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NUMERICAL SIMULATION STUDY OF PARALLEL HYDRAULIC HYBRID SYSTEM FOR A DELIVERY VAN

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Abstract: In this paper an analysis of the hydraulic hybrid system with parallel architecture is carried out. Based on the different delivery van driving regimes a calculation of the van hydraulic hybrid system is conducted and standard hydraulic components are chosen. After that a numerical model of the parallel hydraulic hybrid system is developed using the Matlab/Simulink software. Using a developed numerical model, simulations for breaking and starting modes of the vehicle are performed. Simulation results have proven the real behaviour of the developed numerical model.

Keywords: Delivery van, Parallel hydraulic hybrid system, Numerical simulation,

1 INTRODUCTION

The idea of a hybrid vehicle is the result of the need to increase the efficiency of a conventional vehicle. This creates a vehicle with primary and secondary systems. The primary system is used to drive the vehicle and this is always the internal combustion engine (IC engine). The task of the secondary system is to produce, store and reuse energy which would be irrevocably lost in the environment. Losses in conventional vehicles such as losses in the IC engine, power transmission, as well as losses due to air resistance and rolling are inevitable. However, the energy that is lost during braking can be stored and reused, thus increasing the overall efficiency of the vehicle. This principle of the braking energy utilisation is called regenerative braking [1].

Depending on the design of the secondary system, hybrids are divided into electrical, hydraulic and mechanical hybrids. Hybrid electric vehicles are the most known hybrid vehicles because of the application of the electric hybrid technology in passenger cars. The main components of the hybrid electric vehicle are the generator, electric motor and electric battery. The biggest advantage of the hybrid electric vehicle is the ability to store a lot of energy in the compact and weightless electric battery. The main disadvantage of the hybrid electric vehicle is the low amount of stored energy density in the electric battery; therefore, the hybrid electric vehicle has to use an extra gearbox. Consequently, the hybrid electric technology is used in weightless passenger cars, while it is not used in commercial vehicles, buses, trucks and other heavy vehicles. In the aforementioned vehicles hydraulic hybrid technology is usually used [2].

The main components of hydraulic hybrid systems are the hydraulic pump, hydraulic motor and hydraulic accumulator, which are equivalent to the generator, electric motor and electric battery in electric hybrids. The main advantage of hydraulic hybrids compared to electric hybrids is the greater amount of stored energy density which is up to five times higher [3]. Another advantage of the hydraulic hybrid is great efficiency in cases of frequent braking and the starting of vehicles, e.g. garbage trucks and delivery vans. Taking into account the mentioned advantages of the hydraulic hybrid system, in this paper we will analyse the hydraulic hybrid system with parallel architecture for a delivery van.

2 PARALLEL HYDRAULIC HYBRID SYSTEM

The development of hybrid vehicles began in 1898 when Ferdinand Porsche introduced the first hybrid electric vehicle "Locher-Porsche Mixte Hybrida". This vehicle used an IC engine and two electric motors on the front wheels. When Henry Ford introduced an assembly line in car manufacturing, which resulted in his cheap car "Model T", hybrid vehicles no longer produced. After the first and second oil crisis, the hybrid vehicle became interesting again. In 1997, the modern hybrid electric car was presented by Toyota and Audi with their models “Prius” and “Duo”, respectively [4].

The development of the hydraulic hybrid began in the1980s. Hydraulic hybrid technology was originally developed by Volvo Flygmotor. Volvo developed a hydraulic hybrid system which was installed on buses in Stockholm. The reduction of the fuel consumption was between 16% and 25%. The first parallel hydraulic hybrid was "Cumolu Brake Drive" of the Parker Hannifin Corporation. In 1991, the Parker Hannifin Corporation also developed the first serial hydraulic hybrid. Nowadays many companies and university teams work on the hydraulic hybrid systems development such as Eaton, Parker, Bosch-Rexroth, Artemis, and others.

The basic division of the hydraulic hybrid is into serial and parallel hybrids. Typical serial hybrids are mounted on vehicles that already have a hydrostatic transmission, such as trucks or vehicles, whose driving mode is made up of a lot of stopping and starting. The serial hydraulic hybrid vehicle (Fig. 1) is composed of a hydraulic pump which is power operated by an IC engine, hydraulic motor which drives vehicle wheel and low and high pressure accumulators.

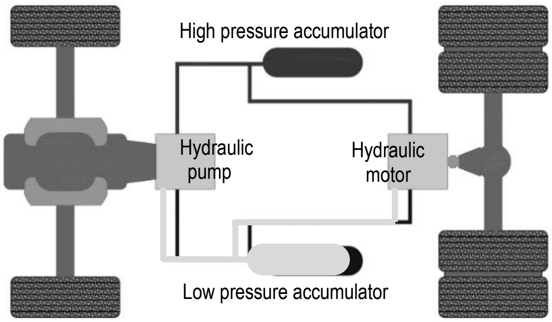

Figure 1 Serial hydraulic hybrid vehicle

The lack of a serial hybrid is the inability to achieve high-speeds, making it ideal for use on heavy vehicles, but not on vehicles in intercity traffic. For such a vehicle it is more convenient to use a parallel hydraulic hybrid system, because it allows the storage of energy during braking, but when it needs to achieve higher speeds, it uses the primary system. The parallel hydraulic hybrid vehicle is composed of a hydraulic pump which can work as a hydraulic motor, low and high pressure accumulators and a gearbox that is used for power take-off from the drive shaft (Fig. 2).

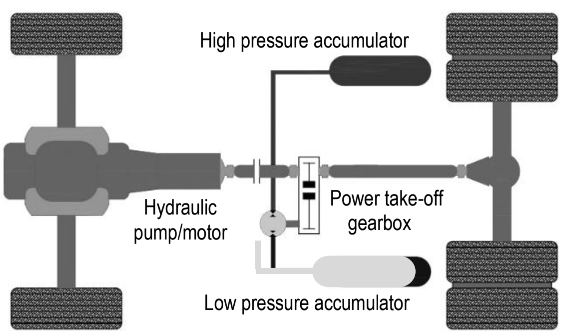

Figure 2 Parallel hydraulic hybrid vehicle

When the vehicle starts to brake, the hydraulic pump connected to the drive wheels via the drive shaft and the gearbox establishes the flow of the working fluid from the low to high pressure accumulator. The fluid that enters into the hydraulic accumulator must overcome the gas pressure, which is currently acting in the high pressure accumulator. The compressed gas creates resistance to the flow of the fluid in the accumulator, which creates the effect of braking on the wheels and slows down the vehicle. Thus, the kinetic energy of the vehicle is converted into energy of compressed gas in the accumulator.

When the vehicle begins to accelerate, the working fluid flows from the high to low pressure accumulator. Now the pump is acting as a hydraulic motor. When the vehicle consumes all the stored energy, the drive from the secondary system switches to the primary system, i.e. IC engine. At higher speeds (over 60 km/h) a vehicle can also store energy in the accumulator in short-term braking, although it does not stop. Accumulated energy, can later be used to accelerate the vehicle (e.g. when shifting gears) to help the IC engine, which also reduces fuel consumption.

3 CALCULATION OF A PARALLEL HYDRAULIC HYBRID SYSTEM FOR A DELIVERY VAN

Calculation of the parallel hydraulic hybrid system is carried out for the delivery van "Volkswagen Crafter TDI400" [5]. The vehicle has a 120 kW diesel engine, 6-speed manual gearbox, and a rear wheel drive. The maximum value of the payload plus vehicle weight is 3,500 kg. To design a parallel hydraulic hybrid system, the gearbox, drive shaft and differential gearbox have to be unchangeable, and new parts have to be added, such as the hydraulic pump/motor, low and high pressure accumulators, power take-off gearbox, and a control unit. Therefore, it is necessary to carry out a calculation of the main hydraulic part of the hybrid system, hydraulic pump/motor, high pressure accumulator, and low pressure accumulator.

Calculation has been carried out for two regimes of driving which are defined by the inclination of the road, vehicle driving speed and percentage of vehicle payload, as shown in Tab. 1.

Table 1 Driving regimes of delivery van

Driving regime	1	2
Inclination of road (%)	25	0
Driving speed (km/h)	12.7	120
Vehicle payload (%)	100	0

The first step in the calculation is to estimate all resistances that occur when the delivery van is driving: rolling friction force, air resistance force, resistance acceleration force, and inclination force. The sum of these forces results with a drag force F_{Vd}, which the vehicle must overcome. By knowing the value of the drag force and dimensions of the wheel (235/65R16), we are able to calculate the maximum torque on the vehicle wheels T_{Vmax}. By knowing the wheel dimensions and the vehicle driving speed, we can also calculate the rotational speed of the wheels n_{V}. The obtained results for the defined driving regimes are shown in Tab. 2.

Table 2 Calculated results for two driving regimes

Driving regime	F_{Vd} (N)	T_{Vmax} (N·m)	n_{V} (s^{-1})
1	13 934.9	7803.5	1.01
2	2583.5	1444.6	9.49

The torque on the hydraulic pump/motor shaft T_{shf} is the maximum torque on vehicle wheels divided by the sum of the vehicle differential gearbox ratio i_{d} = 3,923 and the power take-off gearbox ratio i_{pto} = 2. The chosen value of the take-off gearbox ratio is based on value of gearboxes ratio of similar hybrid vehicles.

Assuming the pressure drop Δp = 400 bar and the hydro-mechanical efficiency η_{hm} = 0.98, it follows that the displacement of the hydraulic pump/motor V_1 = 159.4 cm^3. According to the obtained value for displacement, the axial piston hydraulic pump/motor Bosch Rexroth A6VM has been chosen with characteristics shown in Tab. 3.

The calculation of the high pressure accumulator is based on the ability to absorb sufficient kinetic energy during the regenerative braking. It is assumed that the high pressure accumulator will be fully charged during the vehicle braking from v_2 = 70 km/h to v_1 = 10 km/h. When the vehicle decelerates to 10 km/h, the main braking system stops the vehicle. By knowing the regime of braking and the mass of vehicle, it follows that the maximum kinetic energy which can be absorbed during the braking E_B = 648 kJ.

Table 3 Characteristic of Bosch Rexroth A6VM hydraulic pump/motor

Max. displacement	V_{max}	171.8	cm^3
Max. speed	n_{max}	5750	min^{-1}
Max. flow	Q_{max}	533	l/min
Max. pressure	Δp	450	bar
Max. torque	T_{max}	1230	N·m

However, due to losses in the hydraulic system, 60 % of the whole kinetic energy can be stored in the high pressure accumulator [3], therefore E_B = 389 kJ. The high pressure accumulator with a nominal volume V_0 = 50 l and the minimum operating pressure of p_1 = 200 bar has been chosen as a preliminary. Parameter p_0 is the pre-charge pressure which is 10 % lower than the minimum operating pressure p_1. The maximum operating pressure was previously chosen and amounts to p_2 = 400 bar.

The process in the accumulator has been observed as a fast adiabatic process without heat transfer between a thermodynamic system and its surrounding. This can be explained with a value less than four minutes for charge and discharge of the accumulator. The maximum volume of working fluid, which can be charged in the high pressure accumulator, was calculated using the Eq. (1), whereby V_{oil} = 26 l.

$$V_{oil}=V_0\left(\frac{p_0}{p_1}\right)^{\frac{1}{\kappa}}\left[1-\left(\frac{p_0}{p_2}\right)^{\frac{1}{\kappa}}\right] \tag{1}$$

The maximum stored energy in the high pressure accumulator was calculated using the Eq. (2), whereby E_A = 392 kJ.

$$E_A=p_2\left(V_0-V_{oil}\right)-p_0V_0 \tag{2}$$

The number of high pressure accumulators were calculated in such a way that the maximum kinetic energy attained during the vehicle braking E_B divided by the maximum stored energy in the high pressure accumulator E_A, which results in 0.99 accumulators. Therefore Bosch Rexroth HAB 50-414 bladder type accumulator with the nominal volume of 50 l [6] was chosen. The low pressure accumulator was not calculated because it was chosen according to the minimum operating pressure of the high pressure accumulator. The Bosch Rexroth HAB 50-207 bladder type accumulator was selected.

4 MATHEMATICAL MODELLING OF A PARALLEL HYDRAULIC HYBRID VEHICLE

The mathematical model is the fundamental model for describing the behavior of a physical system. It can be defined, for example, as a set of differential equations which describe the connection between the physical values in the observing system.

The mathematical model of the parallel hydraulic hybrid system has been analyzed as a mathematical model of the hydraulic pump/motor and the high pressure accumulator. The mathematical model of the hydraulic pump and hydraulic motor is described with the torque balance Eq. (3) for the hydraulic pump and the Eq. (4) for the hydraulic motor.

$$J^P\frac{d^2\varphi^P}{dt}+T_f^{HP}\left(\frac{d\rho^P}{dt}\right)=T_{shf}^P-\eta_{vol}^PT_{th}^P \tag{3}$$

$$J^M\frac{d^2\varphi^M}{dt}+T_f^{HM}\left(\frac{d\varphi^M}{dt}\right)=T_{shf}^M+\frac{1}{\eta_{hm}^M}T_{th}^M \tag{4}$$

J is the moment of inertia of the pump/motor, T_{shf} is the torque of the pump/motor shaft, T_f is the frictional torque of the pump/motor, T_{th} is the theoretical torque of the pump/motor, η_{vol} is the volumetric efficiency of the pump/motor and φ is the rotational angle of the pump/motor shaft [7]. The method for modelling the frictional torque of the pump/motor is to model friction as a function of velocity, which is referred to as the Stribeck curve. The three characteristic parts of this equation are the viscous friction T_v, Coulomb friction T_{c0} and static friction with the parameters T_{s0} and c_s, known as the Stribeck velocity (Eq. (5)).

$$T_f\left(\frac{d\varphi}{dt}\right)=T_v\frac{d\varphi}{dt}+\text{sign}\left(\frac{d\varphi}{dt}\right)\left[T_{c0}+T_{s0}\,exp\left(-\frac{\left|\frac{d\varphi}{dt}\right|}{c_s}\right)\right] \tag{5}$$

The mathematical model of the high pressure accumulator is defined with the pressure, temperature and volume of the working fluid in the accumulator. Mathematical equations, which describe the mentioned parameters, are the ideal gas law and the real gas law. The ideal gas law is a good approximation of the behavior of many gases under many conditions. Consequently, the ideal gas law has been used for the mathematical model of the high pressure accumulator. Moreover, the process in the accumulator has been observed as a fast adiabatic process. This can be explained with a value less than four minutes for charge and discharge of the accumulator. The pressure and temperature alterations in the high pressure accumulator are described by using the following Eqs. (6) and (7), where θ_0 is the temperature of the surrounding.

$$\frac{\mathrm{d}p}{\mathrm{d}t} = p_0 \left(\frac{V_0}{\frac{\mathrm{d}V}{\mathrm{d}t}} \right)^{\kappa} \tag{6}$$

$$\frac{\mathrm{d}\theta}{\mathrm{d}t} = \theta_0 \left(\frac{p_0}{\frac{\mathrm{d}p}{\mathrm{d}t}} \right)^{\frac{1-\kappa}{\kappa}} \tag{7}$$

If the real gas law is used for the mathematical model, generally the Beattie-Brigdman equation or the Benedict-Webb-Rubin equation can be applied [8]. The disadvantages of using the mentioned equations for the mathematical model of the accumulator are their complexity and needs for the experimental measurement on the real hydraulic accumulator for determining all parameters.

Knowing the pressure dynamic of the high pressure accumulator the state of charge (*SOC*) of the accumulator can be defined. The state of charge of the high pressure accumulator is defined as the ratio of the current amount and the maximum amount of accumulator energy. If the amount of accumulator energy is described by the pressure of gas in accumulator, Eq. (8) can be used [9].

$$SOC = \frac{p_2 - \frac{\mathrm{d}p}{\mathrm{d}t}}{p_2 - p_0} \tag{8}$$

The state of charge is important for the hydraulic hybrid vehicle system control. This value defines when the high pressure accumulator is able to give energy to the hybrid vehicle and when it is necessary to start charging the high pressure accumulator during vehicle breaking.

5 NUMERICAL MODELLING OF A PARALLEL HYDRAULIC HYBRID VEHICLE

Based on the mathematical model, the numerical model can be developed using certain software. For numerical modelling, two principles can be used, the first is to use a very general description to represent a system (white-box model) and the second shows the use of a particular characterization for certain classes of systems (black-box model). A natural way to develop a numerical model is white-box modelling. It is based on the previous mathematical model, i.e. differential equations, algebraic equations, logical relationships and similar types of equations [10]. White-box modelling has been applied for numerical modelling of the parallel hydraulic hybrid system, using the Matlab/Simulink software. Two numerical models were carried out: the numerical model for the vehicle braking mode, and the numerical model for the vehicle starting mode.

The numerical model of the parallel hydraulic hybrid system for the vehicle breaking mode is composed of the numerical model of the hydraulic pump and the numerical model of the high pressure accumulator. The numerical model of the accumulator is based on the previously described mathematical model. On the other hand, the numerical model of the hydraulic pump did not use the mentioned differential equation, because the rotational speed of the pump is well known, however, the standard equation for the calculation of pump flow and torque was used [11]. Therefore the input value in the numerical model is the rotational speed of the hydraulic pump, i.e. rotational speed of the vehicle wheels.

The numerical model of the parallel hydraulic hybrid system for vehicle starting mode is shown in Fig. 3. Also this model is composed of the numerical model of the hydraulic motor and the numerical model of the high pressure accumulator. The numerical model of the hydraulic motor and the accumulator are based on the previously described mathematical model. The Coulomb friction and static friction were neglected in the numerical model of the hydraulic motor, only the viscosity friction was taken into account. The value of the viscosity friction torque was chosen from similar research [10].

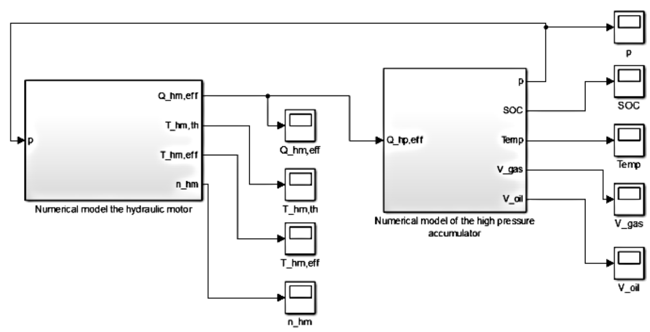

Figure 3 The numerical model of the parallel hydraulic hybrid system for vehicle starting mode

6 SIMULATION OF DEVELOPED NUMERICAL MODEL OF PARALLEL HYDRAULIC HYBRID SYSTEM

Simulations were conducted for the developed numerical model of the parallel hydraulic hybrid system. The aim of simulation results is to prove the accuracy of the developed numerical model. Simulations were carried out for the numerical model of the vehicle braking mode and the vehicle starting mode. Ten seconds of simulation duration were chosen for all simulations.

The simulation results which will be observed from the numerical models are pressure, *SOC*, volume and temperature of the working fluid and gas into the high pressure accumulator. Also, the torque and rotational speed of the hydraulic motor will be observed.

6.1 Simulation of a parallel hybrid system for a vehicle braking mode

The rotational speed of the hydraulic pump was calculated as linear decrease of the rotational speed from 44 s^{-1} to 6 s^{-1} during ten seconds, based on the previous assumption that the high pressure accumulator will be fully charged during the vehicle braking from 70 km/h to 10 km/h. Values of the pump rotational speed are the input into the numerical model of the hydraulic pump. The output result from the numerical model of the pump is the pump flow which decreases from 432 l/min to 60 l/min as a result of the pump rotational speed decrease, as shown in Fig. 4. The decrease of the vehicle speed during the breaking mode is a result of the fluid that enters into the hydraulic accumulator and must overcome gas pressure in the high pressure accumulator. The compressed gas creates resistance to the flow of the fluid in the accumulator, which creates increases of pressure of the working fluid and a decrease of the pump rotational speed.

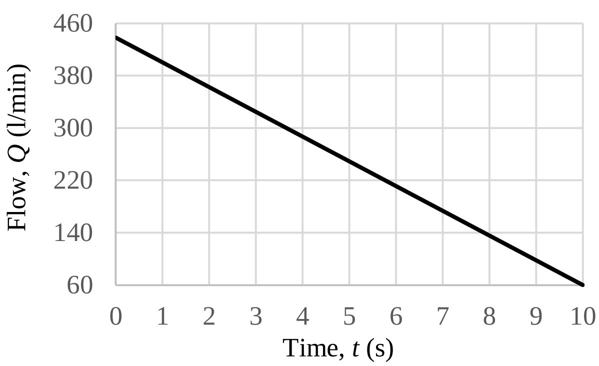

Figure 4 Simulation results of flow from hydraulic pump during vehicle braking mode

The input value in the numerical model of the high pressure accumulator is the output value from the numerical model of the pump, i.e. simulation results of the hydraulic pump flow. The simulation results of the pressure of working fluid in the high pressure accumulator (Fig. 5) show that at the beginning of the simulation the pressure is equal to the pre-charged pressure in the accumulator p_0 = 180 bar.

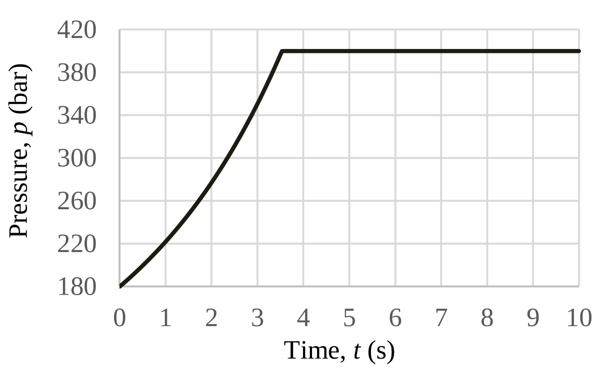

Figure 5 Simulation results of pressure of working fluid in the high pressure accumulator during vehicle braking mode

During the vehicle breaking, the hydraulic pump achieves a flow of the working fluid from the low to high pressure accumulator, which results in the charging of the high pressure accumulator and the increase of the pressure of the working fluid. After 3.5 seconds the maximum pressure of the selected accumulator p_2 = 400 bar is achieved. It can be concluded that the results indicate the real behavior of the developed numerical model. Also, it can be concluded that the chosen adiabatic process was properly chosen because the time of accumulator charge is less than four minutes.

When we compare the obtained results of the pressure with the previous assumption that the high pressure accumulator will be fully charged during the vehicle braking from 70 km/h to 10 km/h during 10 seconds, we can conclude that for the fully charged accumulator 3.5 seconds are necessary; therefore, during the first 3.5 seconds the vehicle is braking by charging the high pressure accumulator and after that the vehicle is braking with main braking system.

The state of charge of the high pressure accumulator is the function of pressure in an accumulator, which is described by the Eq. (8). Therefore, the simulation result (Fig. 6) shows identical behavior as a result of pressure in the high pressure accumulator. At the beginning of the simulation the accumulator is empty, *SOC* = 0, and after 3.5 seconds the accumulator is fully charged, *SOC* = 1.

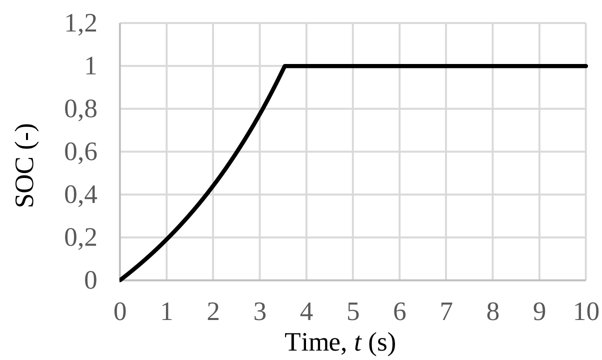

Figure 6 Simulation results of *SOC* of high pressure accumulator during vehicle braking mode

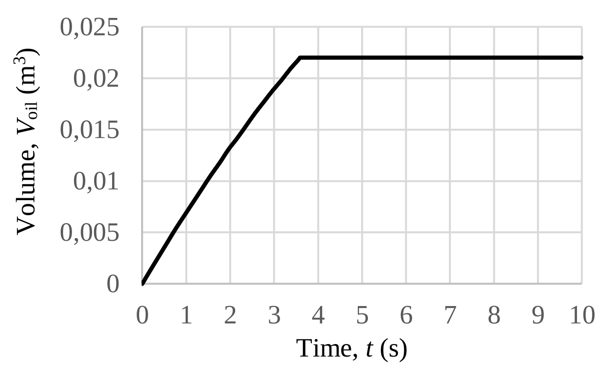

Figure 7 Simulation results of volume of working fluid in the high pressure accumulator during vehicle braking mode

The simulation results of volume of the working fluid in the high pressure accumulator (Fig. 7) show that at the beginning of the simulation the accumulator is empty, V_{oil}= 0. During vehicle braking the volume of the working fluid is increased and after 3.5 seconds V_{oil}=0.022 m^3 (22 l). It can be concluded that during the vehicle braking mode the accumulator is charged about 85% of the maximum value of the accumulator volume (26 l), because after 3.5 seconds of the simulation the pressure in accumulator achieves the maximum value of the pressure of the hydraulic system (400 bar). In that moment, the hydraulic hybrid systems switch to the main braking system.

The simulation results of the volume of gas in the high pressure accumulator (Fig. 8) show that at the beginning of the simulation the gas has maximum volume, V_{gas}= 0.05 m^3 (50 l). During vehicle braking the volume decreases and after 3.5 seconds the value V_{gas} = 0.028 m^3 (28 l) is achieved. It can be concluded that the compression of gas is realized, and the value of compressed volume of gas is equal to the value of the charged volume of the working fluid.

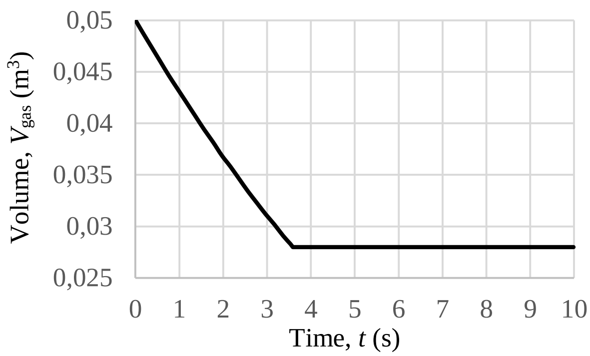

Figure 8 Simulation results of volume of gas in the high pressure accumulator during vehicle braking mode

The simulation results of the temperature in the high pressure accumulator (Fig. 9) show that at the beginning of the simulation the temperature is equal to the environment temperature θ_0 = 293 K. During vehicle braking the temperature increases and after 3.5 seconds the value θ = 369 K is achieved, which is lower than the maximum allowed temperature of the selected accumulator, θ_{max} = 380 K. Once more, it can be concluded that the chosen adiabatic process was properly chosen.

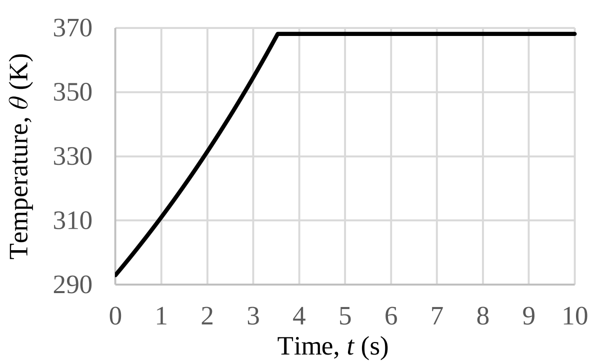

Figure 9 Simulation results of temperature in the high pressure accumulator during vehicle braking mode

The simulation results of the developed numerical model of a parallel hybrid system for a vehicle braking mode have proven the accurateness of the developed model. Also, the results have proven correctness of the previous calculation and selection of hydraulic components.

Furthermore, the simulation results have shown that after 3.5 seconds the maximum pressure (p_2 = 400 bar) of the hydraulic system was achieved and the accumulator was charged about 85%. That simulation result values prove that if the observed hydraulic hybrid system is mounted in the real delivery van, the sufficient amount of energy will be stored in short time. In this way the advantage of using the numerical simulation is also proven, which enables the observation of the dynamic of hydraulic hybrid vehicle in the time domain.

6.2 Simulation of a parallel hybrid system for a vehicle starting mode

When the vehicle begins to accelerate, the working fluid flows from the high to low pressure accumulator. The simulation results of the pressure of the working fluid in the high pressure accumulator (Fig. 10) show that at the beginning of the simulation the accumulator is fully charged and the pressure has maximum value, p_2 = 400 bar. During vehicle starting mode the pressure increases, which is the result of accumulator discharging, and after four seconds the value 330 bar is achieved.

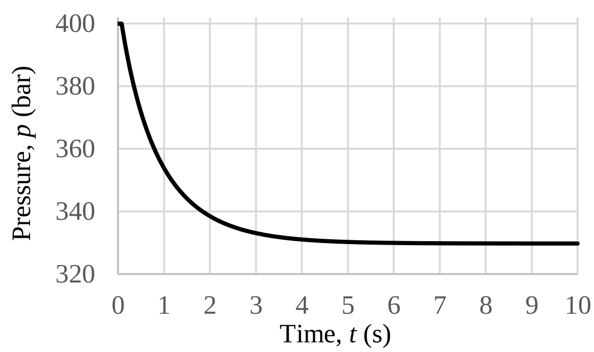

Figure 10 Simulation results of pressure of the working fluid in the high pressure accumulator during vehicle starting mode

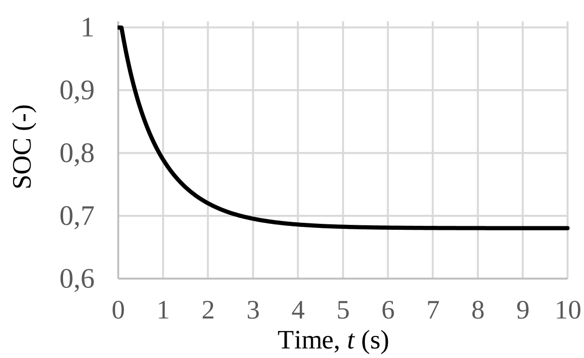

Figure 11 Simulation results of SOC of the high pressure accumulator during vehicle starting mode

Same as the result in the previous simulation, the results of the *SOC* of the high pressure accumulator (Fig. 11) show identical behavior as a result of the pressure in the high pressure accumulator. At the beginning of the simulation the accumulator is fully charged *SOC* = 1 and after four seconds the result of the *SOC* is 0.68. It can be concluded that the high pressure accumulator was not fully discharged because of dynamic of the hydraulic motor effects to the high pressure accumulator, which will be described in following text.

The simulation results of the volume of the working fluid in the high pressure accumulator (Fig. 12) show that at the beginning of the simulation the volume V_{oil} = 0.022 m^3 (22 l). This value of volume was achieved after the vehicle braking (Fig. 7). During vehicle starting the volume of the working fluid is decreasing and after four seconds V_{oil} = 0.0176 m^3 (17.6 l) is achieved. It can be concluded that during the vehicle starting mode the accumulator discharges 4.4 liters of working fluid.

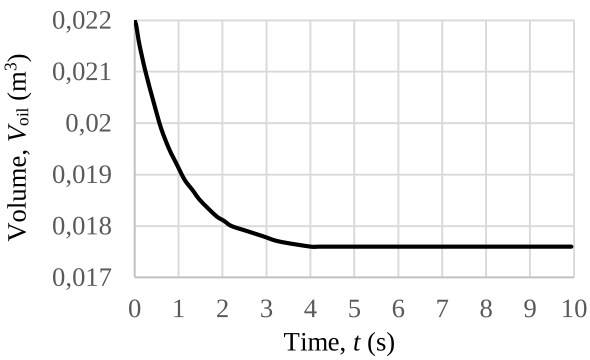

Figure 12 Simulation results of volume of working fluid in the high pressure accumulator during vehicle starting mode

The simulation results of the volume of gas in the high pressure accumulator (Fig. 13) show that at the beginning of the simulation the gas has the volume V_{gas} = 0.028 m^3. This value of the volume of gas is achieved after the vehicle braking (Fig 8.). During vehicle starting the volume increases and after four seconds the value V_{gas} = 0.0324 m^3 is achieved. It can be concluded that the expansion of the gas is realized, and the value of the expansion volume of gas is equal to the value of the discharged volume of working fluid.

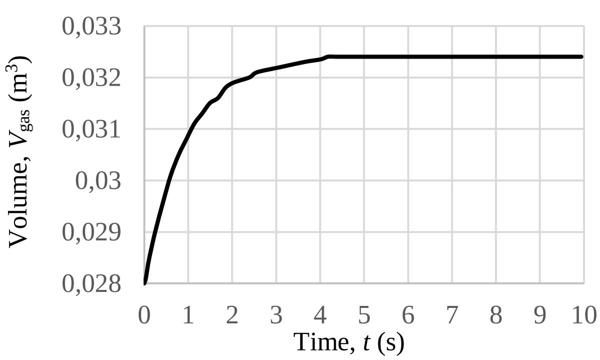

Figure 13 Simulation results of volume of gas in the high pressure accumulator during vehicle starting mode

During the starting mode, the pump acts as a hydraulic motor. The hydraulic motor has to be able to overcome the maximum torque which is needed to start the vehicle. The most unfavorable situation is when the vehicle is driving on an inclined road with the maximum payload where it is necessary to overcome the load torque of 920 Nm. The mentioned value of torque was used for the simulation of the vehicle starting mode during the whole simulation, although in real vehicles the value of load torque decreases after starting the vehicles. The simulation results of the torque of the hydraulic motor shaft (Fig. 14) show that during the first four seconds the hydraulic motor is able to overcome the torque necessary for starting the vehicle. Moreover, the simulation results show that after four seconds the value of the torque of the hydraulic motor is equal to the value of the maximum load torque, which is the result of the mathematical model of the hydraulic motor. Therefore, the hydraulic motor is not able to overcome the load torque of 920 N·m and the hydraulic motor stops rotating.

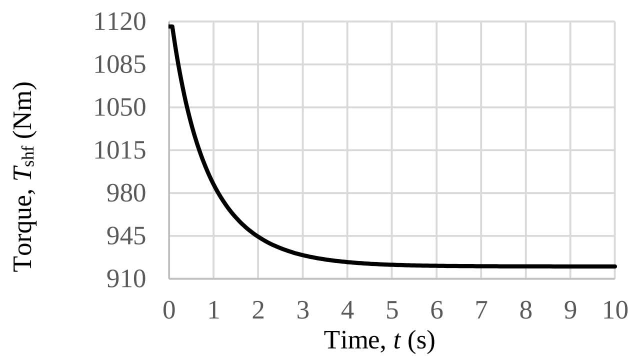

Figure 14. Simulation results of torque of hydraulic motor shaft during vehicle starting mode

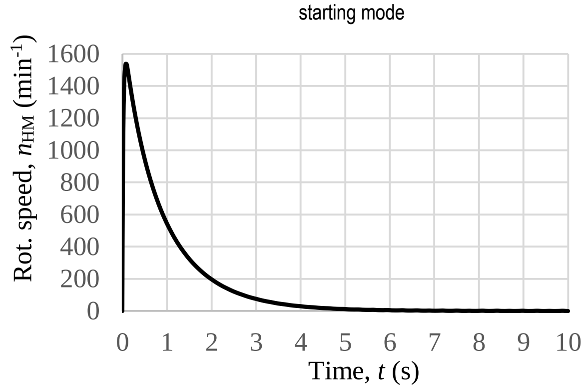

Figure 15 Simulation results of hydraulic motor rotational speed during vehicle starting mode

The simulation results of the rotational speed of the hydraulic motor (Fig. 15) show that at the beginning of the simulation the rotational speed increases to n_{HM} = 1540 min^{-1} and then decreases as a result of decreasing the pressure in the high pressure accumulator. After four seconds the rotational speed of the hydraulic motor n_{HM} = 0 is achieved. This is a result of decreasing the pressure in the high pressure accumulator, which after four seconds is not able to achieve the sufficient torque to overcome the load torque of 920 N·m and the hydraulic motor stops rotating. When the hydraulic motor has stopped rotating, the flow through the hydraulic motor also stops; therefore, the high pressure accumulator stops discharging. Consequently, after four seconds of the simulation, the pressure and volume of the working fluid and gas in the high pressure accumulator have stopped changing.

From the simulation results it can be observed that after two seconds the rotational speed decreases under the minimum value of the hydraulic motor rotational speed 476 min^{-1}, which is defined by the driving regime. Therefore, when the hydraulic motor rotational speed decreases to the minimum value the hydraulic hybrid systems switch to the primary IC engine.

The simulation results of the developed numerical model of a parallel hybrid system for a vehicle starting mode have also proven the accurateness of the developed model and correctness of the previous calculation and selection of hydraulic components.

Moreover, the simulation results have proved that the stored energy in high pressure accumulator during the vehicle braking is sufficient for starting the vehicle in the most unfavorable situation when the vehicle is driving on an inclined road with the maximum payload. In this way, important energy saving is achieved because the delivery van required a lot of energy for starting.

7 CONCLUSIONS

The main objective of this research was to develop a numerical model of the parallel hydraulic hybrid system for the delivery van. The reason for that is that the numerical model can observe the dynamic of hydraulic hybrid vehicle in time domain. Based on the different driving regimes, a calculation of the van hydraulic hybrid system was conducted and standard hydraulic components were chosen. The numerical model of the parallel hydraulic hybrid system was developed using the Matlab/Simulink software. Two numerical models were carried out: the numerical model for the vehicle braking mode and the numerical model for the vehicle starting mode.

Simulation results of the developed numerical model for the vehicle braking mode have shown that during the vehicle braking, the pressure in the high pressure accumulator achieved the maximum value and the accumulator was charged about 85%. Simulation results of the developed numerical model for the vehicle starting mode have shown that the hydraulic motor is able to overcome the torque necessary for starting the vehicle, i.e. the stored energy in the high pressure accumulator is sufficient for starting the vehicle.

Simulation results have proven that the designed hydraulic hybrid system is able to produce, store and reuse energy for all driving regime for the chosen delivery van, and energy saving was demonstrated.

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