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Comparison of hydro-pneumatic accumulator's charging performance under different thermal process for dual hybrid driveline

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Abstract. The hydro-pneumatic accumulator is broadly used in heavy industry as leaking compensator. Lately, much interest has been developed to study the component as energy storage. The important element in the selection of accumulator is performance. Hence, research was conducted to examine the use of thermal process condition and its impingement on the accumulator as an energy store. For charging process, the performance is influenced by the thermal process because it involves temperature change and heat transfer. Both processes are producing a different performance. The storage system is planned to be adapted in the dual hybrid driveline. A simulation study has been conducted by using Automation Studio software which focusses on two different processes called isothermal and adiabatic. The process has involved a schematic design, functional testing, parameter setting, and pretense. The result has shown that the thermal process affected the fluid power parameters such as power, effective storage capacity, and temperature differentials. The isothermal process produced higher effective volume compared to adiabatic process, stored higher power and had lower temperature differentials. Regarding charging speed, the adiabatic was faster. However, it was a lack of storage capacity.

1. Introduction

The hydro-pneumatic accumulator is one of a critical component in the heavy industry sector. They are employed in the hydraulic system for processing high workload processes such as stamping and pressing. This is imputable to the ability of this component to run as a leaking compensator, backup energy system, and surge control. In the sectors of automobiles, the hydro-pneumatic accumulator is used as shock dampening component and also energy storage [1][2]. As a shock dampening, the component reduces fatigue of hydraulic and mechanical components. As for the energy storage, it is used to store the energy obtained from the active or passive way to be used as a backup system, and sometimes it functions as a secondary energy source. In this project, the focus is the application of hydro-pneumatic accumulator to the hybrid driveline of the hybrid vehicle. It will be installed to the dual hybrid hydro-pneumatic driveline. This component acts as secondary energy storage. In this hybrid system, accumulator conducts three (3) main functions, which is charging, storage and discharge process. Charging is the process of compression to allow conversion of the kinetic energy transformed into potential energy. It will cut down the bulk of the compressed gas and increase the force per unit area. Storage function, on the other hand, is the process of storing energy and serves as a cooling system. While the discharge is the process of converting the stored potential energy, transmitted to the propulsion system to move the vehicles when needed. The primary focal point of this report is the charging process which is tender to the thermal reaction. Thermal change can affect the performance of components in terms of power, charging time and the capacity of effective volume.

There are three (3) types of accumulators used in the industry, namely, piston, diaphragm and the bladder type. In terms of output power, the bladder type is the best. While about compression ratio, piston accumulator would be preferred. For the application of dual hybrid driveline, the bladder-type accumulator is most suitable based on the characteristics of high power output, average effective volume, and high operating pressure. Bladder-type accumulator consists of various components, namely shell, bladder, nitrogen gas and safety valve. The shell is the main casing and usually made of steel or composite material. It is designed based on the standards of the pressure vessel. The bladder is elastomer materials having characteristics of high elastic to keep the pre-charge pressure and also to stand the compression of hydraulic oil. Nitrogen gas is an ideal gas to absorb the compression energy because it delivers excellent properties to withstand the temperature change. In this matter, the nitrogen is better than the air [3]. The charging process detail described through the figure 1. The process of compression results in an increase of temperature of the system. It can cause some unwanted behavior such as the volume change due to the thermal expansion. This behavior at once affects the system's maximum pressure, power and shorten the lifespan of the parts. Thus, the temperature of the compression process needs to be controlled to obtain the optimal operation of the energy storage. In this case, the most common process condition that has been applied to thermal storage are isothermal and adiabatic. In isothermal condition, the fluid temperature remains constant. The heat exchange between gas and atmosphere is instantaneous. Meanwhile, the adiabatic occurs without transfer of heat or matter between a thermodynamic system and its surroundings. In an adiabatic process, energy is transferred only as work. It also defined as zero heat transfer, and all the work that has been done on the system raises the temperature of the system. So, the internal energy between these two systems are different and affected the performance of the application system.

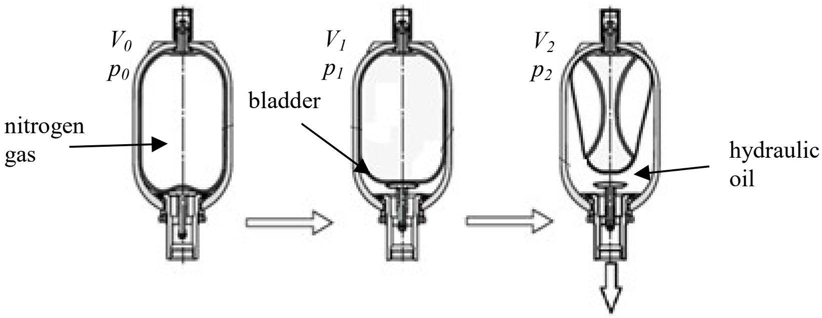

Figure 1. Bladder condition during the charging process [4]

One of the important element that affects the performance of compression is the thermal process. The thermal process commonly studied by researchers are isothermal and adiabatic. The energy storage divided into two because each process has their unique behavior. The concept of thermal energy storage is mostly applied to electric energy generation system, hydraulic system and also vehicle system [3]. Isothermal energy storage is a storage system that maintains the inside temperature during the process of compression. As there is no change in internal temperature, $\Delta T = 0$, the temperature increase due to pressure change converted in the form of heat transfer. Since the internal temperature does not alter, then higher pressure could be kept in the storage. While adiabatic energy storage has some sort of insulation resulting occurrence of heat transfer is zero. The temperature in the system as a whole is transformed into internal energy. Therefore, each of the process has unique features and contribute to the application of the storage system. However, application compatibility is dependent on the process used [5]. In wind-tidal electric energy generation [6], energy storage is used to store pressure energy produced by wind blade. This pressure energy stored and cooled. When needed, it will be channeled to Pelton turbine to turn the generator. The hydraulic medium used is seawater. The analysis indicated that the energy efficiency could exceed 100% as the small input speeds are seen, and the exergy efficiency has the consistent change trends with demand power. At the same time, the study also said that there is an inversely proportional relationship between the accumulator and hydraulic equipment including the pump and nozzle in terms of proportions. Among the energy storages that have been introduced, two well-known energy storages have been proposed called isothermal compressed air energy storage (Isothermal CAES) and adiabatic compressed air energy storage (Adiabatic CAES) [7]. Each has its unique solutions. Based on the study, isothermal has a potential round-trip efficiency of 70-80%, meanwhile adiabatic efficiencies more than 70%. Although the two systems have advantages, however, several challenges must be solved to optimize the efficiency of the compression process. Then it can be manipulated to suits for expansion process [8]. In an ideal solution, the isothermal process is used to maintain the compression so that a higher pressure is obtained and expand the energy in adiabatic[9]. Means, there are no losses occurred during the expansion process. However, to acquire a combination of two processes in one system is very challenging.

Due to the above statement, simulations are carried out to study the effect of isothermal, and adiabatic compression (charging process) on the performance of hydro-pneumatic accumulator for dual hybrid driveline system. By knowing the characteristics of each charging condition, all aspects can be considered in the process of designing energy storage as well as facilitate optimization process.

2. Methodology

2.1. Schematic diagram

The schematic diagram consists of various components. Pump system was employed to supply hydraulic oil in hydrostatic approach. Accumulator works as energy storage, a pressure gauge is used to measure the pressure change and pressure relief valve to secure the circuit. The pump runs to create a flow that will go to the accumulator. The accumulator consists of the bladder that filled with nitrogen gas. Once the bladder being compressed, it stored energy in terms of potential energy. The pressure will keep increased, and when it reaches the setting limit, the pressure relief valve will open and secure the system as illustrated in Figure 2.

2.2. Equation derivation

Based on figure 2, derivation has been developed. The first approach to study the effect of the input parameter to the capacity of the storage and the effect of the different thermal process during compression. Charging process starts with pumping fluid from pump to accumulator unit. The pressure different through the pump can be expressed as follows [10],

$$\Delta p = p_{out} - p_{in} \tag{1}$$

Where Δp is the pressure different (N/m^2), p_{out} is pump output pressure (N/m^2) and p_{in} is pump input pressure (N/m^2). Pressure entering the accumulator is defined as

$$p_{acc} = p_{out} - h_m - h_f \tag{2}$$

Where p_3 is the input pressure (N/m^2), p_2 is pump output pressure (N/m^2), h_m is minor losses and h_f is friction losses (N/m^2). Since the main function of the pump is to create flow, the actual output flows produced by the pump can be expressed as

$$Q_{out} = C \cdot \omega \cdot \eta_V \tag{3}$$

Where Q_{out} is pump output flow rate (m^3/s), C is displacement (m^3/rad), ω is nominal speed (rad/s) and η_V is volumetric efficiency. The pressure relief valve (PRV) works as a safety valve. The preset value is a maximum pressure allowed to be working in the system. If the charging pressure were exceeded, the PRV opened to stabilize the pressure. So, the pressure in the system can be controlled.

For the internal compression effects on nitrogen bladder, the analysis made by using equations derived from Boyle-Mariotte's law for gasses [10][4], which takes into account the compressibility of the gas.

$$\Delta V_{eff} \leq V_1 - V_2 \tag{4}$$

Where ΔV_{ideal} is the effective volume, V_1 is volume before compression and V_2 is volume after compression.

$$p_0 \times V_0 = p_1 \times V_1 = p_2 \times V_2 \tag{5}$$

Where p_0, p_1, p_2 is pressure at a certain stage of compression and V_0, V_1, V_2 is the volume at a certain stage of compression. Refer to figure 1 for detail stage of charging. In terms of storage capacity, in adiabatic condition, the total volume represented as;

$$V_{total} = \frac{p_{\min} \cdot Z_0 \cdot \Delta V_{eff;T\min}}{p_{0;T\min} \cdot Z_1} \cdot \frac{1}{1 - \left(\frac{p_{\min} \cdot Z_2}{Z_1 \cdot p_{\max}}\right)^{\frac{1}{n_c}}} \tag{6}$$

However, if the condition is not in isothermal then, the equation is as follows,

$$V_{total} = \frac{p_{\min} \cdot Z_0 \cdot \Delta V_{eff;T\min}}{p_{0;T\min} \cdot Z_1} \cdot \frac{\left(\frac{Z_1 \cdot p_{\max}}{p_{\min} \cdot Z_2}\right)^{\frac{1}{n_d}} \cdot \left(p_{\min} \cdot Z_2\right)^{\frac{1}{n_d}}}{\left(Z_1 \cdot p_{\max}\right)^{\frac{1}{n_d}} - \left(p_{\min} \cdot Z_2\right)^{\frac{1}{n_d}}} \tag{7}$$

Where V_{total} is the total volume of the accumulator. ΔV_{eff} is effective volume. $p_{\min}$ is minimum working pressure, $p_{0;T\min}$ is pre-charge pressure at minimum temperature, $p_{\max}$ is maximum working pressure, $Z_{0,1,2}$ are compressibility factor, n_c is charge coefficient and n_d is discharge coefficient. This is the simplified equation to calculate the average time required for the filling process.

$$V_{eff} = \Delta V_{total} \cdot K_i \tag{8}$$

$$V_{eff} = Q_i \cdot t_i \tag{9}$$

Where V_{eff} is effective volume, and K_i is correction factor, Q_i is the flow rate and t_i is the filling time. The equation (12) is then rearranged,

$$t_i = \frac{\Delta V_{effective}}{Q_i} \tag{10}$$

2.3. Simulation

Automation Studio is a tool for design, functional simulation of complex automation and documentation. The software includes hydraulic, pneumatic and electrical operative devices as well as a command part diagram as shown in figure 2(a). It also provides technical and commercial data for simulation. Undoubtedly, software such as Matlab has high flexibility to run a simulation related to the case study, but the Automation Studio has the advantage in terms of circuit design, functional simulation, fluid power component sizing, system design, validation and virtual simulation [6]. These advantages had resulted the selected software was more suited for this project. Currently, Automation Studio has improved their functional by connecting some applications with Simulink, Matlab [11]. All specifications for components, fitting, and measurement tool can be inserted into the provided dialogue box as illustrated in figure 2(b). Since the study involves bulk modulus and heat transfer, so the two functions were enabled. Bulk modulus effect occurs because of the process of nitrogen compression in the bladder [12]. By enabling the bulk modulus setting, the simulation considers the effects of compression in charging and discharging process. The thermal setting considers the fluid heating in contact with the pipe component inner surface. Pressure elevation is set as an independent variable. Temperature, effective volume and charging time as dependence variable. As for pre-charge pressure and volume displacement, the parameters are set as a control variable as shown in table 1.

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The system powered by motor and pump which turns at 2000 RPM and 90% thermal efficiency. The accumulator is 225 mm in diameter set zero as initial volume with 25°C nitrogen gas in the bladder. The simulation system secured by pressure relief valve that set as 400 bar cracking pressure.

Table 1. Simulation parameters

Motor rotational speed	2000 RPM	Initial liquid volume	0 liter
Pump thermal efficiency	90%	Accumulator internal diameter	225 mm
Pump heat transfer coefficient	10 W/m^{2}K	Type of process	Isothermal/Adiabatic
Hydraulic oil	Hydraulic AW-32	Gas type	Nitrogen
Ambient temperature	25°C	Gas temperature	25°C
Cracking pressure	400 bar	Hydraulic line	Steel
Port 1 (C_v)	12	Line type	NPS ¼-DN8

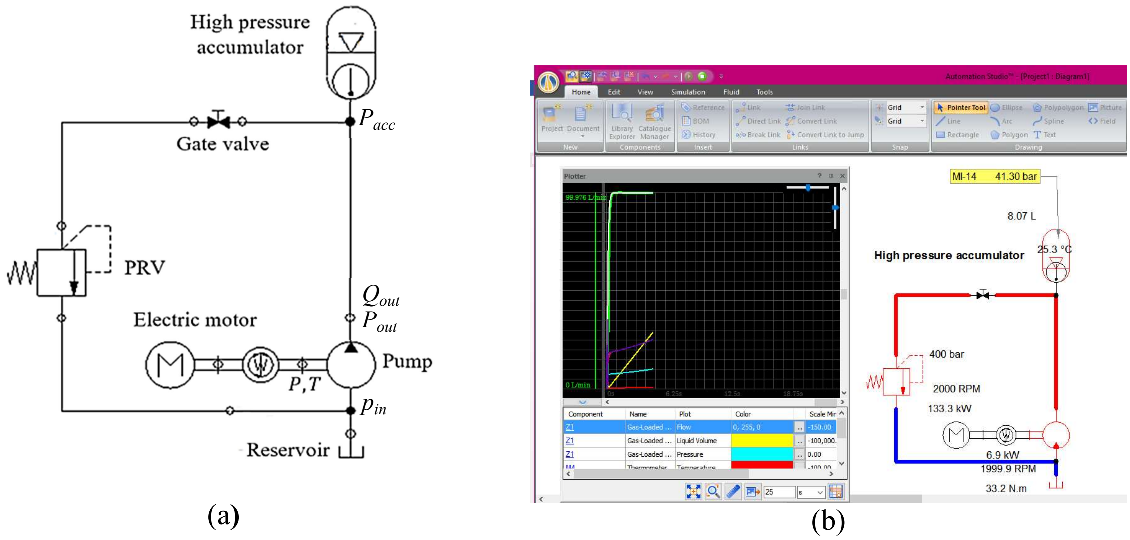


Figure 2. Schematic diagram (a) and Automation Studio GUI (b)

3. Results and discussion

3.1. Effect of pre-charge pressure to effective volume

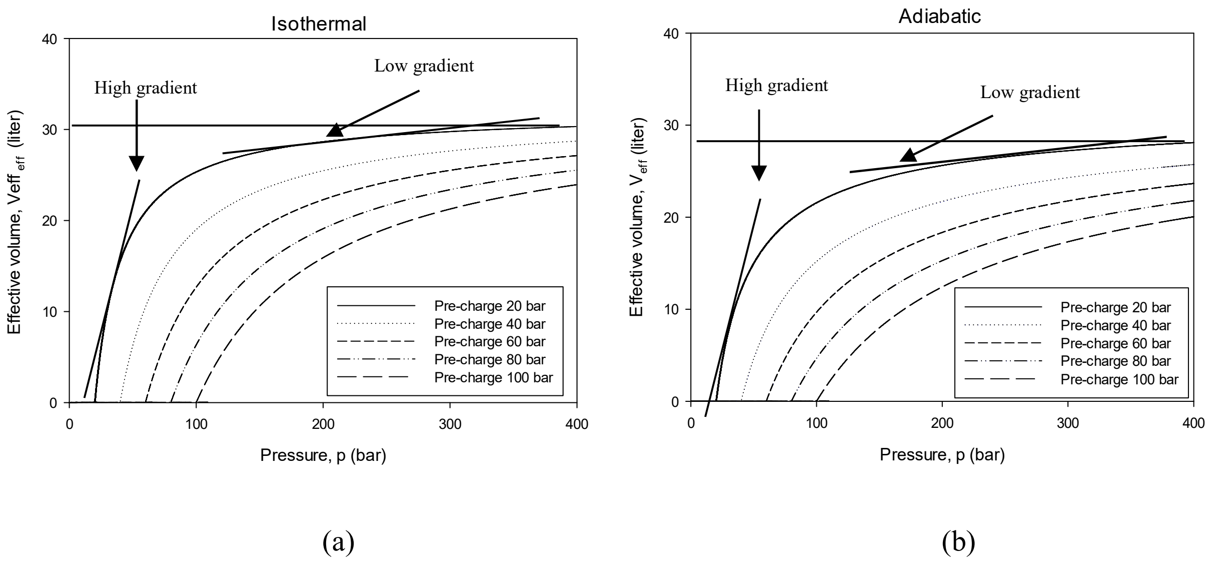


Figure 3. Effects on effective volume

Figure 3(a) Isothermal process produced better output in term of effective volume compare to adiabatic. The highest Isothermal output is 30.35 liter at pre-charge 20 bar and the lowest is 23.8 liter at 100 bar pre-charge. Both occurred at 400 bar system pressure as illustrated in figure 3(b). These are due to the resistive force created by 100 bar pre-charge are more superior compared to the 20 bar. The highest adiabatic output is 28 liter at pre-charge 20 bar and the lowest is 20 liter at 100 bar pre-charge. In term of graph gradient for the pre-charge 20 bar, rapid pressure changes occurred below 80 bars. For the pre-charge 40 bar, rapid pressure changes occurred below 160 bars. At pre-charge 60 bar, rapid pressure changes occurred below 230-240 bars. At pre-charge 80 bar, rapid pressure changes occurred below 80 bars. At pre-charge 100 bar, rapid pressure changes occurred below 380-400 bars. The value of high gradient showed that the filling proses (compression) happen efficiently. However, the lower gradient compression consuming time and it is representing over-pressure condition. Over pressure is not good for the accumulator since it will shorten the lifespan of the bladder and service time. Based on the graph, it might be the reason why a hydraulic company such as Bosch Rexroth, Parker and Hydac decided to limit the compression ratio of the accumulator. It is to prevent overpressure [13][4]. In term of efficiency, at 400 bar, the isothermal process at pre-charge pressure of 20 bar shown 3% higher volume than the adiabatic process at the same pre-charge pressure. Meanwhile for the pre-charge pressure 100 bar, the isothermal shown about 10% higher volume than the adiabatic process at the same pre-charge pressure.

To ensure that the simulation data and result are reliable and correct, the pre-charge pressure effect on storage pressure and effective volume has been compared with Bosch type industrial accumulator. Since all the specifications are different, then the comparison cannot be made a point to point. However, it is just a reference to proceed with the next simulation. In this case, the resulting profile is the same. It started at the pre-charge pressure, increased in quadratic, once it achieved maximum pressure, the gradient start fall which translated to inefficient compression. It called inefficient because the input given was high, but the volumetric output is small. It also called over-pressure condition.

Figure 4 has shown the comparison of effective volume based on pre-charge pressure. It was found that the higher the pre-charge pressure, the wider the difference of effective volume between isothermal and adiabatic. The lowest difference is at 20 bar pre-charge, and the highest was 100 bar pre-charge pressure. It was noted that the volume changes from 20 bar to 100 bar pre-charge was 6.32

liter in isothermal and 8.04 liter in adiabatic. These differentials are due to the effect of temperature change and heat exchange.

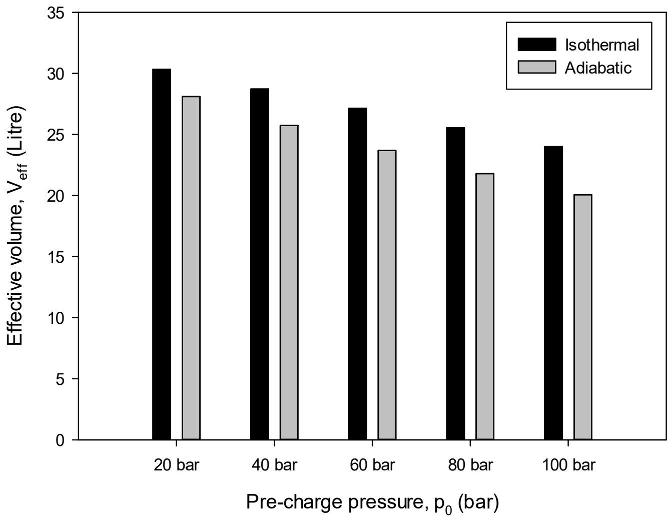
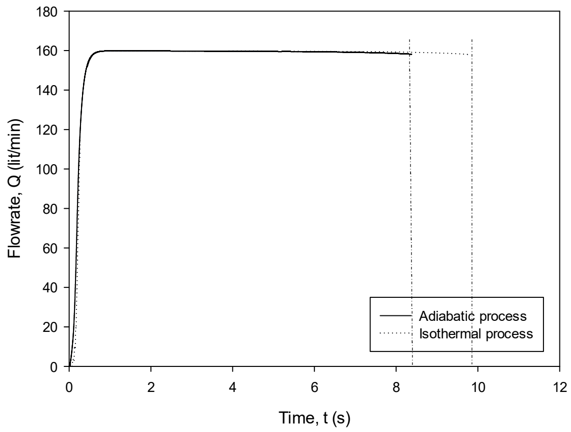
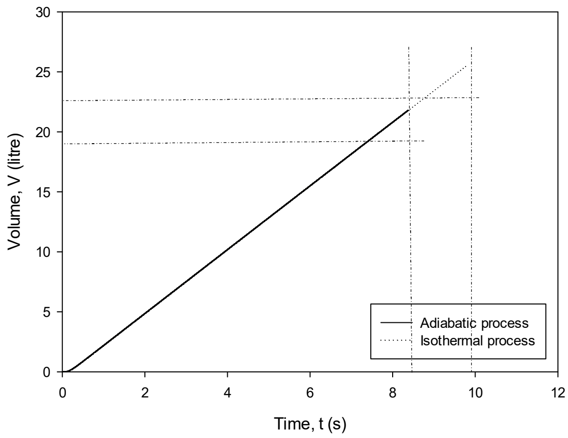


Figure 4. Comparison between isothermal and adiabatic effective volume capacity

Based on figure 3 and figure 4, it was found that the safe, reasonable effective volume and comes with lower inefficient volume is 80 bar pre-charge pressure. The pre-charge pressure does not experience significant over-pressure condition and maintains adequate effective volume. However, this decision does not count the input power given to the system. More detail selection of input type and performance characteristic is required to further the study. Therefore, the next simulation will focus on 80 bar pre-charge pressure.

3.2. Flowrate and effective volume



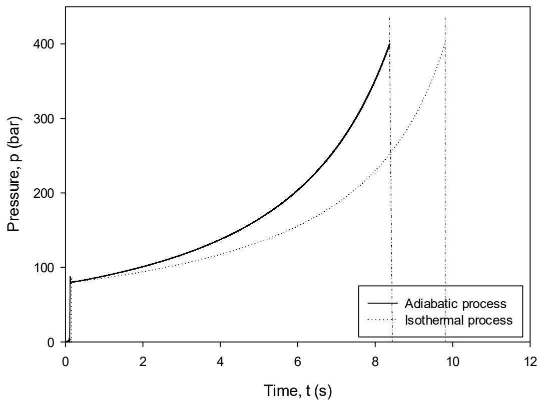


(a) (b)

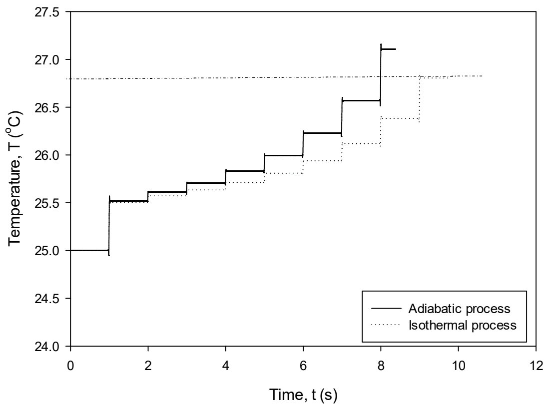
Figure 5. Filling time-based on flow rate (a) and from empty to full (b)

Based on the Bosch catalog [14], the pump speed has been set to the average speed of 2000 rpm and 80 cm^3/rev and it took only 1 to 2 seconds for the pump to raise 0 – 160 liter/min for both process condition as illustrated in figure 5(a). However, from the beginning, the flow rate for the adiabatic process is higher compared to the isothermal until it reached consistency flow rate of 160 liters/min. The flow rate for both (isothermal and adiabatic conditions) had no significant difference when it achieved the stable flow rate. It remains stable at 160 liters/min until about 2 seconds before it reached its effective volume. The isothermal process requires more time about 9.8 seconds to achieve effective volume meanwhile adiabatic process took about 8.2 seconds which is 1.8 seconds faster. This statement has proven that the thermal process condition has affected the flow rate of the filling process and directly influence the filling time. Figure 5(b) shows the effective volume are related to the flow rate. The isothermal process requires more time about 9.8 s to achieve an effective volume of 25.5 liters. Meanwhile, the adiabatic process took about 8.2 seconds to achieve 22 liter which is 3.5 liters less than the isothermal process. The different volume between both processes consider significant (13.7%) and will affect the running time of the hydraulic motor. It can be seen clearly that adiabatic compression able to fill the accumulator much faster but as a consequence, the volume was less than the isothermal.

3.3. Pressure elevation



(a)

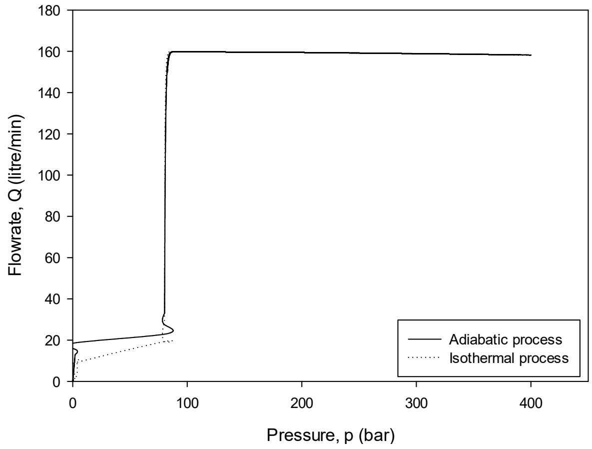


(b)

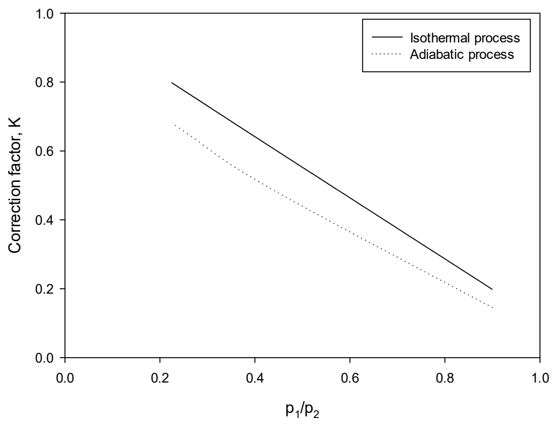
Figure 6. Pressure profile during filling process (a) and temperature elevation (b)

Figure 6(a) has shown that isothermal process requires more time about 9.8 seconds to achieve effective volume meanwhile adiabatic process took about 8.2 s which is 1.6 s faster. The adiabatic process elevates pressure quicker than the isothermal from the beginning to the cracking pre-charge pressure and finally to the final system pressure of 400 bar. It clearly showed that the adiabatic was faster in term of the filling process compared to isothermal. The flow rate difference and the non-existence of heat exchange are the reasons to justify this behavior. The setting temperature of Nitrogen is 25°C. Maximum temperature recorded by the adiabatic process is 27.2 °C from 25°C. Meanwhile, the isothermal recorded maximum 26.7°C. Less about 0.5 °C as illustrated in figure 6(b). This value seems small, but it contributes to the ability to store the energy. Adiabatic process kept higher temperature, but as consequences, it stores less capacity. Isothermal produces less heat, but it can store more volume capacity. One thing to remember that the temperature value extract at the fluid temperature. Since the hydraulic oil is cooling medium, so it neutralized the heat produced by the compression process inside and of the bladder. There is a tendency that the temperature at the compressed nitrogen and bladder is higher than the recorded.

3.4. Performance: Hydraulic power and correction factor



(a)



(b)

Figure 7. Flowrate profile based on pressure storage increase (a) and correction factor for isothermal and adiabatic condition (b)

Power is a correlation between pressure and flow rate as represented in figure 7(a). At the beginning of the compression, the graph shows fluctuation until the pressure exceeds pre-charge pressure. This behavior happens because of the resistive force produced by the gas and bladder. When the flow rate increase, the pressure remains constant. Moreover, when the flow stabilized, the pressure increased from 80 bar to 400 bar. This represents that the hydraulic power keeps increase when the pressure increase. Validation has been done by comparing the graph pattern to the existing industrial Bosch Rexroth accumulator that available in the market. It was found that the profile of effective volume is similar, but the value is different because of the differences in parameter used. The ratio of maximum and minimum operating pressure also within the same range as illustrated in figure 7(b). The isothermal correction factor is higher compared to the adiabatic. The maximum correction factor for an isothermal process is 0.8 while the lowest is 0.22. Adiabatic correction factor started lower than isothermal from 0.17 to 0.7.

4. Conclusions

For the pre-charge pressure, the higher the pre-charge pressure, the lower the effective volume capacity. Due to the resistive produced by the nitrogen gas and bladder. The isothermal process produced higher effective volume compared to adiabatic process. In term of charging speed, the adiabatic was faster however it is lack of storage capacity. This is due to its higher flow rate. No significant thermal increase recorded since the hydraulic oil itself acts as a coolant to the system. The results of this simulation clearly showed that if the capacity and temperature change is decisive to the selection process, then the isothermal process is the best. While if the filling time is decisive, then the adiabatic is the best results.

Acknowledgement

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