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IMPACT OF CHANGES IN FRICTION FACTOR ON THE LOADING OF THE PUMPING SECTION IN A HIGH PRESSURE INJECTION PUMP

The operational problems of currently manufactured high-pressure injection pumps are related to the use of fuels with inadequate lubricating properties. Even one-time use of fuel that does not meet the requirements assumed by the manufacturer may lead to irreversible changes in the structure of joints, which is tantamount to, among other things, change of the friction coefficient and attenuation in mobile nodes. This article presents and compares the results of dynamic simulation performed in the Autodesk Inventor Professional environment, in which the influence of changes of the friction coefficient value in the mobile nodes of pumps with cam drive and hypocycloid drive on loading of the pumping section was analyzed.

Key words: cam, common rail, high pressure fuel pump, friction

1. INTRODUCTION

The environmental requirements posed towards modern combustion engines are increasingly rigorous. To meet these requirements, it is necessary to apply many additional systems to improve control over the combustion process and purification of exhaust gases. Adapting modern compression-ignition engines to meet currently binding emissions standards is related to ensuring the proper fuel injection. Modern

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injection systems generate injection pressure reaching up to 3,000 bar, and the injection dose can be divided into up to 10 parts.

The generated pressure requires very durable fuel pumps, which must additionally work with a medium of low lubricity and viscosity. In the case of known pumps applied in passenger vehicles, pump elements may be loaded with a force exceeding 6,000 N when maximum injection pressures are generated.

Most high-pressure pumps currently being manufactured employ a cam mechanism. This mechanism, as shown on Fig. 1, consists of a cam installed eccentrically relative to the shaft's axis, on which a cam ring is mounted. The shaft's revolutions raise the follower and plunger, filling the chamber with fuel.

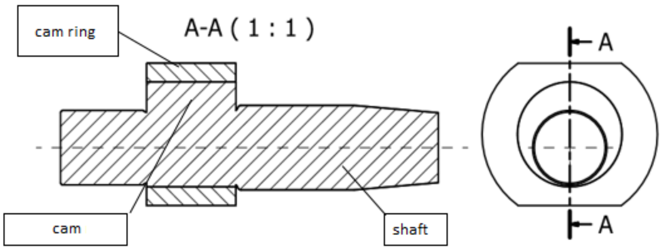

Fig. 1. Diagram of cam drive used in the bisectional common rail pump from Continental

Solutions of this type enjoy great popularity thanks to, above all, the simplicity of their design, which translates to low production costs.

For functional reasons however, the application of a cam mechanism has a fundamental flaw – as pressure is generated, the friction between the cam ring and the base of the plunger increases, leading to unfavorable contact stresses in the area of the cam and follower. This phenomenon has been illustrated using the example of a Bosch CP1 pump in Fig. 2. In the case of the latest designs, a series of solutions

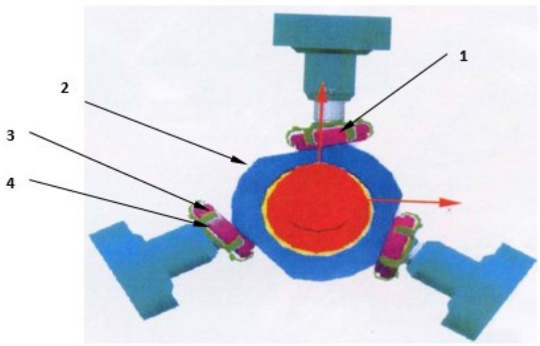

Fig. 2. Cross-section of Bosch CP1 pump in work position [Karpiuk et al. 2014]
1 – plunger base, 2 – cam, 3 – plate, 4 – cage

have been introduced to minimize the unfavorable effect of applying a cam drive, but due to the mechanics of the joint, lateral force cannot be eliminated and can only be distributed over a large surface area, and this is achieved e.g. by applying a guide roller inside which the plunger is freely installed.

Karpiuk et al. [2015] determined that such interaction leads to skewing of the plunger inside the cylinder, which accelerates the wear of cooperating surfaces of the precision friction pair. Meanwhile, Kałdoński [2008] states that the presence of a force perpendicular to the plunger's longitudinal axis makes it difficult to apply super-hard materials (e.g. engineering ceramics), which are characterized by high brittleness. The design of commonly used high-pressure injection pumps utilizing a cam drive means that their further development, oriented towards generating greater pressures, is made difficult, as increasing the pressure also increases the value of friction forces, thus accelerating tribological processes. In addition, fuels available around the world vary with regard to rheological properties [Mandler and Younshonis 2000].

Another type of drive mechanism that is free of this flaw is to hypocycloid transmission. The hypocycloid transmission is built of two wheels, of which the larger wheel has interior toothing and the smaller wheel has exterior toothing. Torque is applied to the smaller wheel, making it turn, however the larger wheel cannot rotate around its axis. The smaller wheel moves over the circumference of the larger wheel, and any point on the smaller wheel's radius moves in a curve called the hypocycloid. To create the working drive of the pump, a hypocycloid transmission with a gear radius ratio of R/r = 2 was applied. This selection of wheels makes it possible to achieve resultant linear motion, as presented in Figure 3.

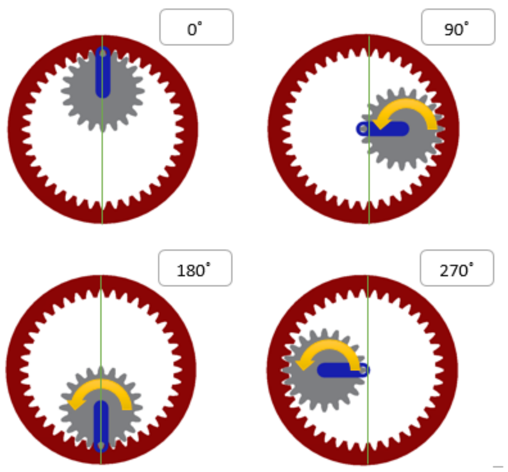

Fig. 3. Principle of operation of hypocycloid transmission

2. DESCRIPTION OF RESEARCH SUBJECT

The application of a hypocycloid mechanism to drive an injection pump has many benefits – it eliminates the lateral forces in the plunger-cylinder system described in the introduction of this article. High stroke of the working element (piston or plunger), equal to the reference diameter of a large gear, is another advantage. In the analyzed example, for a tooth module equal to 1 mm and number of teeth equal to 40, the stroke of the executive element is 40 mm. A characteristic feature of the hypocycloid drive applied in the model is a strictly defined gear ratio, which must be equal to exactly 2. What is more, for the mechanism to convert rotational motion into linear motion, it is necessary to apply an even number of teeth. The application of a hypocycloid mechanism furthermore makes it possible to increase the pump's resistance to difficult fuels – in the presented pump design, the pumping section is separated from the drive section, under the assumption that it will be lubricated using motor or transmission oil. Due to the piston's high stroke and the pumping section's length, it is possible to apply double labyrinth sealing along with a channel draining leaking fuel into injectors' overflow conduit. This will make it possible to achieve the minimum, acceptable leakage of injected fuel into the lubricating fluid. The absence of a return spring should also be counted as one of the pump's benefits. Pump solutions utilizing a cam system for each section have a spring that ensures that the follower is constantly pressed into the cam. Besides obviously increasing the weight of the system, the spring is an element that receives energy and gives it off in the form of heat to a small extent. In addition, considering a high number of cycles, springs are susceptible to damage caused by material fatigue.

The next part of this article contains a comparison of selected features of the pump model with hypocycloid drive and the pump model with classical cam drive. The pump model with hypocycloid drive is presented in Figure 4.

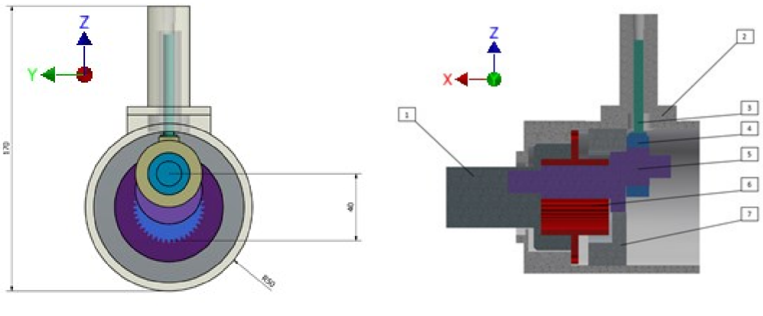

Fig. 4. Concept of pump model with hypocycloid drive
1 – drive shaft, 2 – body, 3 – plunger, 4 – ring cooperating with plunger, 5 – intermediate shaft with gear, 6 – gear with interior toothing, 7 – support on bearings

A pump from the Continental company (formerly Siemens VDO) was selected as the example of a high-pressure pump utilizing a cam drive. This pump is commonly applied in engines from the Volkswagen concern – most often in units with capacities of 1.6 and 2.0 dm^3. In comparison to the hypocycloid pump model, Continental's pump is characterized by a stroke of only 7.8 mm.

It consists of two pumping sections situated in opposition to one another. This solution, above all, makes it possible to reduce pressure pulsation in the system and to achieve the proper output required for high-power engines. The camshaft, whose dimensions are presented in Figure 5, is the element responsible for reciprocating motion.

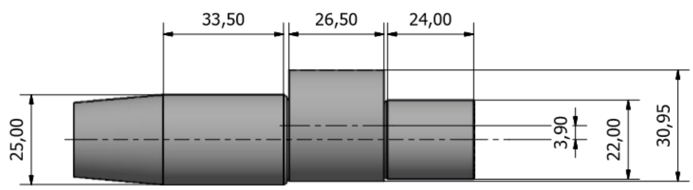

Fig. 5. Dimensions of drive shaft with eccentric cam

To conduct a simulation, three-dimensional models of both pump types were made. Considering the requirements of simulation software, it was necessary to simplify the pump model with the cam drive. Simplifications pertain to the application of only one section, consisting of the plunger and follower base, without guidance, as well as a cylinder made directly in the body, as presented in Figure 6.

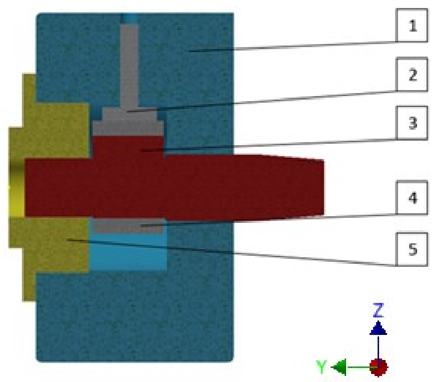

Fig. 6. Pump model accounting for the adopted simplifications
1 – body, 2 – plunger, 3 – drive camshaft, 4 – cam sleeve, 5 – body cover

The dynamic simulation module available in Autodesk Inventor Professional 2016 software makes it possible to change the parameters of applied mobile nodes. In the case under discussion, the goal of simulation was the learn the dependency between a change in the friction coefficient value in the discussed examples of high-pressure pumps and the load on the pumping section. To facilitate the comparison of two different systems, it was necessary to define the starting parameters applied in the dynamic simulation. Parameter types and values are presented in Table 2.

Table 2. Parameters and their values used in dynamic simulations of the analyzed design solutions

Parameter	Value
drive shaft revolutions	1500 rpm
pressure generated	1500 bar
supply pressure	8 bar
section filling	100 %
friction coefficient	from $\mu = 0.0$ to $\mu = 0.1$ (every 0.01)
efficiency of gear transmission	$\eta = 0.98$
plunger diameter	6.3 mm
initial plunger position	extreme upper position
maximum force acting on plunger	5300 N
direction of force	parallel to axis
number of simulation steps	500

The value of the force loading the plunger mainly originates from the fuel pressure acting on the plunger's face surface, the diameter of which is 6.3 mm in both cases. Due the change in pressure depending on the angle of the driveshaft's rotation, the value of the force loading the shaft also changes. The value of fuel pressure in the pumping section is dependent on the work phase in which the pump's section is found. In both pump types, the following cycles can be distinguished:

– filling phase – during which the pumping section is filled with fuel fed by the supply pump (pressure approx. 7 bar),

– pumping phase – during which very rapid pressure growth occurs due to the low compressibility of fuel and pumping of fuel at constant pressure (valve opening pressure 1,500 bar)

The progression of the force loading the plunger, accounting for the work phases of both pump types, is presented in Figure 7.

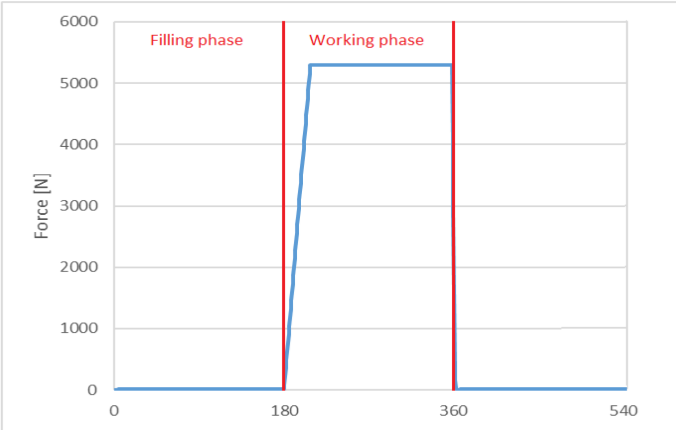

Fig. 7. Progression of force loading the plunger, accounting for pump work phases

3. RESULTS OF CONDUCTED SIMULATIONS

3.1. Pump with hypocycloid drive

In the case of a change in the friction coefficient in mobile nodes of the hypocycloid drive pump, obtained results were identical for all of its values. Due to the nature of the hypocycloid transmission's operation, the load of the pumping section on the X and Y axes is equal to zero. Figure 8 presents the distribution of force loading the base of the piston and the progression of curves loading the pumping section. It should be noted that the force loading the plunger acts on the Z axis of the described model. Components of the force loading the pumping section act in the directions of the X and Y axes. The pumping section is not loaded on the Z axis, since we do not account for closure of the pumping section by means of a head in the model under consideration. Similarly as in the case of the cam drive, the directions of force are consistent with those presented in Figure 4.

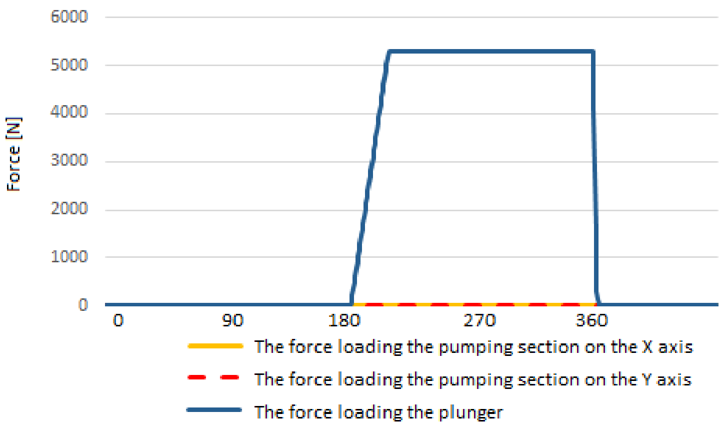

Fig. 8. Values of forces loading the pumping section in the pump with hypocycloid drive

3.2. Pump with cam drive

In the case of the pump with cam drive, simulations were conducted starting from a friction coefficient value $\mu = 0$, and then this value was increased by 0.01 up to a value of $\mu = 0.1$. Finally, 11 curves presenting the change in load values acting on the pumping section over the course of one drive shaft revolution were obtained. The directions of obtained forces loading the pumping section are consistent with the coordinate system in Figure 7.

a) b)

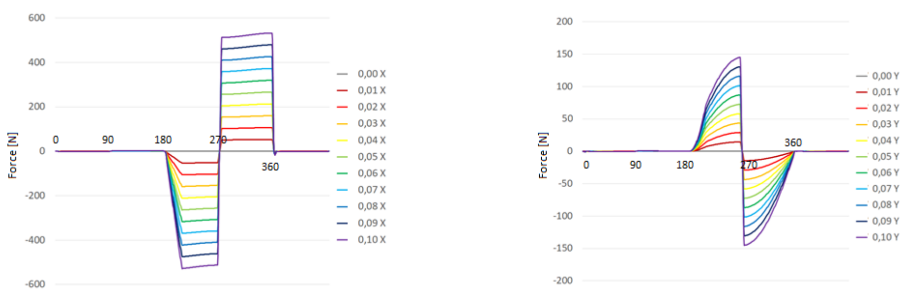

Fig. 9. Value of force loading the pumping section:
a) acting on X axis, b) acting on Y axis

4. CONCLUSIONS

Thanks to conducted simulations, it was observed that for the pump with cam drive, there is a linear dependency between the friction coefficient and values reached by forces loading the pumping section. Loads pressing the plunger to the cylinder's walls are only absent in the case where the friction coefficient is equal to zero. A characteristic feature of the present load is the very sudden change of its sense. This change occurs when the shaft performs a 270° rotation relative to its starting position. A change in the sense of the force's action may lead to very unfavorable impact loads, which may lead to the occurrence of fatigue damage.

Different simulation results were obtained in the case of the hypocycloid drive. Due to the "rolling" nature of the gear transmission's work, which is directly responsible for the conversion of rotational motion to reciprocating motion of the plunger, as change in the friction coefficient does not generate unfavorable loads in the pumping section.

The subject matter undertaken here, concerning determination of dependencies between the friction coefficient and load of the pumping section via simulation, is very current. Thanks to software packages such as Autodesk Inventor Professional, it is possible to accelerate the designing process and implement new products.

There are many important aspects that have not been touched upon in this article, however obtained simulation results confirm the existence of beneficial features of the hypocycloid drive and show its potential for development. Currently available simulation tools enable determination of many parameters by using only virtual models, saving time and money.

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OCENA ZMIANY WSPÓŁCZYNNIKA TARCIA W POMPACH WYSOKOCIŚNIENIOWYCH – PORÓWNANIE NAPĘDU HIPOCYKLOIDALNEGO Z KRZYWKOWYM

Streszczenie

Problemy eksploatacyjne dotyczące współcześnie produkowanych wysokociśnieniowych pomp wtryskowych związane są ze stosowaniem paliw o niedostatecznych właściwościach smarnych. Nawet jednorazowe zastosowanie paliwa niespełniającego założonych przez producenta wymagań może prowadzić do nieodwracalnych zmian w strukturze połączeń, co jest równoważne między innymi ze zmianą współczynnika tarcia i tłumienia w węzłach ruchowych. W artykule przedstawiono i porównano wyniki symulacji dynamicznej wykonanej w środowisku Autodesk Inventor Professional, w której przeanalizowano wpływ zmian wartości współczynnika tarcia w węzłach ruchowych pompy o napędzie krzywkowym oraz napędzie hipocykloidalnym na obciążenie sekcji tłoczącej.

Słowa kluczowe: common rail, napęd krzywkowy, pompa wysokociśnieniowa, tarcie