. The above is a definite advantage of the pump based on the hypocycloidal drive, because it is possible to reduce the rotational speed or reduce the diameter of the piston to adapt the fuel flow. The first solution allows reducing the power consumption of the pump. The second option appears particularly advantageous because in this solution, the pressure force acting on the piston is reduced. Given the fuel flow in the pump fitted with the hypocycloidal drive, which is five times greater compared to the conventional system, a greater torque demand (but only 2.5 times) results in a situation when the collective power demand of such a design is lower.

There are many important aspects that have not been discussed in this work, yet, the obtained results of the simulation confirm the advantageous characteristics of the hypocycloidal drive and show its development potential. The currently available simulation tool allows determining a variety of parameters based on virtual models exclusively, which reduces cost and time consumption.

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