INCREASING HYDRAULIC ENERGY STORAGE CAPACITY: FLYWHEEL-ACCUMULATOR

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Abstract

The energy storage density of hydraulic accumulators is significantly lower than energy storage devices in other energy domains. As a novel solution to improve the energy density of hydraulic systems, a flywheel-accumulator is presented. Energy is stored in the flywheel-accumulator by compressing a gas, increasing the moment of inertia of the flywheel by adding hydraulic fluid, and by increasing the angular velocity of the flywheel. Through a numerical model of the energy flows in the system, the energy storage of the flywheel-accumulator was demonstrated to be approximately 10 times greater than a conventional accumulator. Furthermore, the flywheel-accumulator allows the hydraulic system pressure to be independent of the quantity of energy stored. The integral flywheel-accumulator presents numerous future research challenges, yet offers the potential to transform and enable numerous applications including plug-in hydraulic hybrid vehicles.

Keywords: hydraulic energy storage, flywheel, accumulator, hydraulic hybrid vehicle

1 Introduction

Hydraulic energy storage is important to numerous applications including hydraulic hybrid vehicles and alternative energy sources such as wind turbines. Hydraulic energy is typically stored in a hydraulic accumulator, which is a pressure vessel containing a gas that is compressed by the addition of hydraulic oil to the pressure vessel. The energy storage density of hydraulic accumulators is significantly lower than other energy storage mediums. To illustrate this point, consider the specific energy of batteries at 216 kJ/kg and 432 kJ/kg for nickel-metal hydride and lithium-ion batteries respectively (Sclater, 1999), compared to the specific energy of a composite hydraulic accumulator at approximately 6 kJ/kg (Pourmovahed, 1993). The consequence of this for applications such as hydraulic hybrid vehicles is a concession in the energy storage capacity based on packaging and weight considerations. The limited energy storage is a barrier to technologies such as “plug-in” hydraulic hybrid vehicles that can operate for a considerable distance solely on energy storage.

Previous research on improving the energy density of hydraulic accumulators has primarily focused on isothermalizing the compression and expansion of the gas in the accumulator. Successful results have been obtained by adding foam (Otis, 1973; Pourmovahed et al., 1988; Pourmovahed and Otis, 1990) or fine metallic strands to the gas volume (Sherman and Karlekar, 1973). These approaches have provided incremental increases in the energy density of hydraulic accumulators, yet the energy density is still orders of magnitude lower than competing technologies.

The previously proposed open accumulator is a revolutionary approach to improving the energy density of hydraulic systems with theoretical improvements of an order of magnitude (Li et al., 2007). Conceptually, the open accumulator allows either air or oil to be added or removed from the accumulator, where the air passes through an air motor/compressor that is coupled to a hydraulic pump/motor. The energy density of compressed air is significantly higher than the gas in a conventional accumulator because of the higher pressure ratio. For example, the hydraulic accumulator in a hydraulic hybrid vehicle might operate between 35 MPa and 17.5 MPa, creating a pressure ratio of 2:1. In a compressed air system, the air expands in an air motor from 35 MPa to atmospheric pressure, creating a pressure ratio of 350:1. By allowing either hydraulic fluid or compressed air to enter or exit the open accumulator, the hydraulic system pressure can be controlled and is no longer a function of the quantity of energy stored. Implementing the open accumulator concept with reasonable efficiency is quite challenging. The prime

This manuscript was received on 21 May 2009 and was accepted after revision for publication on 14 October 2009

challenges surround managing the heat transfer and air sealing at high pressure ratios to avoid extremely high temperatures during compression and extremely low temperatures causing icing during expansion (Li et al., 2007).

Building on an aspect of the open accumulator, the energy storage density of a hydraulic system can be increased with storage in other energy domains. There are numerous domains of kinetic and potential energy storage, including a few promising alternatives such as pneumatic, electric battery, flywheel, mechanical spring, and thermal. Important factors influencing the selection of an auxiliary energy storage domain include energy density, power density, energy conversion efficiency from hydraulic, and operational factors including safety and environmental concerns.

Due to the high specific energy at approximately 325 kJ/kg (Bitterly, 1998), extremely high power density, and negligible environmental concerns, this paper will focus on augmenting the hydraulic system with a flywheel for energy storage. This will be accomplished by integrating the hydraulic accumulator and the flywheel into a single energy storage unit. The benefit of this approach is the natural crossing of energy domains by using the hydraulic fluid as a means of changing the inertia of the flywheel. As will be discussed in the modeling section below, varying the inertia of the flywheel allows the hydraulic pressure to be controlled independently of the quantity of energy stored. This concept is similar to the interaction of the pneumatic and hydraulic systems in the open accumulator to maintain near constant pressure.

Previous works have explored variable inertia flywheels using a variety of means. Harrowell discussed the use of a vulcanized rubber elastomeric flywheel; across a limited range of energy storage, the angular velocity is approximately constant due to the centripetal acceleration causing a radius strain in the material (Harrowell, 1994). Numerous works have utilized moving solid masses to change the inertia of a flywheel including flyball governors, masses on tracks (Leung, 1991), and the band-style variable inertia flywheel (Moosavi-Rad, 1995; Ullman and Velkoff, 1977). Multiple patents have addressed using fluid to change the inertia of a flywheel, such as (Burstall, 2005; Lewis, 1966; Wagner, 1992). To the author's knowledge, no previous work has integrated a flywheel and a hydraulic accumulator to minimize fluctuation in hydraulic system pressure and increase the energy density.

This paper will develop a model of the energy flows in an integrated flywheel-accumulator. The model will be used to compare different modes of energy storage. The results of the energy storage simulations will then be presented and discussed. To provide further insight, the energy storage efficiency due to losses in the hydraulic pump/motor will be discussed, followed by concluding remarks.

2 Method of Approach

There are multiple ways to integrate a flywheel and a hydraulic accumulator including radially spaced chambers acting as bladder, piston, or diaphragm accumulators and a single chamber rotated about the central axis with bladder, diaphragm, piston, or no separation between the gas and the liquid. This paper will focus on a single cylindrical piston-style accumulator rotated about its central axis because this design maintains a cylindrical body of hydraulic fluid, unlike flexible separation methods which result in varying geometry based on the centripetal acceleration of the fluids. As seen in Fig. 1, the flywheel-accumulator is coupled to a pump/motor. Hydraulic fluid enters and exits the flywheel-accumulator at the center of one end of the cylinder. On the opposing side of the piston is nitrogen gas at a precharged pressure.

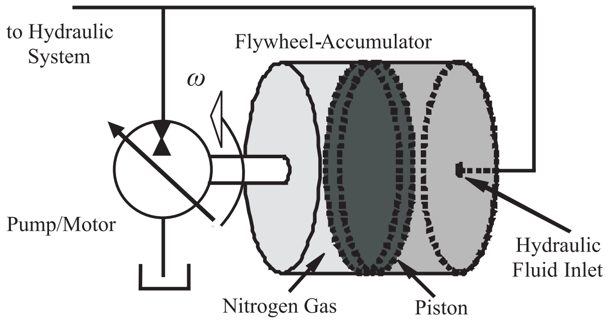

Fig. 1: *Diagram of the flywheel-accumulator system. The pump/motor is coupled to the flywheel directly or through a geared connection. Hydraulic fluid enters the center of the accumulator on the opposite side of the pump/motor*

While the scope of this paper is to present the flywheel-accumulator concept and the increase in energy density that the technology offers, a brief discussion of practical design considerations is required. One issue critical to the modeling section below is fluid swirl, defined as a difference between the angular velocity of the fluid and the flywheel shell. Fluid swirl results in significant viscous energy loss and a reduction in the effective inertia and will be minimized with interlocking or telescoping baffles. Other design issues include a high-speed live swivel to permit the fluid inlet to the center of the device, dynamic balancing, safety, piston sealing, and stress and strain of the device. The discussion section will further address these issues.

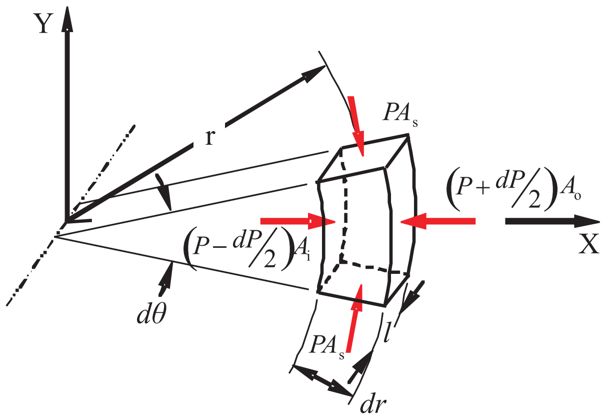

Fig. 2: *Free body diagram of an infinitesimal element of the hydraulic fluid subject to centripetal acceleration in the flywheel-accumulator. This acceleration develops a pressure gradient that varies with the radius*

As in a conventional hydraulic accumulator, when hydraulic fluid enters the flywheel-accumulator, the piston moves axially and the nitrogen gas compresses. Because the density of the nitrogen gas is low, the centrifugal force created by the rotation of the flywheel has little influence on the gas pressure distribution. However, the centrifugal force created by the flywheel rotation does have a significant influence on the radial pressure distribution of the hydraulic fluid, due to the higher density. A free-body-diagram of an infinitesimal fluid element is provided in Fig. 2 to aid in developing the equations to describe the pressure distribution in the hydraulic fluid.

Starting from Newton's second law and assuming steady-state operation with no fluid swirl, the force balance along the positive x-axis is described by:

$$\sum F_{\mathrm{x}} = ma \tag{1}$$

where the forces are due to the pressure acting on each side of the infinitesimal fluid volume, m is the mass of the fluid volume, and a is the centripetal acceleration of the fluid volume. Note that during transient operation, tangential, coriolis, and sliding acceleration are also present. Substituting the forces on the fluid volume yields:

$$\left(P-\frac{dP}{2}\right)A_{\mathrm{i}}-\left(P+\frac{dP}{2}\right)A_{\mathrm{o}}+2PA_{\mathrm{s}}\frac{d\theta}{2}=ma \tag{2}$$

where P is the fluid pressure, dP is the change in pressure across the element, $d\theta$ is the arc angle of the volume element, A_{i} is the area of the volume element on the inner radius, A_{o} is the area of the fluid element on the outer radius, and A_{s} is the area of the top and bottom sides of the element. Note that the sine of a small angle, $d\theta/2$, has been substituted by the angle. Substituting the area of the surfaces into Eq. 2 yields:

$$\begin{aligned}\rho V\left(-\omega^2 r\right)=&\left(P-\frac{dP}{2}\right)\left(r-\frac{dr}{2}\right)d\theta\, l-\\&\left(P+\frac{dP}{2}\right)\left(r+\frac{dr}{2}\right)d\theta\, l+2Pdr\, l\frac{d\theta}{2}\end{aligned} \tag{3}$$

where r is the radius to the center of the element, dr is the radial length of the element, l is the thickness of the element, ρ is the mass density, V is the volume of the element, and ω is the angular velocity of the flywheel. The volume of the element can be expressed as:

$$V = r\, d\theta\, dr\, l\ . \tag{4}$$

Expanding and simplifying Eq. 3 and taking the integral of both sides yields:

$$\int_{P_{\mathrm{S}}}^{P} dP = \int_{0}^{r} \rho\omega^2 r dr \tag{5}$$

where P_S is the system pressure at the inlet where $r = 0$, yields an equation for the pressure of the fluid as a function of the radius.

$$P(r)=\frac{\rho\omega^2 r^2}{2}+P_{\mathrm{S}} \tag{6}$$

The above demonstrates that the hydraulic system pressure is a function of the angular velocity of the flywheel. To solve for the system pressure, a force balance analysis will be conducted on the accumulator piston. To aid in this force balance, Fig. 3 presents the pressure distributions on both sides of the piston. The parabolic pressure distribution on the hydraulic fluid side of the piston is described by Eq. 6, while a uniform pressure distribution is assumed for the gas side of the accumulator.

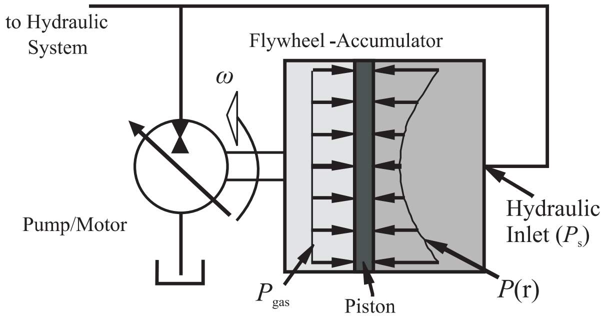

Fig. 3: *Pressure distributions of the gas and hydraulic fluid in the accumulator. Due to the centripetal acceleration and density of the hydraulic fluid, a parabolic pressure distribution is formed*

The forces on the piston, defined by the fluid pressures acting across the piston area, must be equal for the piston to be in equilibrium, as described by:

$$\int P(r)\,dA=\int_0^{r_{\mathrm{o}}}\left(\frac{\rho\omega^2 r^2}{2}+P_{\mathrm{S}}\right)\left(2\pi r dr\right)=P_{\mathrm{gas}}A_{\mathrm{piston}} \tag{7}$$

where r_o is the outer radius of the accumulator piston, P_{gas} is the pressure of the compressed gas, and A_{piston} is the area of the piston. By making the assumptions of an ideal gas and isothermal compression, the pressure of the gas can be further described as:

$$P_{\mathrm{gas}}=P_{\mathrm{charge}}\frac{V_{\mathrm{charge}}}{V_{\mathrm{gas}}}=P_{\mathrm{charge}}\frac{l_{\mathrm{g}}+l_{\mathrm{h}}}{l_{\mathrm{g}}} \tag{8}$$

where P_{charge} is the gas precharge pressure, V_{charge} is the volume of the gas during charging, V_{gas} is the current volume occupied by the gas, l_{g} is the axial length of the gas volume, and l_{h} is the axial length of the hydraulic fluid volume. The isothermal behavior assumption in the hydraulic accumulator is justified when an elastomeric foam is added to the gas chamber, as previously described by Otis and Pourmovahed (Otis and Pourmovahed, 1984). Finally, substituting Eq. 8 into Eq. 7, integrating, and solving for P_S yields:

$$P_{\mathrm{S}}=P_{\mathrm{gas}}-\frac{\rho\omega^2 r_{\mathrm{o}}^2}{4}=P_{\mathrm{charge}}\frac{l_{\mathrm{g}}+l_{\mathrm{h}}}{l_{\mathrm{g}}}-\frac{\rho\omega^2 r_{\mathrm{o}}^2}{4} \tag{9}$$

When hydraulic fluid is added to the flywheel-accumulator, the mass moment of inertia increases. Assuming the density of the hydraulic fluid is constant with radius, the mass moment of inertia of the flywheel-accumulator is described by:

$$I=I_{\mathrm{flywheel}}+I_{\mathrm{fluid}}=I_{\mathrm{flywheel}}+\frac{mr_{\mathrm{o}}^2}{2} \tag{10}$$

where I_{flywheel} is the inertia of the empty flywheel, I_{fluid} is the inertia of the hydraulic fluid, and m is the mass of the hydraulic fluid. Substituting for the mass of the fluid in terms of the density and volume yields:

$$I = I_{\text{flywheel}} + \frac{\pi \rho l_h r_o^4}{2} \tag{11}$$

The model will use the finite-difference method to simulate the energy flows in the system. For a preliminary analysis, the energy losses in the hydraulic motor will be neglected, allowing the torque of the hydraulic motor in the on-state to be described by:

$$T = \frac{P_S D}{2\pi} \tag{12}$$

where D is the displacement of the pump/motor in units of m^3/rev. The volumetric displacement of the idealized hydraulic pump/motor is described by:

$$Q_{P/M} = \frac{D}{2\pi}\omega \tag{13}$$

The volume of hydraulic fluid displaced during time step *i*, when the motor is on is described by:

$$V_{P/M}^{i} = Q_{P/M}\Delta t \tag{14}$$

where Δt is the time step. Using the volume displaced by the hydraulic motor, the length of the gas volume in the accumulator is found by:

$$l_g^{i} = l_g^{i-1} + \frac{V_{P/M}^{i} - Q^{i}\Delta t}{\pi r_o^2} \tag{15}$$

where l_g^{i} and l_g^{i-1} are the thickness of the gas volume in the accumulator at the current and previous time steps and Q^{i} is the volumetric flow rate into the entire energy storage device. The simulation will assume a constant regenerative power, thus the flow rate is described by:

$$Q^{i} = \frac{Power}{P_S^{i}} \tag{16}$$

Both the moment of inertia and the angular velocity of the flywheel-accumulator change at the same time, requiring the basic rotational form of Newton's second law, which states that the change in angular momentum is equal to the applied torque:

$$T = \frac{d(I\omega)}{dt} = \dot{I}\omega + \dot{\omega}I \tag{17}$$

By discretizing the above equation using the finite difference method, the angular velocity at the next time step is:

$$\omega^{i} = \frac{T\Delta t + I^{i}\omega^{i-1}}{2I^{i} - I^{i-1}} \tag{18}$$

3 Simulation

Simulations of the energy flows in the flywheel-accumulator will be used to further illustrate different modes of energy storage. The simulation situation is a constant power regenerative event such as braking in a hydraulic hybrid vehicle, shown in Fig. 4, or lowering a load in the bucket of an excavator. While energy storage is discussed, energy extraction from the flywheel-accumulator follows the same principles. The purpose of the simulations is to demonstrate the energy storage capacity of the flywheel-accumulator and thus the energy losses in the system will temporarily be ignored. Three modes of storing energy in the flywheel-accumulator will now be discussed.

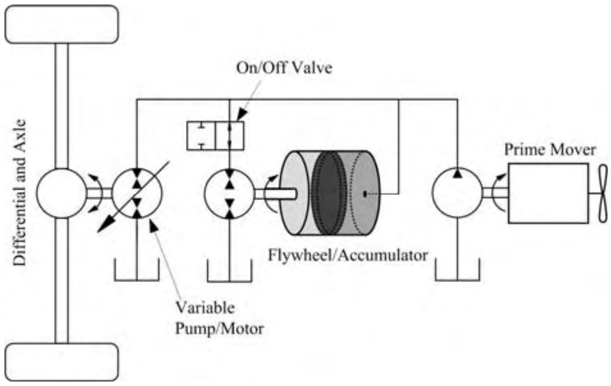

Fig. 4: *Flywheel-Accumulator implemented in a series hydraulic hybrid vehicle drive train. Note that closing the on-off valve causes the energy storage device to function as a conventional hydraulic accumulator*

Mode 1: For the baseline case, purely pneumatic energy storage will be considered where hydraulic fluid is directly added to the flywheel-accumulator. For this mode, the on-off valve shown in Fig. 4 is closed and the displacement of the pump/motor coupled to the flywheel is set to zero to remove any influence of the pump/motor. This is the most efficient way to store energy in the system as it requires no conversion across energy domains; however it provides limited energy density, which is the motivation for this work. In reference to Fig. 1, when hydraulic fluid is added to the flywheel-accumulator, the piston moves to the left and the nitrogen gas is compressed, increasing the gas pressure. As described by Eq. 11, the addition of hydraulic fluid to the flywheel-accumulator increases the inertia. If no external torque is applied to the flywheel, the increase in inertia results in a decrease in angular velocity due to conservation of angular momentum.

Mode 2: The second mode of operation will consider purely rotational kinetic energy storage. In this operation mode, no hydraulic fluid is added or removed from the flywheel-accumulator, maintaining a constant moment of inertia; instead the hydraulic motor is used to increase the angular velocity of the flywheel. To accommodate variations in the angular velocity of the flywheel, a variable displacement pump/motor is required. As described by Eq. 9, an increase in the angular velocity of the flywheel-accumulator decreases the hydraulic pressure of the system, decreasing the potential power of the hydraulic system. This mode of operation does introduce greater energy loss than the first mode as energy is converted from the hydraulic domain to rotating mechanical domain. However, the energy storage capacity is much greater in the rotating mechanical domain than the purely hydraulic domain.

Mode 3: By combining the first two energy storage modes, high energy density and near constant hydraulic pressure can be achieved. Multiple control schemes can be used to store energy in the two domains. For demonstration purposes, a simple approach will be implemented that uses a fixed displacement pump/motor and seeks to maintain the hydraulic pressure within a specific range. When the hydraulic pressure exceeds a high threshold, the valve to the fixed displacement motor is switched on, resulting in the motor increasing the rotating kinetic energy of the flywheel-accumulator. Because the motor has a fixed displacement, when the valve is on, hydraulic fluid may enter or exit the accumulator. When the hydraulic pressure drops below a low threshold, the valve to the hydraulic motor is switched off, and again all energy is stored in pure hydraulic form until the pressure again exceeds the high threshold.

The simulations will begin with a low state of energy charge. The simulations will then be run until either the maximum pressure and/or the maximum angular velocity of the flywheel-accumulator is reached. The maximum pressure of the system will be 35 MPa (5000 psi), typical of high power hydraulic systems. Previous work found that the maximum energy density of a conventional accumulator, assuming isothermal operation, is achieved at an expansion ratio of 2.15 (Li et al., 2007), thus the precharge of the accumulator will be 16 MPa. The maximum angular velocity of the flywheel will be 25,000 rpm. To achieve this velocity, a speed reduction to the hydraulic pump/motor will be used.

The simulations of the flywheel-accumulator will be modestly sized at 10 liters. The dimensions selected for the cylindrical carbon fiber composite flywheel-accumulator are a length of 0.15 m, a radius of 0.145 m, and a wall thickness of 20 mm. The predicted mass of the energy storage device when empty is approximately 6.7 kg with a mass moment of inertia of 0.236 kg m^2. The input power to the system is set at a constant 25 kW. The common system parameters can be found in Table 1, while additional parameters and initial conditions for each operation mode are presented the results section. While some ambiguity is involved in the selection of the values in Table 1, the values approximate an appropriate size device for a hydraulic hybrid vehicle.

Table 1: System parameters for the simulation

Parameter	**Variable**	**Value**	**Units**
Power into the energy storage device	Power	25	kW
Radius of flywheel	r_o	0.145	m
Axial thickness of flywheel	l	0.15	m
Inertia of empty flywheel	I_{flywheel}	0.236	kg m^2
Gas precharge in accumulator	P_{charge}	16	MPa
Density of hydraulic fluid	ρ	876	kg/m^3
Gear increase from pump/motor	R	7	unitless
Time step	Δt	1	ms

4 Results

The computational model was run for the three different energy storage modes discussed above. For each simulation, the primary results of interest are the system pressure, moment of inertia, and angular velocity during the energy storage event, as well as the total energy stored.

Mode 1: For a baseline comparison, the first mode considers only pneumatic energy storage. Throughout the simulation of this mode, the angular velocity of the flywheel is zero. At the start of the simulation, the flywheel-accumulator has no hydraulic fluid. Charging the 10 liter accumulator to maximum pressure, as seen in Fig. 5a, takes 4.97 seconds with total energy storage of 124.1 kJ. As the accumulator charges, the moment of inertia, seen in Fig. 5b, increases as the length of the gas volume in the accumulator decreases from 0.15 m to 0.067 m.

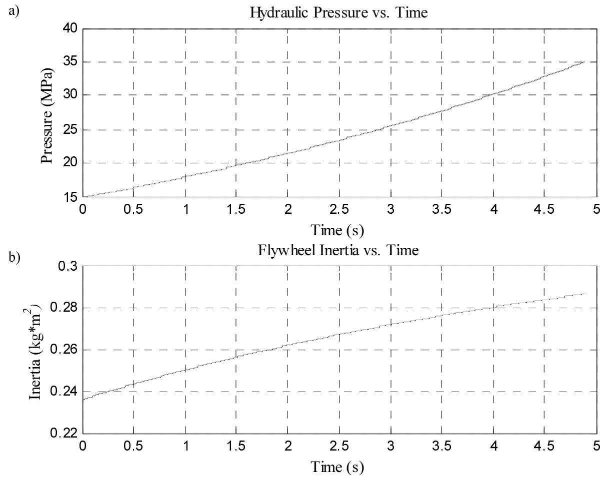

Fig. 5: *System pressure and flywheel inertia during the mode 1 operation where the unit is used for purely pneumatic energy storage*

Mode 2: In the second mode of operation, all of the energy is stored in purely kinetic form and no hydraulic fluid is added to the accumulator. The hydraulic power coming into the system is converted into kinetic energy with a variable displacement hydraulic motor with a maximum displacement of 70 cm^3/rev. The initial angular velocity of the flywheel-accumulator is set to 500 rad/s to achieve adequate power density from the hydraulic motor. For this simulation, the length of the gas volume is fixed at 0.067 m, which is the final condition of the mode 1 simulation. For this mode, the displacement of the hydraulic motor will be continuously varied such that all of the input power is absorbed by the hydraulic motor and no fluid is added or removed from the device.

Using purely kinetic energy storage, the flywheel-accumulator stores 957.2 kJ of energy during a 38.3 second period before the angular velocity exceeds 25,000 rev/min. As expected, the angular velocity, seen in Fig. 6a, increases with time. The influence of the centripetal acceleration on the hydraulic fluid is apparent in Fig. 6b, a plot of the system pressure with time. Recall that the system pressure is defined by the pressure at the center hydraulic inlet of the flywheel-accumulator. From the figure it can be noted that the system pressure ceases to decrease near 6 MPa. At low pressure, the power density of the hydraulic motor decreases, preventing the kinetic energy storage from absorbing all of the input power.

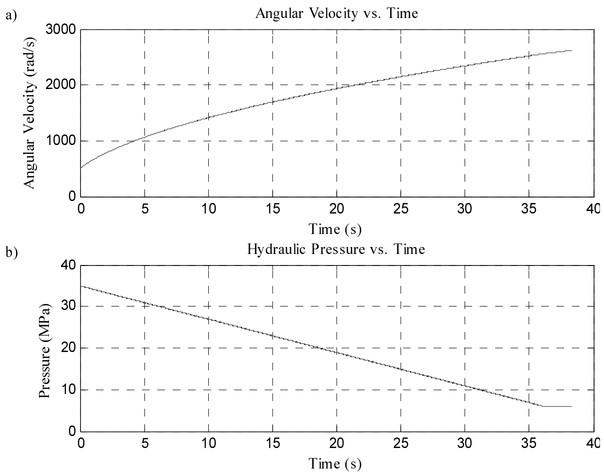

Fig. 6: *Angular velocity and hydraulic pressure of the flywheel accumulator during mode 2, the purely kinetic energy storage operation. Near 6 MPa the hydraulic motor becomes power limited, causing the pressure plot to level off during the final seconds of the simulation*

Mode 3: The third mode of operation uses both pneumatic and kinetic energy storage. The operating parameters and initial conditions for this mode are presented in Table 2. For this simulation the hydraulic motor is treated as a fixed displacement unit and switched on and off based on high and low system pressure thresholds. As with the purely pneumatic energy storage case, the simulation is started with the accumulator empty of hydraulic fluid.

Table 2: The operating parameters and initial conditions of mode 3 energy storage simulation

Parameter	Variable	Value	Units
Pump/motor displacement	D	42	cm^3/rev
Initial angular velocity of flywheel	ω_i	500	rad/s
Initial thickness of the gas volume	$l_{g,i}$	0.15	m
High pressure threshold	P_{high}	32	MPa
Low pressure threshold	P_{low}	25	MPa

The mode 3 simulation results in 1208.6 kJ of energy storage during a period of 48.3 seconds. Plots of the pressure, angular velocity, and moment of inertia can be found in Fig. 7. Due to the centripetal acceleration of the hydraulic fluid causing a decrease in the hydraulic system pressure, more hydraulic fluid can be added to the flywheel-accumulator than in the first mode where the flywheel is at rest. The length of the gas volume at the end of the simulation is 0.038 m.

The on-off operation of the fixed displacement motor is evident in the ripples in all three plots shown in Fig. 7. To minimize energy losses during motor switching, it is desirable to decrease the frequency of the ripples. The time that the motor is on can be increased by decreasing the motor displacement and increasing the high pressure threshold. This can be accommodated by turning the motor on at the start of the simulation. As before, the motor is switched off when the pressure drops below the low threshold and then operates in the standard manner. Using this modified control method, the motor displacement can be decreased from 42 cc/rev to 28 cc/rev and the high pressure threshold can be increased to 34 MPa. A plot of the system pressure using this method can be found in Fig. 8.

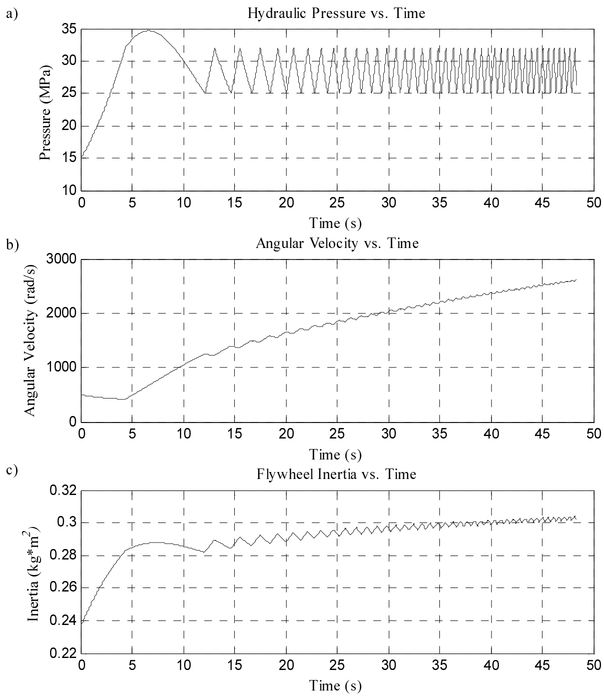

Fig. 7: *The system pressure, flywheel angular velocity, and flywheel inertia during the mode 3 simulation*

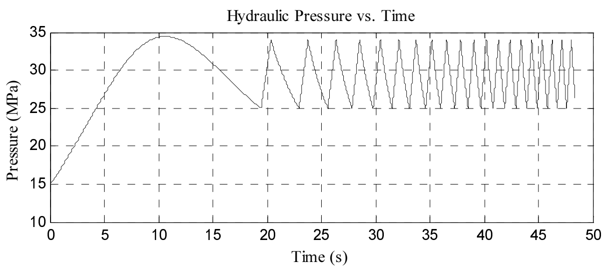

Fig. 8: *Plot of the system pressure using a modification of the mode 3 simulation where the motor displacement is decreased and the high pressure threshold is increased for the purpose of decreasing the frequency of switching the motor and off*

For comparative purposes, a summary of the results of the three main operating modes is presented in Table 3.

Table 3: Summary of the results of the three energy storage operating modes

Parameter	**Mode 1**	**Mode 2**	**Mode 3**
Initial pressure (MPa)	16.0	35.0	14.8
Final pressure (MPa)	35.0	5.9	26.2
Initial moment of inertia (kg m^2)	0.236	0.287	0.236
Initial moment of inertia (kg m^2)	0.286	0.287	0.304
Operating time (seconds)	5.0	38.3	48.3
Pneumatic energy storage (kJ)	124.1	0	169.6
Rotating kinetic energy storage (kJ)	0	957.2	1039.0
Total energy stored (kJ)	124.1	957.2	1208.6

5 Discussion

The results of the simulation demonstrate the increase in energy storage potential of the flywheel-accumulator. In comparison to a purely pneumatic storage, used in a conventional hydraulic accumulator, the combined pneumatic and rotating kinetic energy storage increased the energy storage from 124 kJ to 1209 kJ, approximately a 10X increase in energy storage for the same volume. While the flywheel-accumulator used in the simulation had an internal volume of 10 liters, it must be noted that the total system volume for either the pneumatic or the pneumatic-kinetic system would be significantly larger. Additional major components that add to the volume of the system include a tank for the hydraulic fluid, the hydraulic pump/motor, and the vacuum chamber/containment vessel that is used to absorb energy in case of a failure. Because the volume of these components is not specified, the results should be used to compare the various energy storage modes, but not for calculation of the energy density or specific energy of the system.

An important consideration for any energy storage system is the maximum potential power. In a hydraulic system, the power is directly proportional to the system pressure. In purely pneumatic operation, mode 1, the system pressure increases with increasing energy storage. The change in pressure as a function of energy creates control challenges for applications such as hybrid vehicles. Furthermore, at low levels of energy storage, the potential power of the system is limited.

Using the purely rotating kinetic energy storage, mode 2, the hydraulic system pressure decreases with increasing energy storage due to the centripetal acceleration on the hydraulic fluid in the flywheel-accumulator. The decrease in potential power with increasing energy storage is apparent in Fig. 6b where the hydraulic motor becomes power limited due to the decreasing system pressure. Note that the centripetal force on the hydraulic fluid in the flywheel creates a pressure gradient that increases with radius. If the hydraulic fluid were removed from the outer rim, similar to a centrifugal pump, the hydraulic system pressure would increase with increasing energy storage. If purely kinetic energy storage is desired, a constant inertia flywheel with a hydro-static transmission would be a better choice to decouple the hydraulic pressure from the angular velocity of the flywheel.

Combining the pneumatic and rotational kinetic energy storage, mode 3, allows the hydraulic system pressure to be independent of the energy storage. In the simulation, the hydraulic system pressure was controlled by switching on and off a fixed displacement hydraulic motor. If desired, a constant hydraulic system pressure could be achieved regardless of energy storage by using a variable displacement pump/motor. Another option is to make a fixed displacement pump/motor have virtually variable displacement through the use of a switch-mode hydraulic circuit as previously described by the author (Van de Ven and Katz, 2008). Integrating pneumatic and rotational kinetic energy storage allows the displacement of the pump/motor to be decreased to meet the average power requirements, while hydraulic fluid can be directly added or removed from the accumulator to meet high power transient requirements. This is demonstrated by using the 28 or 42 cc/rev motor for mode 3, while a 70 cc/rev motor was required for mode 2 operation.

Another interesting benefit of the combined pneumatic and rotational kinetic energy storage is the increase in quantity of both forms of energy storage over the individual systems. The simulation demonstrated a 37% increase in the pneumatic energy storage from mode 1 to mode 3 and a 9 % increase in the rotating kinetic energy storage from mode 2 to mode 3. The quantity of pneumatic energy stored increases because the gas can be compressed to a higher pressure for the same hydraulic system pressure due to the centripetal acceleration on the hydraulic fluid. The kinetic energy storage capacity increases because of the increase in the moment of inertia caused by the addition of hydraulic fluid to the flywheel-accumulator.

The energy storage density and pressure management of the flywheel-accumulator are promising, but the concept introduces multiple challenges. First, the dynamic balance of the flywheel-accumulator is critical for smooth operation at high speed. This requires tight tolerance control during manufacturing and likely a post-manufacturing balancing procedure. Second, for operation in mobile applications, changing the pitch, yaw, or roll of the vehicle creates a gyroscopic torque. The gyroscopic torque can be minimized through mounting the flywheel to rotate about the vertical axis so changes in yaw to not result in an applied torque. Alternatively gyroscopic torque can be eliminated by using two flywheels rotating in opposite directions or mounting a single flywheel in a gimbal. Third, as with any form of energy storage, in the event of a failure, the flywheel-accumulator can release energy in a dangerous manner. This hazard can be mitigated by using the vacuum chamber around the flywheel as a containment unit and designing of the flywheel to fail in an energy dissipating manner.

Three additional design challenges of the flywheel-accumulator, briefly discussed in the methods section, relate to fluid swirl, rim stress, and the hydraulic fluid inlet. During angular acceleration of the flywheel, there is a tendency for the fluid in the flywheel to remain at the previous angular velocity, creating a velocity gradient in the fluid. The shearing of the fluid due to this velocity gradient results in an energy loss. To alleviate this issue, baffles are required inside the flywheel-accumulator to prevent fluid swirl. Possible designs that allow piston movement are interlocking or telescoping baffles. A second design challenge results from the rim stress in the flywheel created by the internal fluid pressure and the centripetal acceleration of the rim. To provide perspective, the hydraulic fluid pressure acting on the rim can easily be found by substituting Eq. 10 into Eq. 7 and solving where $r = r_o$. Using the maximum hydraulic system pressure and angular velocity of 35 MPa and 25,000 rev/min respectively, the gas pressure is 63 MPa and the hydraulic fluid pressure at the rim is 98 MPa. Third, because hydraulic fluid enters the center of the rotating flywheel-accumulator, a live swivel is required. The combination of high pressure and high angular velocity make the design of this swivel particularly challenging.

Up to this point, the energy losses in the system have been neglected for the purpose of focusing on the energy storage capacity. Primary sources of energy loss in the flywheel-accumulator storage device include the hydraulic motor volumetric and mechanical losses, bearing friction, seal friction, aerodynamic drag of the flywheel rotor, and vacuum pumping losses. High speed flywheel research has addressed many of these challenges through avenues such as magnetic bearings (Genta, 1985; Little and Palazzolo, 2005) and the study of flywheel aerodynamics as a function of vacuum pressure (Genta, 1985; Saint Raymond et al., 2008). To illustrate the energy losses of converting hydraulic energy to rotational kinetic energy, efficiency equations will be developed for the hydraulic motor.

The efficiency of a fixed displacement pump or motor is dependent on the angular velocity, system pressure, oil viscosity, and coefficients based on the characteristics of the pump or motor. Using linearized efficiency equations developed by McCandlish and Dorey, Eq. 13, the torque applied to the flywheel-accumulator by the hydraulic motor , is replaced by (Mccandlish and Dorey, 1984):

$$T = \frac{P_S D}{2\pi}\left[1 - C_v\left(\frac{\mu\omega}{P_S}\right) - C_f\right] \tag{19}$$

where C_v is the coefficient of viscous drag, μ is the absolute viscosity of the hydraulic fluid, and C_f is the coefficient of coulomb friction. Similarly, using the linearized efficiency model, Eq. 14, the volumetric displacement of the hydraulic pump/motor, is replaced by (Mccandlish, D. and Dorey, R. E., 1984):

$$Q_{P/M} = \frac{\omega D}{2\pi}\left[1 + C_s\left(\frac{P_S}{\mu\omega}\right) + \left(\frac{P_S}{B}\right)(V_r + 1)\right] \tag{20}$$

where C_s is the coefficient of slip, B is the bulk modulus of the hydraulic fluid, and V_r is the volume ratio of hydraulic unit, defined as the clearance volume divided by the swept volume. Values for the coefficients of a specific pump or motor can be obtained using a least-squares fit of experimental or manufacturer's data. Values for a typical low-grade axial piston unit can be found in Table 3. The total efficient, defined as the mechanical efficiency multiplied by the volumetric efficiency, of the selected motor peaks at 90.3 %, seen in Fig. 9.

Table 3: Parameters used in estimating the mechanical and volumetric efficiency of the hydraulic pump/motor. These values represent a typical axial piston device

Parameter	Variable	Value	Units
Absolute viscosity of hydraulic fluid	μ	0.012	N s/m^2
Bulk modulus of hydraulic fluid	B	1.72	GPa
Coefficient of slip	C_s	1.88×10^{-9}	unitless
Coefficient of viscous drag	C_v	4.91×10^{5}	unitless
Coefficient of coulomb friction	C_f	0.024	unitless
Volume ratio of hydraulic unit	V_r	0.25	unitless

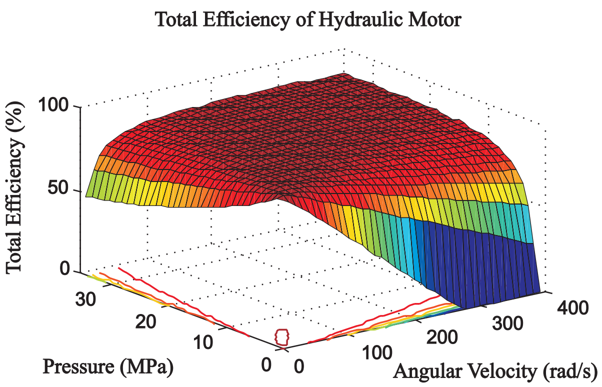

Fig. 9: *Total efficiency of the hydraulic motor used in the simulation*

The torque and flow rate equations in the simulation were replaced with the modified linear efficiency equations presented above. When applied to mode 3 operation, the combined pneumatic and rotating kinetic energy storage, the average pump motor efficiency during the simulation is 78 %. Recall that the simulation uses an on-off control strategy with a fixed displacement motor. If a variable displacement unit were used instead of the fixed displacement unit, the efficiency equations developed by McCandlish and Dorey need to be modified based on the current pump/motor displacement (Mccandlish and Dorey, 1984).

6 Conclusion

In this paper a novel flywheel-accumulator for increasing the energy storage in hydraulic systems was presented. By integrating a flywheel and an accumulator, energy can be stored in both pneumatic and rotating kinetic form. A novel feature of the system is the ability to change the flywheel inertia by adding or removing hydraulic fluid from the flywheel-accumulator. Through a numerical simulation, the energy storage capacity of the flywheel-accumulator system was demonstrated to be approximately 10 times greater than a conventional accumulator occupying the same volume. Furthermore, because energy can be stored in two domains, the hydraulic system pressure becomes independent of the quantity of energy stored, allowing the potential power of the hydraulic system to remain constant.

The significant increase in energy storage density and the ability to manage the hydraulic pressure make the flywheel-accumulator an enabling technology. A logical application for the technology is hydraulic hybrid vehicles where the low energy density of conventional hydraulic accumulators has been a barrier to compete with electric hybrid systems. The flywheel-accumulator could be coupled to the internal combustion engine, directly storing energy from the engine in the flywheel. The significant increase in energy density also enables a plug-in version of a hydraulic hybrid vehicle where the flywheel is charged from grid energy, allowing the vehicle to drive a limited distance without operating an internal combustion engine. Additional applications for the flywheel-hybrid include regeneration in construction, mining, agriculture, and industrial equipment that heavily rely on hydraulic systems. Other alternative energy applications for the technology include storing energy for wind turbines, tidal energy, wave energy, or other areas requiring energy storage.

The flywheel-accumulator presents numerous opportunities for future research and development. While there are numerous challenges for this technology, none of them appear insurmountable. This promising technology has the potential to be transformative for numerous applications.

Nomenclature

a	Acceleration of the fluid volume	[m/s^2]
A_i	Area of a fluid volume element on the inner surface	[m^2]
A_o	Area of a fluid volume element on the outer surface	[m^2]
A_{piston}	Cross-sectional area of the piston	[m^2]
A_s	Area of a fluid volume element on the top and bottom surfaces	[m^2]
B	Bulk modulus of the hydraulic fluid	[Pa]
C_f	Coefficient of coulomb friction in the hydraulic pump/motor	-
C_s	Coefficient of slip in the hydraulic pump/motor	-
C_v	Coefficient of viscous drag in the hydraulic pump/motor	-
D	Displacement of the pump/motor	[m^3/rev]
dP	Change in pressure across a fluid element	[Pa]
dr	Radial length of a fluid element	[m]
$d\theta$	Arc angle of a fluid volume element	[radians]
F_X	Forces acting on the fluid element in the X-direction	[N]
I	Mass moment of inertia of the flywheel and hydraulic fluid	[kgm^2]
I_{fluid}	Mass moment of inertia of the hydraulic fluid	[kgm^2]
$I_{flywheel}$	Mass moment of inertia of the empty flywheel	[kgm^2]
l	Thickness of a fluid element	[m]
l_g	Axial length of the gas volume	[m]
l_h	Axial length of the hydraulic fluid volume	[m]
l_g^i	Thickness of the gas volume in the accumulator at the current time step	[m]
l_g^{i-1}	Thickness of the gas volume in the accumulator at the previous time step	[m]
m	Mass of the fluid volume	[kg]
P	Absolute pressure of the fluid	[Pa]
P_{charge}	Gas precharge pressure	[Pa]
P_{gas}	Pressure of the compressed gas	[Pa]
$Power$	Regenerative power	[W]
P_S	System pressure at r = 0	[Pa]
Q^i	Volumetric flow rate into the accumulator at the current time step	[m^3/s]
$Q_{P/M}$	Volumetric flow rate of the pump/motor	[m^3/s]
r	Radius to the center of a fluid element	[m]
r_o	Outer radius of the piston	[m]
T	Torque of the hydraulic pump/motor	[Nm]
V	Volume of a fluid element	[m^3]
V_{charge}	Gas volume at precharge pressure	[m^3]
V_{gas}	Current volume of the gas	[m^3]
$V_{P/M}^i$	Volume of hydraulic fluid displaced by the pump/motor during the current time step	[m^3]
V_r	Volume ratio of hydraulic pump/motor unit	-
Δt	Time step	[s]
μ	Absolute viscosity of the hydraulic fluid	[Ns/m^2]
ρ	Mass density of the fluid	[kg/m^3]
ω	Angular velocity of the flywheel	[rad/s]

References

Bitterly, J. G. 1998. Flywheel technology: Past, present, and 21st century projections. *Aerospace and Electronic Systems Magazine*, Vol. 13, No. 8, pp. 13-16.

Burstall, O. W. J. 2005. *Variable inertia flywheel*. Perkins Engine Company Limitted, United States Patent 6,883,399.

Genta, G. 1985. *Kinetic energy storage; theory and practice of advanced flywheel systems*. Butterworths, London.

Harrowell, R. V. 1994. Elastomer flywheel energy store. *International journal of mechanical sciences*, Vol. 36, No. 2, pp. 95-103.

Katz, A. and **Van de Ven, J. D.** 2009. Design of a High-Speed On-Off Valve. *ASME International Mechanical Engineering Congress and Exposition*, Lake Buena Vista, FL.

Leung, T. T. 1991. Concept of a modified flywheel for megajoule storage and pulse conditioning. *IEEE Transactions on Magnetics*, Vol. 27, No. 1, pp. 403-408.

Lewis, O. G. 1966. *Variable inertia liquid flywheel*. Esso Re, United States Patent 3,248,967.

Li, P. Y., **Van de Ven, J. D.** and **Sancken, C.** 2007. Open accumulator concept for compact fluid power energy storage. *Proceedings of the ASME International Mechanical Engineering Congress* Seattle, WA. pp. 42580.

Little, F. E. and **Palazzolo, A.** 2005. Testing of magnetic bearings for flywheel energy storage in simulated space conditions. *International Energy Conversion Engineering Conference*, Vol 2, San Francisco, CA.

McCandlish, D. and **Dorey, R. E.** 1984. The mathematical modelling of hydrostatic pumps and motors. *Proceedings of the Institution of Mechanical Engineers, Part B*, Vol. 198, No. 10, pp. 165-174.

Moosavi-rad, H. 1995. A BVIF-integrated hybrid bus. *Proceedings of the Institution of Mechanical Engineers. Part D, Journal of automobile engineering*, Vol. 209, No. 2, pp. 95-101.

Otis, D. R. 1973. Thermal losses in gas-charged hydraulic accumulators. *Intersociety Energy Conversion Engineering Conference* Philadelphia, PA. pp. 198-201.

Otis, D. R. and **Pourmovahed, A.** 1984. Improving performance of gas-charged accumulators using elastomeric foam. *Symposium on Advanced and Hybrid Vehicles* Glasgow, Scotland.

Pourmovahed, A. 1993. Sizing energy storage units for hydraulic hybrid vehicle applications. *American Society of Mechanical Engineers, Dynamic Systems and Control Division*, Vol. 52, New Orleans, LA. pp. 231-246.

Pourmovahed, A., **Baum, S. A.**, **Fronczak, F. J.** and **Beachley, N. H.** 1988. Experimental evaluation of hydraulic accumulator efficiency with and without elastomeric foam. *Journal of Propulsion and Power*, Vol. 4, No. 2, pp. 185-192.

Pourmovahed, A. and **Otis, D. R.** 1990. An experimental thermal time-constant correlation for hydraulic accumulators. *Journal of Dynamic Systems, Measurement and Control*, Vol. 112, No. 3, pp. 116-121.

Saint Raymond, M., **Kasarda, M. E. F.** and **Allaire, P. E.** 2008. Windage power loss modeling of a smooth rotor supported by homopolar active magnetic bearings. *Journal of Tribology*, Vol. 130, No. 2, pp. 021101.

Sclater, N. 1999. *Electronic technology handbook*. McGraw-Hill, New York.

Sherman, M. P. and **Karlekar, B. V.** 1973. Improving the energy storage capacity of hydraulic accumulators. *Intersociety Energy Conversion Engineering Conference* Philadelphia, PA. pp. 202-207.

Ullman, D. G. and **Velkoff, H. R.** 1977. The variable inertia flywheel (vif), an introduction to its potential. *Flywheel Technology Symposium Proceedings* San Francisco, CA.

Wagner, J. T. 1992. *Energy storage flywheels using fluid transfer to vary moments of inertia*. United States Patent 5,086,664.

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