SOLUTIONS - EXERCISE 5

Stepper motors and hydraulics

1. STEPPER VS. SERVO

Resources: https://www.youtube.com/watch?v=bngx2dKl5jU. Your answer could contain for example the following elements:

- Stepper motors are known for their ability to move precise, predefined angular steps without feedback control loop. The angular position resolution of a stepper motor is limited to a certain amount of steps in one revolution (a certain number of degrees per step).
- Stepper motors have multiple toothed electromagnets around a gear-shaped rotor. The electromagnets are energized by turns to get the motor rotating. A good visualization of this: http://en.wikipedia.org/wiki/Stepper_motor#/media/File:StepperMotor.gif.
- The stepper motor controller masks the stepper motor controlling for the user. It takes care of the right energizing sequence of the electromagnets. The user sends the controller pulse trains (typically square wave pulse) where every pulse is considered as one step – the controller energizes the windings accordingly. The desired direction of motion is another information to be send to the controller. The pulse frequency defines the motor velocity and the total amount of pulses sets the angle to be rotated.

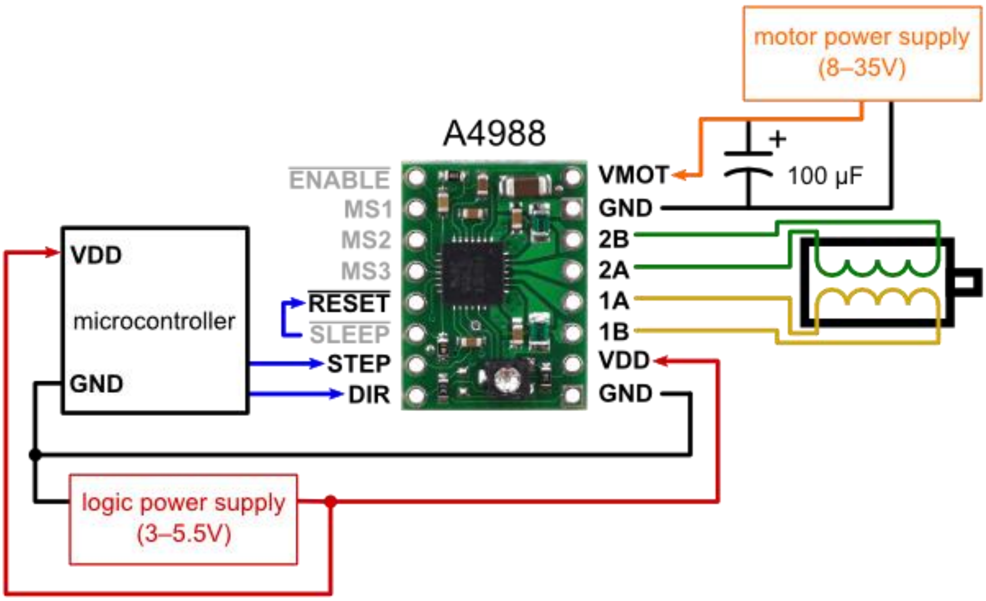

FIGURE 1. STEPPER MOTOR CONTROLLER.

- A servo DC motor is always closed feedback loop controlled, i.e. the rotating angle is measured and the windings are then energized according to the desired state and measured state difference. The positioning accuracy of a servo motor can be much better than the accuracy of a stepper motor; there are no predefined steps where the shaft can be positioned, but the resolution is limited by the resolution of the feedback measurement system (e.g. encoder).

- Stepper pros:
    - simplicity
    - low cost for relatively accurate open loop control
- Stepper cons:
    - torque reduction at higher velocities
    - torque varies depending on the angle if using other excitation mode than full step
    - uses more current than DCs
    - vibration due to operation mechanism
    - risk to skip steps due to overload -> the position is lost (open loop)
- DC servo pros:
    - faster and smoother response
    - continuous operation and torque
    - closed loop control
- DC servo cons:
    - more complicated control electronics
    - price

2. Stepper motors

Q1. The pitch of the screw is *p* mm and thus the slider moves *p* mm during one revolution. The demanded amount of revolutions N depends on the demanded movement x.

$$N = \frac{x}{p}. \tag{1}$$

The step of the motor is *S*, so we need

$$k = \frac{\frac{360°}{rev}}{\frac{S°}{step}} = \frac{360}{S} \frac{steps}{rev} \tag{2}$$

to rotate the motor one revolution. Thus we need

$$N\,rev \cdot \frac{360}{S} \frac{steps}{rev} = \frac{360N}{S} = \frac{360x}{Sp} \tag{3}$$

steps and correspondingly as many control pulses to complete the task.

Q2. Since we know how many pulses it takes to rotate one revolution, we can first calculate the shaft velocity:

$$n = \frac{f}{k}. \tag{4}$$

Now it is easy to calculate the slider velocity:

$$v = np. \tag{5}$$

Q3. The linear sensor resolution equivalent to one step is

$$R = \frac{S}{360} p. \tag{6}$$

Q4. First, let’s calculate the needed pulses to rotate one revolution (half stepping):

$$q = 2 \cdot 50 \cdot i. \tag{7}$$

These pulses must be transmitted to the motor controller during one day, resulting in frequency of

$$f = \frac{q}{24 \cdot 60 \cdot 60\ s} \tag{8}$$

Q5. The real frequency of the signal, if the mechanism turns *R* revolutions during 12 days:

$$f = \frac{R \cdot 2 \cdot 50 \cdot i}{12 \cdot 24 \cdot 60 \cdot 60\ s}. \tag{9}$$

3. HYDRAULIC SYSTEMS

Q1. Because the required pressure is defined as the pressure shown by a gauge type pressure sensor, i.e. pressure relative to atmospheric pressure, and our tank line is in atmospheric pressure, the pressure in the tank line can be considered to be zero in the calculations. Thus the pressure p can be simply calculated with the external force *F* and the area of the cylinder's piston A_p.

$$p = \frac{F_1}{A_{p1}} = \frac{m_1 g}{\pi r_{piston}^2} \tag{10}$$

It is also possible to get the same result by calculating with absolute pressures and taking to account the atmospheric pressure in the opposing cylinder chamber but then you will have to take to account also the atmospheric pressure acting on the area of the rod from outside of the cylinder. Usually this is neglected since the effect is very small.

Q2. Calculations are the same as in Q1 except that now the pressure is acting on the rod side of the piston. Therefore, the area of the rod must be reduced from the area of the piston.

$$p = \frac{F_2}{A_{p2}} = \frac{m_2 g}{\pi (r_{piston}^2 - r_{rod}^2)} \tag{11}$$

Q3. Because lifting mass 1 requires less pressure than lifting mass 2, mass 1 will start moving first and will arrive at the end of its movement first.

When the directional valve is opened, pressure starts increasing in the fluid volume of the channels because the pump is pumping more fluid in to that constant volume. Because cylinders 1 and 2 are connected to the same volume, the pressure in their chambers is the same. The pressure in the channel increases as long as the volume of the channel stays constant and the pump is increasing the amount of fluid in the channel. Since hydraulic fluid is only minimally compressible, this happens almost instantaneously.

When the pressure reaches the level which is required to lift mass 1, the piston of cylinder 1 starts moving, which increases the volume of the fluid channel. The pressure in the channel stays constant during the movement of mass 1 since the fluid volume increases due to the piston's movement by the same amount as the pump is transferring fluid to the channel. The velocity of the piston in cylinder 1, and thus also the velocity of mass 1, is therefore defined by the flow rate produced by the pump.

When the piston in cylinder 1 reaches the end of its movement, the volume of the channel can no longer increase but the pump is still increasing the amount of fluid in it. Therefore, the pressure starts increasing again. When the pressure is high enough to lift mass 2, it starts moving again at the speed defined by the pump's flow rate. The pressure stays again constant. When mass 2 reaches the end of its movement, pressure rises to the cracking pressure of the pressure relied valve and all the flow rate produced by the pump flow through the pressure relief valve.

Q4. As explained in Q3, the speed of mass 1 is defined by the flow rate produced by the pump and the area of the piston.

$$v = \frac{q}{A_{p1}} = \frac{V_r n}{\pi r_{piston}^2} \tag{12}$$

Remember to convert rpm to 1/s and cubic centimeters to cubic meters.

4. Hydraulic servomotor with Simulink

Q1. The higher level structure of the model stays unchanged compared to the DC servomotor model. The only change is that in this case our PI controlled produces torque instead of voltage. In reality, the controller would only produce the control signal for a valve which controls the pressure difference over the hydraulic motor.

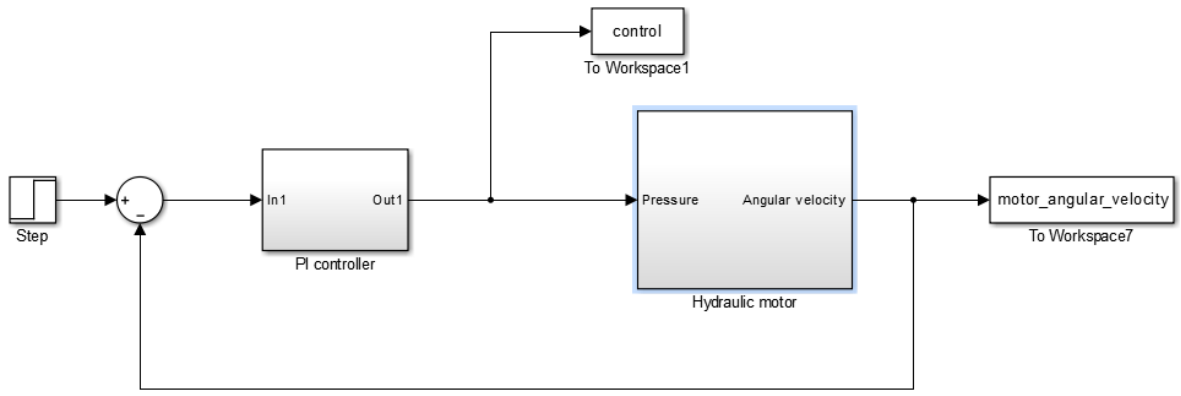

Figure 2 Main structure of the model

The important changes are done inside the motor model. First, all the blocks representing the electrical part of the motor should be removed since there are no electrical parts in a hydraulic motor. These include the resistance, inductance, current integrator, back emf constant and the torque constant which links the electrical and mechanical parts of the model.

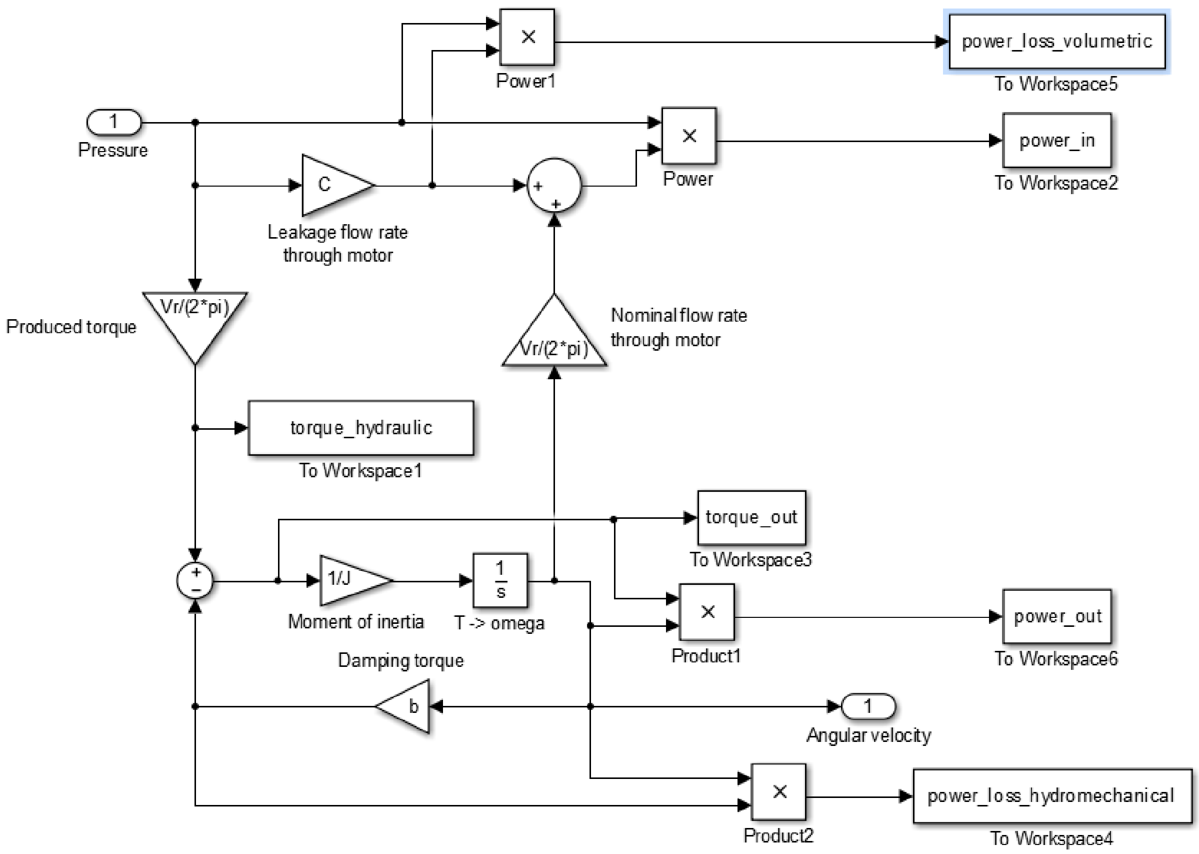

Figure 3 Hydraulic motor model

The electrical part is replaced, as instructed, with an ideal pressure dependent hydraulic torque source which can be represented by a single gain block which has the pressure differential as input and torque as output.

The hydromechanical efficiency, which represents all mechanical and fluid flow related frictions inside the motor, is in this model represented by the damping torque so no hydromechanical efficiency constant is required in the torque production equation.

The volumetric efficiency in the model is represented by the laminar leakage flow through the motor. When the motor rotates, it requires input flow according to the angular velocity and the displacement per rotation. In addition to that, some fluid leaks through the motor for example between the teeth of a gear motor or past the pistons in a piston pump. The total flow through the motor is the sum of the nominal flow and the leakage flow. Therefore, also the total input flow into the motor is the sum of the nominal flow and the leakage flow.

Q2 & Q3. Because the gear ratio was changed, also the reduced inertia of the load must be updated from the previous exercise.

Since the output of our PI controller is now in pascals, which is a very small unit, the gains of the PI controller must be increased by several magnitudes compared to the previous exercise round. With Kp = 55000 and Ki = 20000 the results are following.

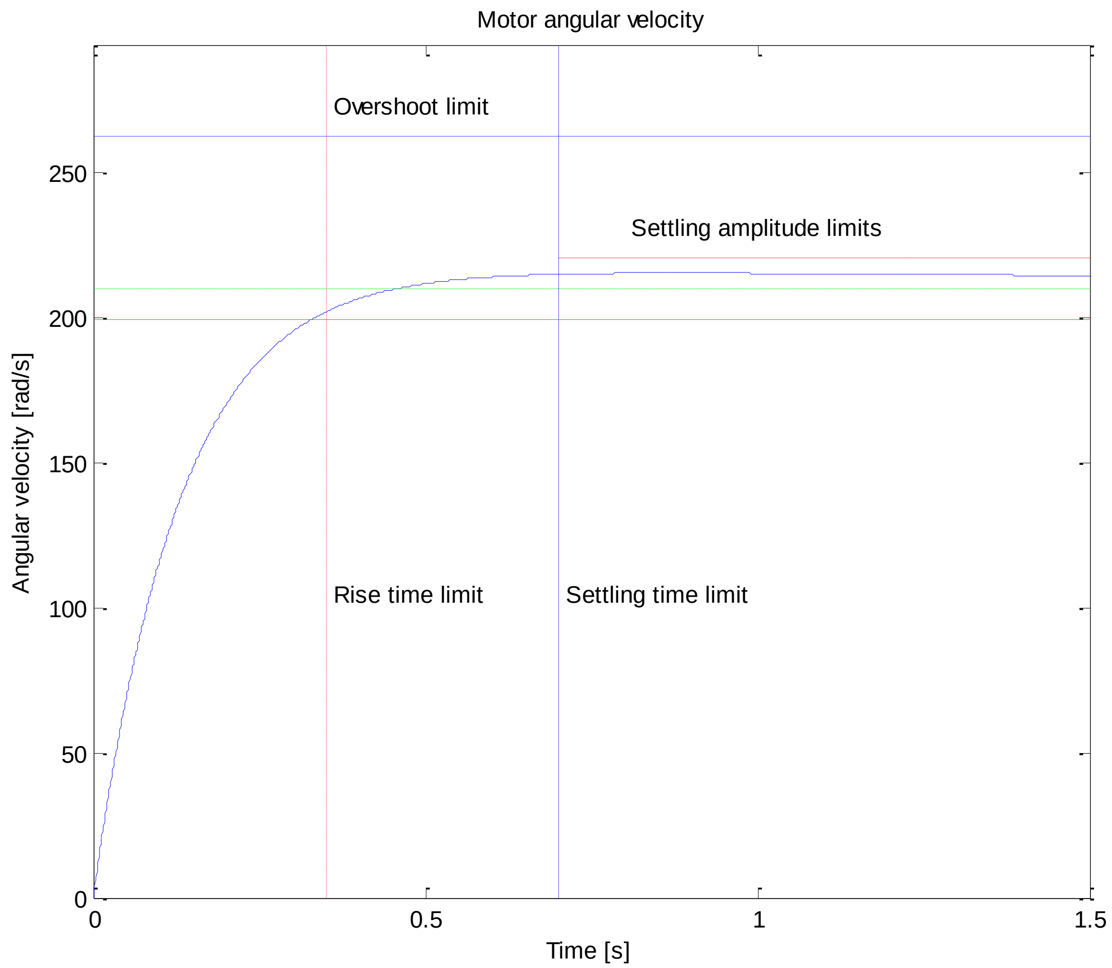

FIGURE 4 ANGULAR VELOCITY RESPONSE

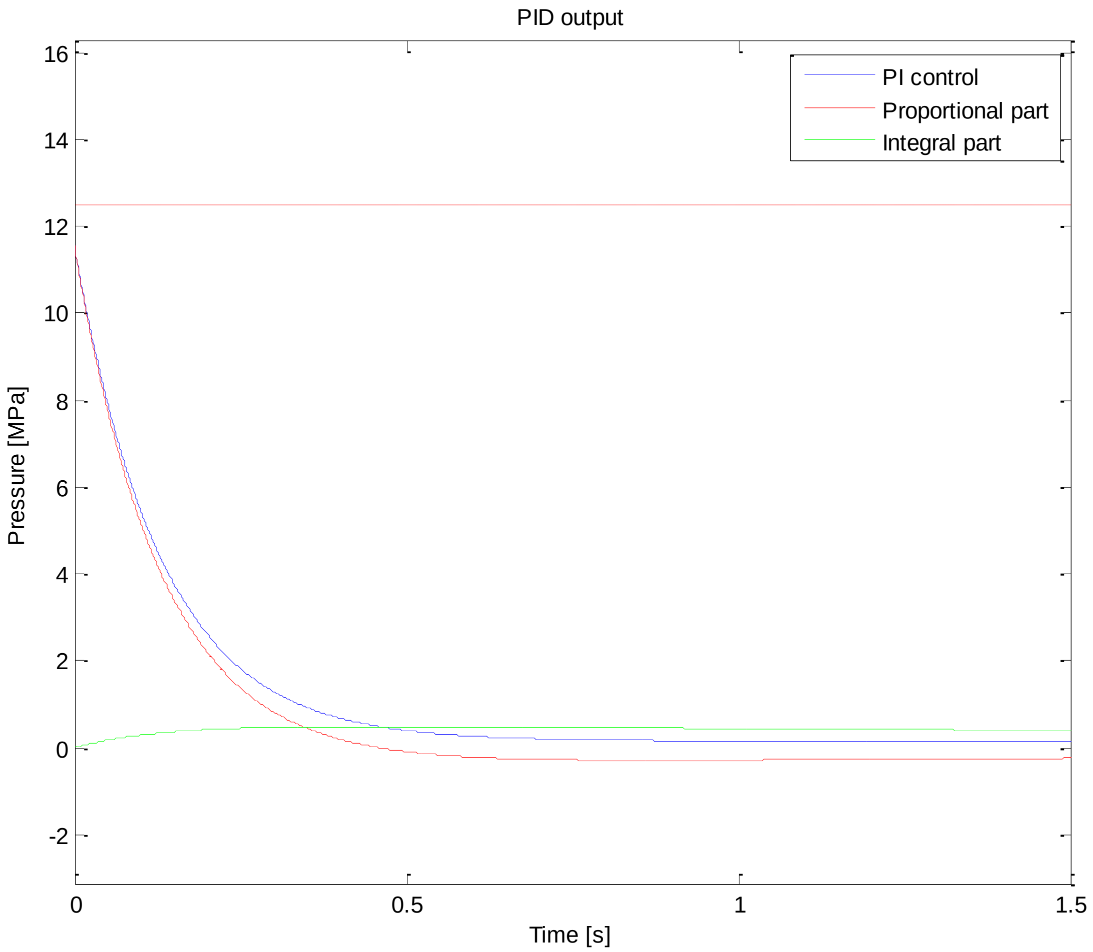

FIGURE 5 PI CONTROLLER OUTPUT I.E. PRESSURE DIFFERENCE

If you got result curves with sharp edges, it is probably because you did not limit the time step of the solver.

Q4. The output power is still the product of output torque and angular velocity. The input power is the product of total input flow rate and pressure difference or it can be also calculated as the sum of output power and volumetric and hydromechanical losses. Power loss due to the leakage is the product of leakage flow rate and pressure difference.

Since the volumetric and hydromechanical losses stay pretty low, the input and the output powers stay close to each other. The volumetric loss depends on the pressure difference and thus it is large initially but since the pressure difference is very small after the initial acceleration, also the volumetric losses reduce to almost zero. At constant speed, all the input power is lost with hydromechanical losses since there is no external load and thus no output power.

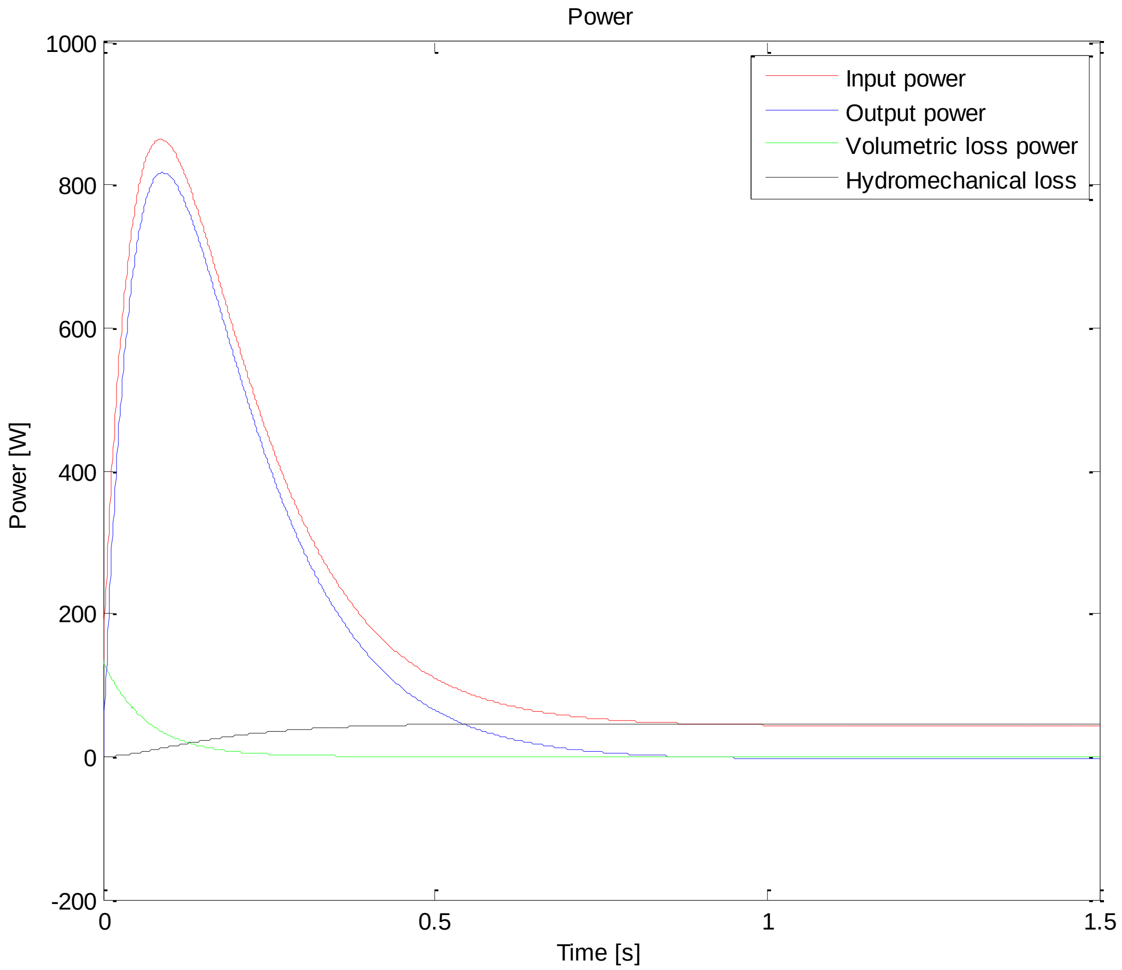

FIGURE 6 POWERS