Strength analysis of the intermediate shaft of the hypocycloidal fuel pump

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Abstract. Diesel engines are currently characterized by very high injection pressure. The high pressure allows to achieve better parameters of the fuel atomization process, thus contributing to the improvement of the engine's ecological parameters. Achieving such a high pressure is possible thanks to the use of reciprocating positive displacement pumps, enabling the fuel pressure to be increased up to 3000 bar. A common feature of currently produced pumps is the drive of the pressing sections through the camshaft, which is lubricated using fuel. For this reason, cam-driven pumps are very sensitive to the quality of the fuel used, and one of its disadvantageous features is also the generation of lateral force on the piston of the section. For this reason, the authors of the article have developed a pump solution, using a hypocycloid drive, in which the disadvantages of traditional cam-driven pumps have been eliminated. The hypocycloid mechanism used has a separate lubrication system, and due to the generated rectilinear motion only the force parallel to the direction of the movement works on the piston. An important problem in the construction of a hypocycloid pump is the high loads of cooperating elements. The intermediate pump shaft is considered to be the most loaded element due to the complex state of stress. The article presents the methodology and results of simulation tests. Stress states have been determined for the position of the shaft in which it takes the highest load. As a result, it was recognized that the pump shaft was designed properly and despite the high force acting on the piston section, it is possible to ensure its long-term and reliable operation.

1. Introduction

Modern fuel pumps in compression-ignition engines must meet two basic criteria: fuel supply under possibly high pressure and possibly lowest power consumption, which directly contributes to the mechanical efficiency of the engine. In order to obtain the required indexes, the pump manufacturers use various solutions such as, inter alia, fuel rate control, reduced number of the pumping sections or reduction of the piston-cylinder gap. A significant common feature for the applied fuel pumps is the use of a cam mechanism for driving of the pumping sections (Fig. 1). A shaft (2) is fitted in the pump body with a cam placed in its eccentric (3). The rotating motion of the shaft causes the motion of the cam in the plane perpendicular to the shaft axis, moving the piston in the pumping section (1). The cyclical back and forth motion facilitates the intake and the fuel pumping strokes [2].
Figure 1. A transverse cross-section of a CP1 pump with a section drive shown.

A great advantage of the use of the cam mechanism is its simple design and, consequently, low costs of production. However, the practice in maintenance shows that the pumps are particularly sensitive to the quality of fuel that lubricates the cooperating elements within the drive system. Inappropriate lubrication of the cooperating elements leads to their seizure, which, consequently, can lead to the destruction of other elements of the injection system. Therefore, the pumps are not suitable for use in engines fueled with low quality fuels, including, as we call it, difficult fuels [4,5].

A significant increase in strength and resistance of a pump to low quality of fuel can be obtained by changing the drive to one that does not generate any lateral force on the piston and allows independent lubrication. The proposed solution of a pump driving the pumping section uses a hypocycloidal mechanism (Fig. 2). The mechanism includes a pair of geared wheels, including one small and one large wheel. The small wheel has external teeth and the large wheel has internal teeth. In order to comply with the assumptions, this has to be a pair of wheels, for which the smaller wheel has half the number of teeth of the large wheel. In this case, if the small wheel starts to run on the external circumference of the large wheel, a selected point on the wheel circumference will determine a straight line [3]. If a section piston is fitted at this point, it will move in a reciprocating motion and the force applied to it will only include a component parallel to the direction of the motion. It is noteworthy that, in this solution, the piston stroke will equal the diameter of the large geared wheel, which allows an obtainment of a high pump output despite its small size.

Figure 2. The principle of operation of a hypocycloidal mechanism
The presented characteristics were decisive of the use of this mechanism in a high-pressure fuel pump. Its efficiency and applicability have been confirmed in other designs, in which mechanisms converting the rotating motion into a reciprocating motion and vice versa were substituted. An example may include the Wisemann engine, in which the classical crosshead mechanism was eliminated and substituted with a hypocycloidal mechanism. The designers have proven that the use of this drive may help to provide good operating indexes as compared to a conventional mechanism and eliminating its defects. The structure has been validated in an engine of the power of 20kW and no other issues have been found within the pair of geared wheels.

The discussed pump design has been presented in Fig. 3. The pump body houses a shaft (1), to which torque is transferred. The intermediate pump shaft (2) is set eccentrically in such a manner that it is ¼ of the diameter of the geared wheel (3). Setting the shaft (2) into movement through the shaft (1) causes the shaft to roll on the circumference of the large wheel. The motion results in a change of the position of the connecting component (6) based on the support (5) in the vertical direction, to which the section piston is attached (4). The solution may also feature a pump with two or four pumping sections and a hypocycloidal mechanism located symmetrically on both sides of the pump body.

![Figure 3. Cross-section of a hypocycloidal pump](image)

2. The strength analysis of the intermediate shaft – assumptions

Due to complex stresses (twisting, bending) and maximum force load resulting from the accumulation of fuel, the intermediate shaft is a component of the pump that is most exposed to damage. Therefore, the author of this paper has focused on presenting the results related to this particular component. Loading of the pumping section results from the assumed maximum fuel pressure of 1600 bar. The force affecting the bottom (as calculated) and resulting from the piston diameter amounts to 5300N. The force curve and the position of the intermediate shaft also affect the torque curve on the shaft as presented in Fig. 4.
The analyses did not include the individual phenomena occurring on the contacts of the geared wheels, e.g. precise distribution of the contact stresses. The author only considered the very fact of transfer of torque by the gear units. The cooperation between the geared wheels of the shaft was reproduced by the module for modelling of the contact stresses available in the calculation software. The applied technique ensures only appropriate transfer of forces by the teeth in the direct contact. The readouts of the stresses can be burdened by errors. The calculation model was built with the use of 3D four-wall finite elements of the second order. The elements allow an obtainment of good accuracy and, at the same time, are suitable for filling of non-axisymmetric objects having irregular shapes and complex geometrical form. Loads can be applied in two ways, depending on what is deemed to be the action and response. The first technique involves blocking of the movement of the main shaft and application of an over-piston force in an appropriate angular position of the system. The other method involves blocking of the frame (to which the over-piston force is applied) and application of torque to the shaft. The author has decided to use the first method. Therefore, a force in the place of a piston foot was applied. In analyses it has been assumed that the coefficient of friction in all contact pairs will equal zero. Thus, forces are transferred entirely from the point of application to the point of response. This makes stresses generally higher. The procedure increases certainty of the analyses.

3. Measurement results
In order to obtain information concerning the minimum and the maximum stresses required for the initial assessment of fatigue strength, results for three positions of the main shaft (0°, 180° and 270°) were presented. The first analysis related to a situation, in which there is no play. This case is used as reference and facilitates the assessment of the effects of the play. The second case relates to a situation, in which play has minimum values (max. 0.1 mm) and in the third case it has maximum values (max. 0.2 mm). In subsequent figures (5,6,7) a profile of deformation has been presented with the Huber-Misses maximum reduced stresses marked. Due to the similar nature of the changes, a case was presented, for which no play was used. In table 1,2 and 3 results of the maximum stresses have been presented, which were taken into account for the assessment of the strength of the component.
Figure 5. Map of the Huber-Misses-Hencky stresses for the position of 180° (DMP), deformation scale of 500:1.

Figure 6. Map of the Huber-Misses-Hencky stresses for the position of 270° (geometrical pumping center), deformation scale of 100:1

Figure 7. Map of the Huber-Misses-Hencky stresses for the position of 0° (GMP), deformation scale of 500:1
Table 1. Stress concentration in case of absence of play

| Stress concentration area                                      | $\sigma(0^\circ)$ | $\sigma(180^\circ)$ | $\sigma_{A,1}$ | $\sigma_{M,1}$ | $\sigma_{SWT,1}$ |
|---------------------------------------------------------------|-------------------|---------------------|----------------|----------------|-----------------|
| A geared shaft – rounding of the shaft for change of the diameter (upper surface) | 0 MPa             | 450 MPa             | 225 MPa        | 225 MPa        | 318 MPa         |
| A geared shaft – rounding of the shaft for change of the diameter (bottom surface) | 0 MPa             | 275 MPa             | 135 MPa        | 135 MPa        | 190 MPa         |

Table 2. Stress concentration in case of minimum of play

| Stress concentration area                                      | $\sigma(0^\circ)$ | $\sigma(180^\circ)$ | $\sigma_{A,1}$ | $\sigma_{M,1}$ | $\sigma_{SWT,1}$ |
|---------------------------------------------------------------|-------------------|---------------------|----------------|----------------|-----------------|
| A geared shaft – rounding of the shaft for change of the diameter (upper surface) | 317 MPa           | -564 MPa            | 440 MPa        | -123 MPa       | 497 MPa         |
| A geared shaft – rounding of the shaft for change of the diameter (bottom surface) | -185 MPa          | 365 MPa             | 275 MPa        | 90 MPa         | 317 MPa         |

Table 3. Stress concentration in case of maximum of play

| Stress concentration area                                      | $\sigma(0^\circ)$ | $\sigma(180^\circ)$ | $\sigma_{A,1}$ | $\sigma_{M,1}$ | $\sigma_{SWT,1}$ |
|---------------------------------------------------------------|-------------------|---------------------|----------------|----------------|-----------------|
| A geared shaft – rounding of the shaft for change of the diameter (upper surface) | 0 MPa             | 734 MPa             | 367 MPa        | 367 MPa        | 519 MPa         |
| A geared shaft – rounding of the shaft for change of the diameter (bottom surface) | 0 MPa             | 380 MPa             | 190 MPa        | 190 MPa        | 268 MPa         |

The table below presents cycles reduced with the Smith-Watson-Topper method (recommended by [1]), in which:
\[ \sigma_{AZ} = \left( (\sigma_A + \sigma_M)^{1/2} \right)^2 \]  

where: \( \sigma_A \) – amplitude cycle stresses, \( \sigma_M \) – average stresses in a cycle. In this case it is possible to assess the fatigue strength with the use of single parameter fatigue characteristics, e.g. the Wholer’s chart or by comparing the stresses with the constants determining the permanent fatigue strength, e.g. for tensile strength (\( k_R \)). The tables include the results only for those subassemblies and their fragments, for which the stresses obtained relevant values in terms of strength. The assumed limit was >50MPa.

4. Summary

Based on the conducted strength analyses, it was possible to determine the substitute stresses with the use of the Smith-Watson-Topper method for all subassemblies of the tested object. The stresses constitute a basis for the selection of the construction materials. Since, for the evaluation of the influence of the multi-axial nature of the stresses upon the fatigue strength, the H-M-H hypothesis was applied and an assumption was made that a structure should operate within permanent fatigue strength values, the appropriately selected material has to comply with the following condition (\( \sigma_{SWT} < x \cdot k_R \))

\[
\begin{align*}
\bullet & \quad \sigma_{SWT} \text{ - Smith-Watson-Topper stress value} \\
\bullet & \quad x \text{ - safety coefficient (typically 0.9)} \\
\bullet & \quad k_R \text{ - value of permanent tensile strength}
\end{align*}
\]

For an idealized case of the cooperation of components without play, \( k_R \) should be at least 211 [MPa]. However, in a real structure, there is play that fundamentally changes the nature of the cooperation of the pump mechanisms. This, in turn, implies an increase in the stresses and change of their distribution. Therefore, a risk of fatigue damage will increase at a geometrical pace according to the following scenario: wear causes play, which accelerates wear. Therefore, for the assumption of the maximum play, \( k_R \) should be at least 483 [MPa]. The aspect of the geared shaft strength is additionally complicated by the fact that the stresses in transition between the round part of the shaft and the geared part of the shaft are high, \( k_R \geq 483 \text{MPa} \). It turned out that a change of the stresses between 0° and 180° is the most demanding one for the structure, i.e. when no torque affects the system. It is necessary to use alloy steel with increased strength and precise machining of the material in order to eliminate potential factors initiating the fatigue cracks (i.e. avoidance of damage in the top layer and low Ra).

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