Handling improvement of the M1 class vehicles by the way of hydraulic power steering adaptive characteristic optimization

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Abstract. In the paper, a work on handling improvement of the M1 class vehicle by means of optimization of the volume flow-pressure characteristic of the power steering pump with adaptive characteristic is described. A work on the simulation of the liquid flow through the electromagnetic valve of the power steering pump in initial and modified state is described. Results of bench testing of the optimized pump prototypes and subjective and objective road tests of the M1 vehicle with optimized prototype and initial one installed are presented.

1. Introduction

While creating an up-to-date chassis, systems with adaptive regulation of key parameters such as damping, stiffness, reinforcement coefficient as known, are used actively. The steering system operates the boost factor firstly. So we can say that the law of its change is the main characteristic of the power steering. Steering and stability at low and high speed of the vehicle depend on it. It is actual for all existing steering systems: hydraulic power steering (HPS), electro-hydraulic power steering (EHPS), electromechanical power steering (EPS). This paper describes the work on optimization of the law of change of the boost factor of the adaptive HPS system. The law of change of this coefficient and, consequently, of assistant torque usually linearly depends on the lateral acceleration for any vehicle, if other values do not change, see formula 1 [1].

\[ T_{assyst} = \frac{m_1 \times t}{i_{stkin} \times T_{res}} = C_f \times \frac{a_y}{T_{res}} \]  

(1)

Where \( T_{assyst} \) – assistant torque, or torque created by power steering of any type, \( T_{res} \) – resistance torque which is created around the wheel steering axis, \( i_{stkin} \) – steering system kinematic ratio, \( a_y \) – lateral acceleration which appears while cornering, \( m_1 \) – mass that comes to the front axle of the car, \( t \) – stabilization arm that depends on the front suspension kinematics of the certain vehicle.

As \( m_1, t \) and \( i_{stkin} \) are parameters that depend on certain vehicle design, we can name them stabilization factor \( C_f \) which depends on certain parameters of construction of the considered vehicle and appears as stabilization torque to the lateral acceleration unit and is measured in \( N \times m_f/m_s^2 \).

As it could be seen, the assistant torque depends mostly on the lateral acceleration which affects the vehicle. When the lateral acceleration equals or nears zero, it is necessary for the assistant torque to be minimal in some cases such as straight movement with high velocities and on the contrary maximal in case of parking with velocities near to zero.
The aim of this paper is to make a description of handling improvement of M1 class vehicles by means of optimization of the electromagnetic valve, which changes volume flow rate of the hydraulic power steering, with the help of the mathematical modeling of the gas dynamics of fluid flow inside the solenoid valve of the hydraulic power steering pump.

Vehicle handling is assessed by the driver through the feedback feeling of the road surface under various driving conditions. Torque on the steering wheel $T_{sw}$ is the main criterion of “steering feel”. This torque can be presented in formula

$$T_{sw} = \frac{T_{res}}{\text{kin}} \times T_{assyst}$$

(2)

Where $T_{res}$ – turn resistance torque around the wheel steering axis, $\text{kin}$ – steering system kinematic ratio, $T_{assyst}$ – assistant torque which was described mathematically earlier [1].

2. Results

In case of steering system has mechanisms which can affect the boost factor according to law (depending on the electric signals from the sensors such as vehicle velocity, angle, speed of steering wheel, torque on steering wheel), such system can be named as adaptive and the law of change of $T_{assyst}$ will depend on several parameters of vehicle motion. An example of the dependencies of change of the boost factor on steering speed and vehicle longitudinal speed is given in Figure 1.

![Figure 1. Boost factor dependence on external parameters.](image)

To choose the most favorable curve parameters, it is necessary to have a lot of road testing in different modes both at low and high speeds or to use special test benches like HiL benches with the help of which it is possible to have different tests with feedback described in paper [2].

In case of steering system with HPS, it is possible to adjust the boost factor by means of changing the fluid flow in the system. In our special case, the optimization of the HPS pump electromagnetic valve which can change the oil flow depending on the spool position was carried out. It happens due to change of the electric current which is supplied to the valve coil windings that makes the spool move...
inside the body (see sketch in Figure 2a). The optimal level of regulation can be reached by reaching the most correct geometry of the variable flow section (see Figure 2b).

![Figure 2 a. Scheme of the electromagnetic valve spool movement.](image)

![Figure 2 b. Cross-section of the electromagnetic valve.](image)

As a part of optimization, calculation works by finite element method were carried out. The aim of these works was to simulate the work fluid outflow through the flow section of the HPS pump for various geometry options of the work line in two conditions: valve is conditionally closed (conditionally because in closed condition it must provide work fluid minimum volume flow) and also valve is opened, that corresponds to the power steering liquid maximum volume flow.

To begin with, a simulation of the liquid outflow in initial section geometry in two modes of the valve (closed and opened) was done. As initial parameters the parameters of power steering fluid, pressure at the valve entrance of 1 MPa and volume flow of the liquid in the valve exit gotten experimentally – 7.5 l/min for the closed valve and 15.8 l/min for the opened valve were set.

For the grid model of the closed valve (Fig. 3 a) the grid volume consisted of 8.5 mln. elements and for the opened one (Fig. 3 b) – 12 mln. elements with size of the main element 0.5 mm, thickening in the valve zone – 0.1 mm with the total thickness of 8 layers of prismatic sublayer – 0.12 mm.
Figure 3. Grid model of the closed valve (left, a) and opened valve (right, b).

Pressure fields inside the valve in the closed (Fig. 4 a) and opened (Fig. 4 b) conditions were obtained.

Figure 4 a, b. Result of calculations for the initial valve geometry in closed (left, a) and opened (right, b) conditions.

Pressure-flow graphs were plotted based on calculation results for the opened and closed valve conditions shown in Figures 5 a and b respectively.
Figure 5 a, b. Graph of full pressure loss for the opened valve (left, a) and closed valve (right, b).

Based on calculation results and initial conditions, the form of the intervalve geometry was changed according to Figure 6 b.

Figure 6 a, b. Initial intervalve geometry (left, a) and changed (right, b).

Comparative simulation was carried out. Velocity contours in the intervavle space were obtained as a result of this simulation (see Figure 7 a) for changed space and Figure 7 b for initial space for the open valve state.
Figure 7 a, b. Velocity contours for changed (left, a) and initial (right, b) intervalve spaces.

Analogical simulation was carried out also for the closed valve, see Figure 8 a, b.

Figure 8 a, b. Velocity contours for the changed (left, a) and initial (right, b) intervalve spaces.

Pressure-flow characteristic change as a result of intervalve section optimization is submitted below.
Figure 9. Graph of full pressure loss for the opened and closed valve of optimized and initial geometry.

As a result of the intervalve space changes we managed to change pressure-flow characteristic of HPS significantly in the closed valve mode (by 27% or down to 5.5 l/min) and also in open valve mode up to 19.5 l/min (by 24%).

But for further optimization and technological feasibility improvement, the second optimized intervalve space model was prepared (see Figure 10).
Figure 10. Second optimized geometry of intervalve space (the spool is shown in red).

For this one, calculation works were also held, results are shown below. Velocity contours comparison with the opened valve see in Figure 11 a, b.

Figure 11. Velocity contours for changed (left, a) and initial (right, b) intervalve spaces. Pressure-flow characteristic change as a result of intervalve section optimization is shown below.
As a result of the second optimization we managed to achieve decrease of the minimum volumetric flow of the closed valve down to 3.5 l/min, while with the opened valve the volumetric flow practically did not change.

The orifice of the electromagnetic valve was designed and manufactured (see Fig. 13) according to the second optimized model, which was installed into the electromagnetic valve of the HPS prototype pump.

Figure 12. Full pressure loss graph for opened and closed valve of first and second geometry optimization.
HPS prototype pumps were subjected to comparison bench testing with the initial design pumps for validation. As a result of this testing the averaged characteristics were received which are shown in Figures 14 and 15. Valve characteristic $Q(I)$ which is shown in figure 14 characterizes electromagnetic valve operation and represents volumetric flow measured at the valve outlet over the current supplied to the valve windings in the range of 0 to 1 A.

![Flow section geometry of the electromagnetic valve.](image)

**Figure 13.** Flow section geometry of the electromagnetic valve.

![Valve characteristics of initial and optimized HPS pumps.](image)

**Figure 14.** Valve characteristics of initial and optimized HPS pumps.

Dependence of the average volumetric flow values over three pump samples vs pump rotor rpm $Q(n)$ is shown in Figure 15. The characteristic is taken for current values of the closed valve (0 A) and opened valve (1 A).
Change of the volumetric flow for the closed valve condition (0 A) is 25% and for the opened valve is about 10% as you can see.

The prototype of the power steering pump with optimized solenoid valve was installed on the M1 class vehicle prototype for further handling subjective and objective evaluation. During comparative road test, parking maneuvers at low speeds were carried out, and handling and stability of the car at high (more than 100 km/h) speeds were assessed for the two states: with initial and optimized HPS pump. The results of these tests are shown in Figure 16 appearing as circle diagram of the subjective assessments of handling properties by three test drivers. The diagram shows that the optimized geometry of the orifice of the electro-magnetic valve of the power steering pump allows improvement the subjective assessment indicators by 0.5 - 0.7 points.

Figure 15. Flow characteristic of HPS pumps.

![Graph showing averaged volumetric flow Q(n) of HPS pumps in initial and optimized condition.](image)
Data of objective road tests with rotation of the steering wheel (SW) on the standstill car, as the most severe duty for the power steering, is shown in Figure 17. Graph HPS No.1 corresponds to the initial pump condition and HPS No.2 – to the optimized one.
Figure 17. Graphs of SW torque while turning SW from side to side at standstill car.

Decrease of the steering torque at about 1 N\(\cdot\)m is evident.

Graphs of the dependence of the torque applied to the steering wheel on the steering wheel angle is shown in Figure 18 below for driving along a circular path with a radius of 12 meters at a speed of 10 km / h in left and right turns. Similarly to the previous ones, graph HPS No.1 corresponds to the initial pump condition and HPS No.2 to the optimized one.

Figure 18. Graphs of steering torque dependence on SW angle in the turn with radius of 12 meters.

The reduction of torque required to turn the steering wheel is also noticeable in this case.
3. Conclusion

Calculated optimization of the intervalve space of the electro-magnetic valve of the power steering pump was carried out. Objective bench tests were carried out, which showed a decrease in the minimum volumetric flow rate by 25%, and an increase in the maximum volumetric flow rate by 10%.

Road tests were carried out, which showed that the optimization of the electromagnetic valve of the power steering pump allows reducing the torque required to turn the SW by 1 N·m, and increasing the subjective assessment of handling and stability by 0.5 - 0.7 points.

References

[1] Harrer M, Pfeffer P 2017 Steering Handbook 44 p
[2] Potashov I S and Bokarev A I 2019 Research method for adaptive algorithms for changing the steering effort using software and hardware modeling technologies HILS Proceedings of the International Automotive Scientific Forum IASF-2019