Fluid Power System Model-Based Design

Energy Efficiency

Fluid Power Systems

- Construction & Mining
- Manufacturing & Robotics
- Rail & Marine
- Agriculture & Forestry
- Automotive
- Materials Handling & Processing
- Medical & Dental
- Aeronautics & Aerospace

Energy in Fluid Power Systems

- Fluid Power Systems have many advantages:
  - High Power Density
  - Responsiveness and Bandwidth of Operation
  - High Accuracy and Precision
  - Stiffness
  - Reliable, Compact, Light Weight, and Flexible
- However, an area in need of significant improvement is Energy Efficiency. Hydraulic systems usually have poor energy utilization compared with electro-mechanical systems. Any technology which can boost efficiency of hydraulic machinery has potential for significant fuel savings.

- With the great variety of commercial hydraulic components currently available, enough design flexibility can be built into the system to allow a hydraulic positioning system to achieve high-precision tracking accuracy and to increase the system efficiency.
- The dual objectives for system design and control should then be high-precision motion control and energy efficient operation.
- Increasing the efficiency of any system means reducing total energy input required to perform a given task. The energy provided to a hydraulic system from a pump or other pressure or flow supply is:

$$E_s = \int_{t_0}^{t_1} p_s(\tau) Q_s(\tau) d\tau$$

- $p_s(t)$ is the supply pressure
- $Q_s(t)$ is the flow rate into the system
- This equation gives the energy provided to the system via the hydraulic fluid pressure and does not account for factors such as pump efficiency.
- It is clear that the energy used by the system can only be reduced if either a lower supply pressure is used or the flow into the system is reduced. One or both of these can be done if less energy is dissipated by the system.

- The main sources of energy dissipation in hydraulic systems are: mechanical friction, throttling losses in the valves, and leakage.

- Throttling loss is often the largest of the three components and can be most influenced by changing the control valve configuration and control algorithm.
- Throttling losses are caused by friction between the hydraulic fluid and the flow passage and by viscous shear forces within the fluid.
- Reducing the throttling loss in a hydraulic system may be thought of as making it easier for hydraulic fluid to flow, e.g., straightening and widening pipes and fully opening valves. The result is that the same amount of fluid can flow with a lower pressure drop. The flow path obstruction created by the valve results in permanent pressure (and energy) loss.

  - The power dissipated by oil flow through a valve, i.e., due to throttling, is the product of the flow rate and the pressure drop.
  - Thus the most efficient way to supply a given flow is with the smallest pressure drop possible.
- For hydraulic systems controlled by valves, some throttling losses are unavoidable. A valve-controlled system regulates the flow to actuators by changing the valve position. Some pressure drop across the valves is necessary for precision control of flow.

Technologies to Increase Fluid Power System Efficiency

- Examples of technologies which improve the energy utilization in hydraulic systems are:
  - Pump control
  - Load-sensing pumps
  - Independent metering valves
  - Regeneration flow
  - Accumulator-based energy recovery

Individual Pump Control

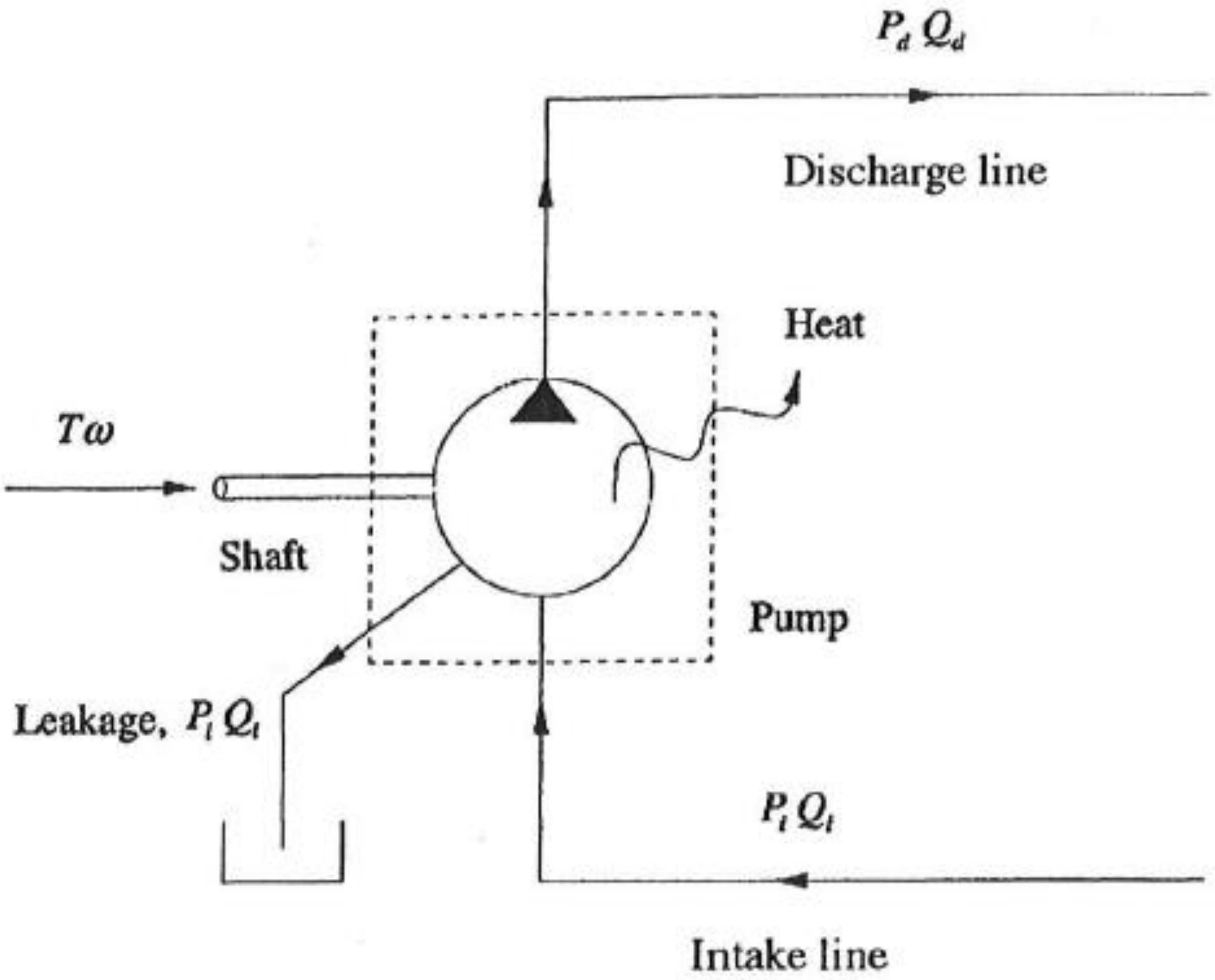
- The only way to eliminate valve throttling losses is to eliminate valves. This can be done if each actuator is supplied by its own variable-displacement pump. The flow into the actuator can be controlled by simply controlling the flow generated by the pump.
- Theoretically, only the flow required by the system is delivered by the pump and the pump outlet pressure will only be as high as required by the load on the actuator. Any valves included in the system are on-off valves which only serve to change the direction of the flow into the actuator. Since the valves are either fully open or fully closed, valve throttling loss is essentially eliminated.

- The losses of the system are mainly due to internal pump leakage and mechanical friction.
- The drawbacks of this type of system include:
  - Increased equipment cost since more than one pump is required.
  - Slower dynamic response since the bandwidth of variable-displacement pumps is lower than for control valves. The system cannot react as quickly to changes in demanded flow or load. This will manifest itself as increased tracking error.
- Let’s review pump-controlled systems.

Pump Efficiency

- The task of the hydraulic pump is to convert rotating mechanical shaft power into fluid power that may be used downstream of the pump.
- None of the hydraulic pumps is 100% efficient. They all lose power in the process of converting power.
  - Fluid leaks away from the main path of power transmission.
  - Friction exists within the machine.
- The diagram on the next slide shows the power that flows in and out of a typical hydraulic pump, regardless of type.

Power Flowing In and Out of The Pump



- Power is supplied to the pump through the rotating shaft by an external drive device (not shown).
- As the shaft rotates, the pump draws fluid into the inlet side and pushes fluid out of the discharge side.
- The input power to the shaft is torque times the shaft angular velocity, i.e., $T\omega$.
- Power is also delivered to the pump on the inlet side by any pressure that may exist at the intake port of the pump. This hydraulic power is equal to the pressure times the volumetric flow rate, i.e., P_iQ_i.
- The discharge power of the pump is equal to the discharge pressure times the discharge volumetric flow rate, i.e., P_dQ_d.

  - Power also leaks away from the pump in the form of internal leakage. This power loss is calculated as $P_\ell Q_\ell$, where P_ℓ is the pressure drop across the leak path, and Q_ℓ is the leakage volumetric flow rate.
  - Finally, power also leaves the pump in the form of dissipating heat.

- The <u>overall pump efficiency</u> is defined as the useful output power divided by the supplied input power:

$$\eta = \frac{P_d Q_d}{T\omega}$$

- We can use the volumetric displacement of the pump V_d to separate the overall efficiency into two components: the volumetric efficiency η_v and the torque efficiency η_t.

$$\eta = \eta_v \eta_t$$

- The volumetric efficiency is given by:

$$\eta_v = \frac{Q_d}{V_d \omega}$$

  - It is used for describing power loses that result from internal leakage and fluid compressibility.

- The torque efficiency is given by:

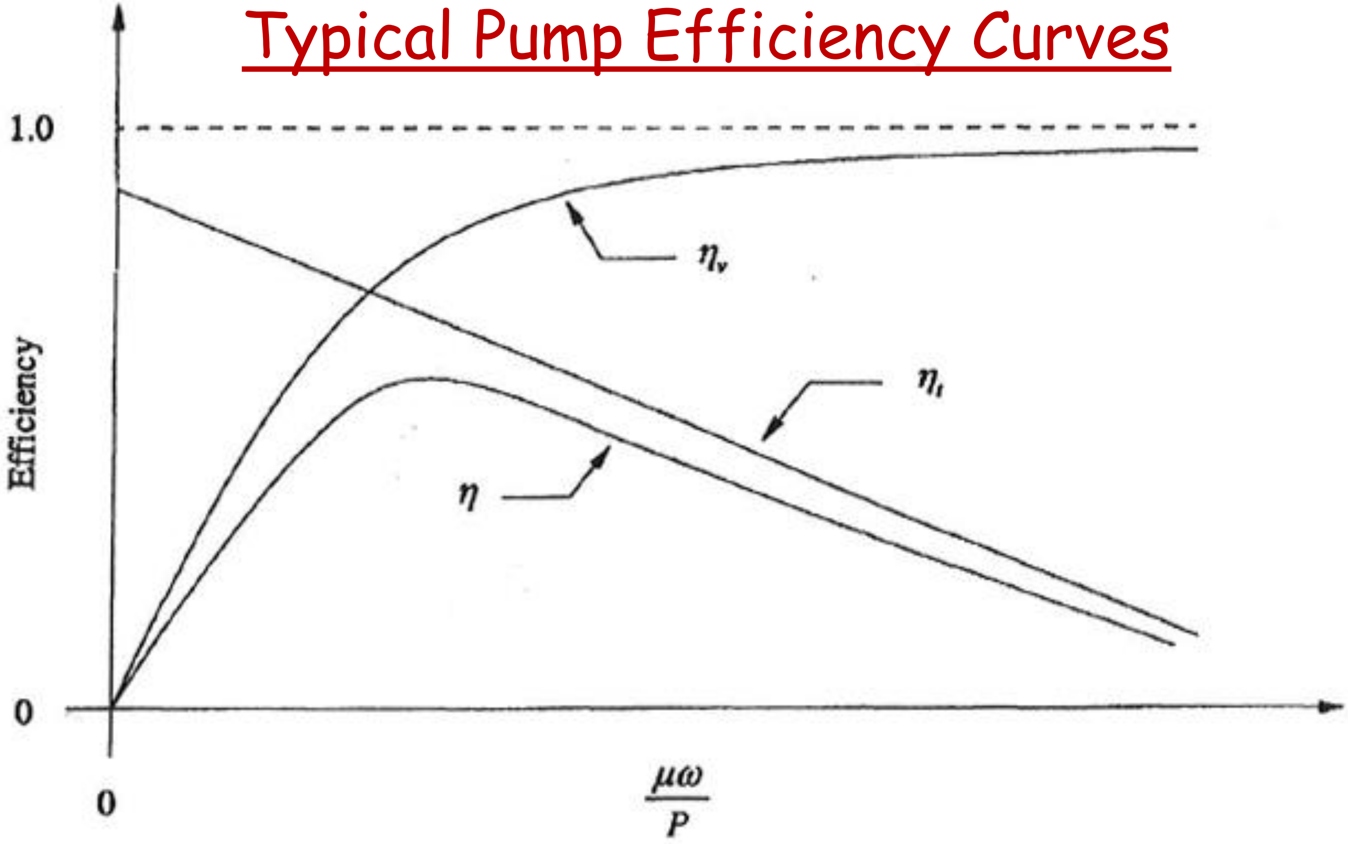
$$\eta_t = \frac{V_d P_d}{T}$$

  - It is used for describing power loses that result from fluid shear and internal friction.

- The volumetric displacement V_d is given in units of volume per radian. The volumetric displacement per revolution is given by $2\pi V_d$.

- The diagram shows a typical graph of the pump efficiency plotted against the nondimensional group μω/P, where μ is the fluid viscosity, ω is the angular shaft speed, and P is the pressure drop across the pump.

Typical Pump Efficiency Curves



- There does not seem to be an accurate way to predict pump efficiency characteristics in an a priori way. Experimental coefficients are required in the modeling process.
- The pump efficiency terms may be grouped into physical expressions and modeled as follows.

$$\eta_v = 1 - C_\ell \frac{P}{\mu\omega} - C_t \sqrt{\frac{P}{\mu\omega}}$$

$$\eta_t = 1 - C_s - C_c \frac{\mu\omega}{P} - C_h \sqrt{\frac{\mu\omega}{P}}$$

- Definitions
  - C_ℓ accounts for compressibility effects and low-Reynolds-number leakage
  - C_t accounts for high-Reynolds-number leakage
  - C_s accounts for starting torque losses
  - C_c accounts for Coulomb friction torque losses that are proportional to applied loads within the pump
  - C_h accounts for hydrodynamic torque losses that result from fluid shear
- All these coefficients must be determined from experiments. The determination of these coefficients is best achieved by using a least-squares evaluation of data points that have been used for determining actual efficiencies corresponding to specific values for $\mu\omega/P$.

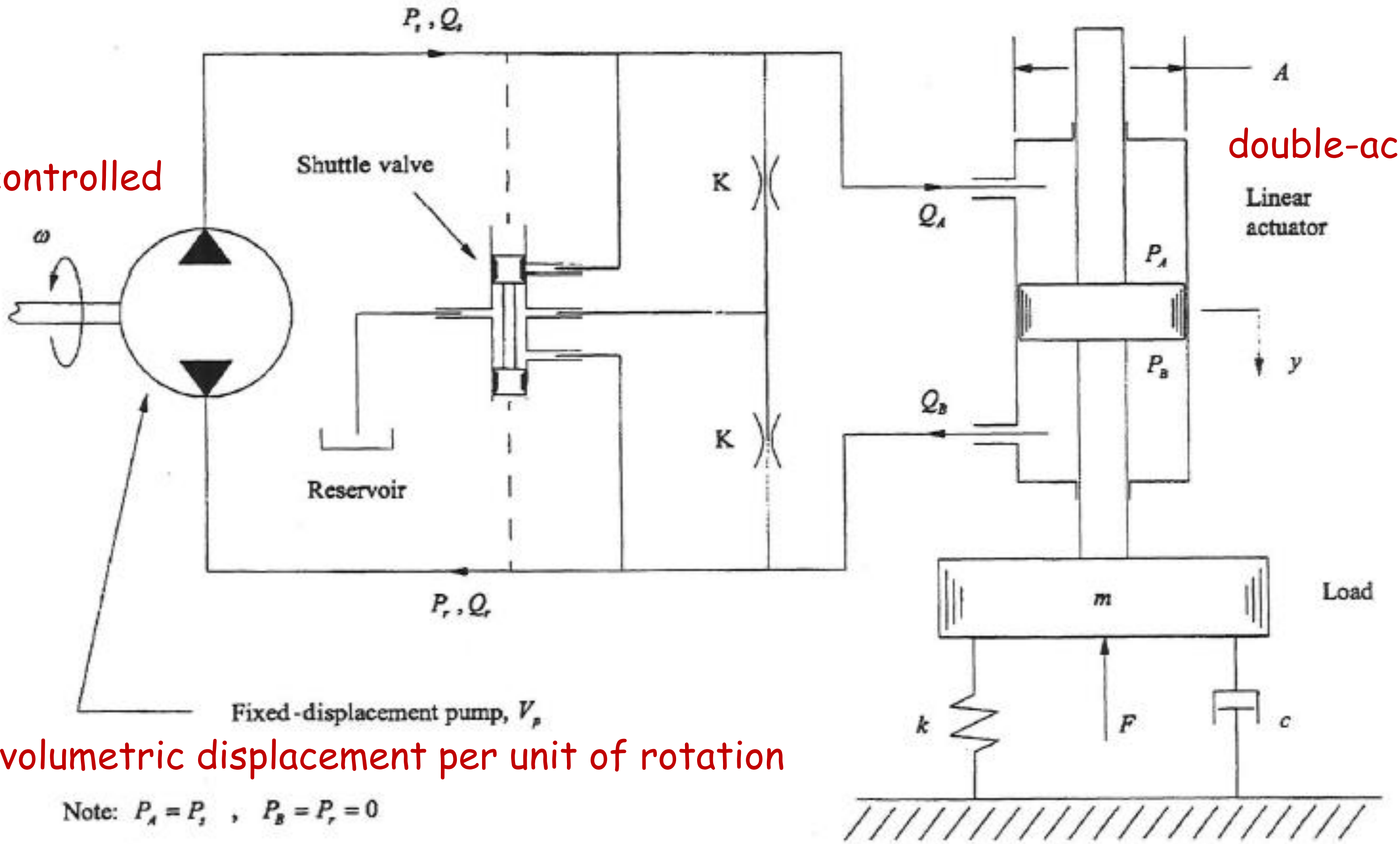
Pump-Controlled Hydraulic Systems

- A pump-controlled hydraulic system uses a pump as opposed to a control valve for directing hydraulic power to and from an actuator that is used to generate useful output.
- Pump-controlled hydraulic systems exhibit an efficiency advantage over valve-controlled systems due to the fact that the control valve introduces a pressure drop that results in significant heat dissipation.
- The pump-controlled system does not use this valve; the immediate power needs of the output are met directly by the power source and that increases the overall operating efficiency of the system.

- However, there are disadvantages of a pump-controlled system:
  - The response characteristics of pump-controlled systems can be slower due to the longer transmission lines that usually are used for reaching the output actuator and the accompanying fluid compressibility effects.
  - To eliminate long transmission lines, often times the pump, actuator, and power source are too bulky to be collocated.
  - Pump-controlled systems consist of a single pump that operates a single actuator. Multiple actuators cannot share the power that is generated from one pump, so the pump cost must be included with the overall cost of a single actuator.

- In pump-control of a linear actuator, since the pump operates symmetrically as it sends flow to and receives flow from the output actuator, only double-rod linear actuators are suitable.
- Typical applications for these systems include industrial robots and flight-surface controls in the aerospace industry.
- Pump-controlled rotary actuators, used to drive a rotating shaft, are often called hydrostatic transmissions. They are frequently used for lawn tractors, off-highway earth-moving equipment, and as a constant-speed drive for various aerospace flight applications.

Fixed-Displacement Pump Control of a Linear Actuator



speed controlled

double-acting

V_p = volumetric displacement per unit of rotation

Note: $P_A = P_s$, $P_B = P_r = 0$

F = load disturbance force

- **Comments:**
  - Speed-controlled (driven by an input shaft rotating at a variable angular velocity ω) fixed-displacement pump with volumetric displacement per unit of rotation V_d
  - Double-rod linear actuator to facilitate symmetric action of the actuator
  - Pressurized areas are the same on both sides of the actuator
  - Load is a single mass-spring-damper system with a load-disturbance force
  - Rod connects the load to the actuator piston
  - Volumetric flow of hydraulic fluid into the actuator is controlled by the output flow of the pump
  - For positive ω, pump flow is to side A of the actuator; the load moves down and flow exits the actuator from side B.

- For negative ω, pump flow is to side B of the actuator; the load moves up and flow exits the actuator from side A
- Q_A and Q_B are the volumetric flow rates into and out of the actuator
- Shuttle Valve
  - Connects the low-pressure side of the hydraulic control system to the reservoir
  - It keeps the low-pressure side of the circuit at a constant reservoir pressure, i.e., zero gage pressure
  - It keeps the fixed-displacement pump from drawing a vacuum and causing fluid cavitation
  - It allows for the return flow to be cooled by a low-pressure radiator (not shown)

  - The shuttle valve shifts up or down depending on which side of the circuit is at high pressure; the dashed lines indicate pressure signals that are used to move the shuttle valve

- A leak path on both sides of the hydraulic circuit is shown and is characterized by the leakage coefficient K; this low-Reynolds-number flow occurs naturally due to the inherent internal leakage of the system.

- Analysis
  - Load Analysis

$$m\ddot{y} + c\dot{y} + ky = \eta_{af} A(P_A - P_B) - F$$

    - η_{af} is the force efficiency of the actuator

$$\eta_{af} = \frac{F}{P_A A_A}$$

  - Pressure Analysis
    - Assume that the pressure transients that result from fluid compressibility in the transmission lines are negligible. This assumption is especially valid for a system design in which the transmission lines between the valve and actuator are very short, i.e., small volumes of fluid exist on either side of the actuator.

- The omission of pressure transient effects is also valid for systems in which the load dynamics are much slower (seconds) than the pressure dynamics (milliseconds) themselves.
- If there are long transmission lines between the valve and actuator, or if the bulk modulus is reduced because of entrained air in the fluid, or if the actuator dynamics are very fast, a transient analysis of the pressure conditions on both sides of the actuator may be necessary.
- Here, we assume that pressure transients may be safely neglected.
- Therefore

$$Q_A = \frac{A\dot{y}}{\eta_{av}}$$

η_{av} = actuator volumetric efficiency

- From the diagram we see that for an incompressible fluid:

$$Q_A = Q_s - KP_A$$

- The supply flow is given by:

$$Q_s = \eta_{pv} V_p \omega$$

η_{pv} = pump volumetric efficiency

- Combining equations results in:

$$P_A = \frac{\eta_{pv} V_p}{K} \omega - \frac{A}{\eta_{av} K} \dot{y}$$

- The shuttle valve is used to connect the low-pressure side of the hydraulic circuit to the reservoir. The pressure on side B of the actuator is therefore 0 gage pressure.

$$P_A = \frac{\eta_{pv} V_p}{K}\omega - \frac{A}{\eta_{av} K}\dot{y}$$

- This equation shows a pump velocity and an actuator velocity dependence for the fluid pressure in side A. An adjustment of the pump velocity term will provide a control input to the dynamic load equation. The linear velocity term will be useful in providing favorable damping characteristics.

- ***Analysis Summary***

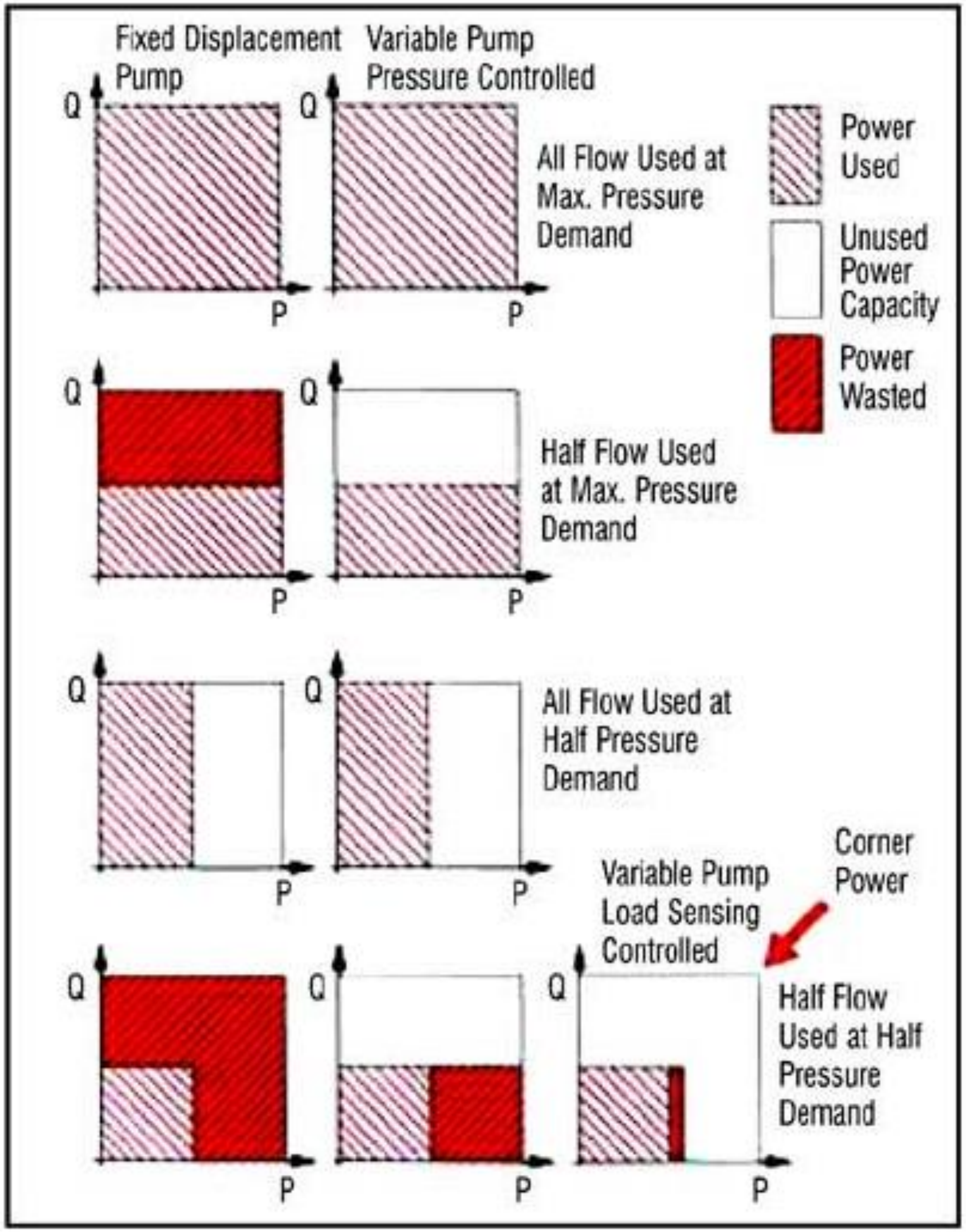
$$m\ddot{y} + \left(c + \frac{\eta_{af}}{\eta_{av}} \frac{A^2}{K} \right)\dot{y} + ky = \frac{\eta_{af}\,\eta_{pv} V_p A}{K}\omega - F$$

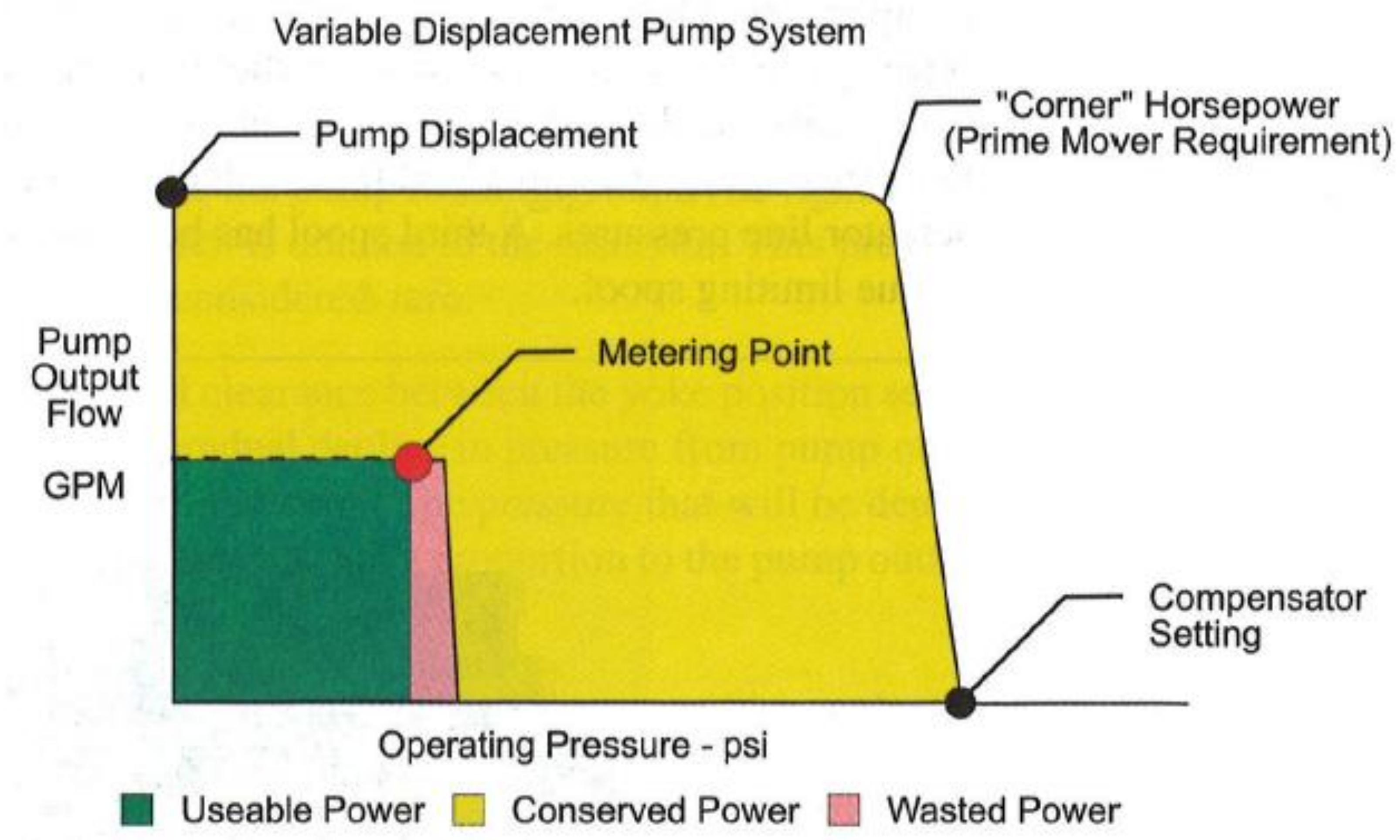
- We see that the mechanical design of the linear actuator and the volumetric displacement of the pump have a decisive impact on the overall dynamics of the hydraulic control system. The design parameters help to shape the effective damping of the system and provide an adequate gain relationship between the input velocity of the pump and the output motion of the load.

Load-Sensing Pumps

- Load-sensing pumps maintain a slightly higher supply pressure than the maximum cylinder chamber pressure.
- Thus, when the highest chamber pressure decreases, so does the supply pressure and less energy is input to the system as compared to a constant pressure supply.
- Because the pressure drop between the source and the chambers is reduced, the throttling losses are also reduced and the same performance can be achieved with lower input energy.

- When a load-sensing pump powers a single actuator, the operation is nearly as efficient as would be achieved in a pump-controlled system.
- When multiple actuators with different pressure requirements are supplied, some energy is wasted in throttling the source pressure down for all actuators except the one with the highest pressure requirement.

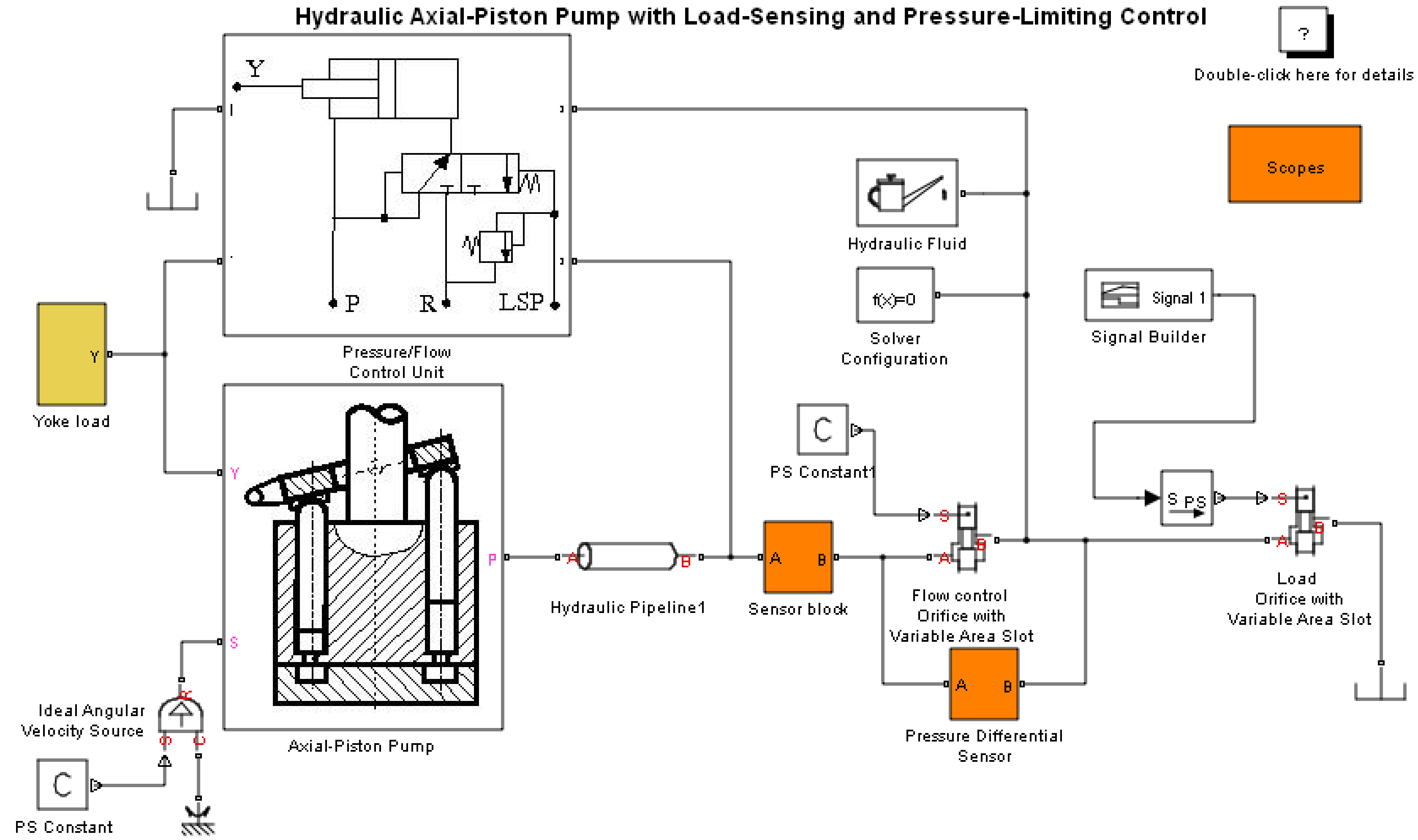

Fixed Displacement Pump
Variable Pump Pressure Controlled
Q
P
All Flow Used at Max. Pressure Demand
Power Used
Unused Power Capacity
Power Wasted
Half Flow Used at Max. Pressure Demand
All Flow Used at Half Pressure Demand
Variable Pump Load Sensing Controlled
Corner Power
Half Flow Used at Half Pressure Demand



Energy Savings Caused by Load Sensing

Hydraulic Axial-Piston Pump with Load-Sensing and Pressure-Limiting Control

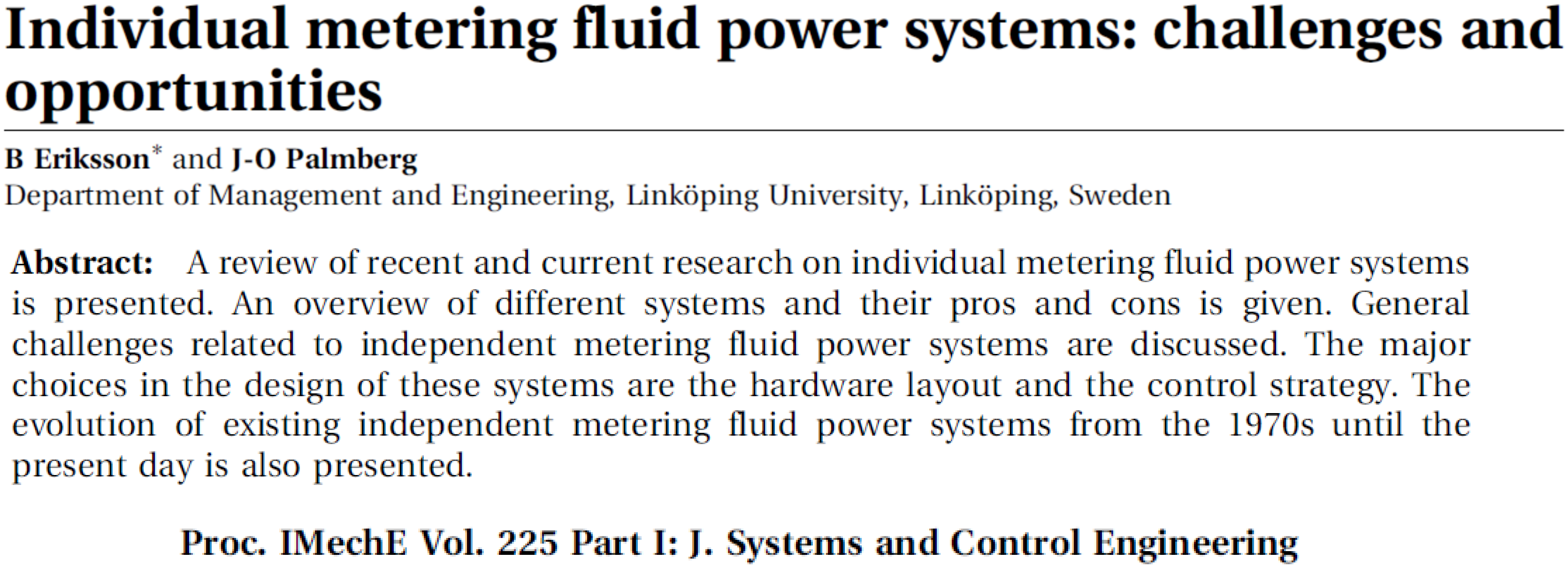
The demo models a test rig designed to investigate interaction between an axial-piston pump and a typical control unit, simultaneously performing the load-sensing and pressure-limiting functions. To assure required accuracy, the model of the pump must account for such features as interaction between pistons, swash plate, and porting plate, which makes it necessary to build a detailed pump model.



Independent Metering Valves

- In many traditional hydraulic applications, a single valve (e.g., a 3-position, 4-way proportional directional valve) controls the flow rates into both cylinder chambers.
- For such a system, it is not possible to independently specify the flow into each cylinder chamber.
- The two flow rates are coupled by the single spool position, making the independent control of both chamber pressures impossible.
- While the net force on the piston (load force) can be controlled, no additional freedom is available to influence the pressures.

- Independent metering valve configurations have been studied to see how separate valves may be used to increase efficiency and performance.
- A detailed review of the state of the art for these types of systems is given in:



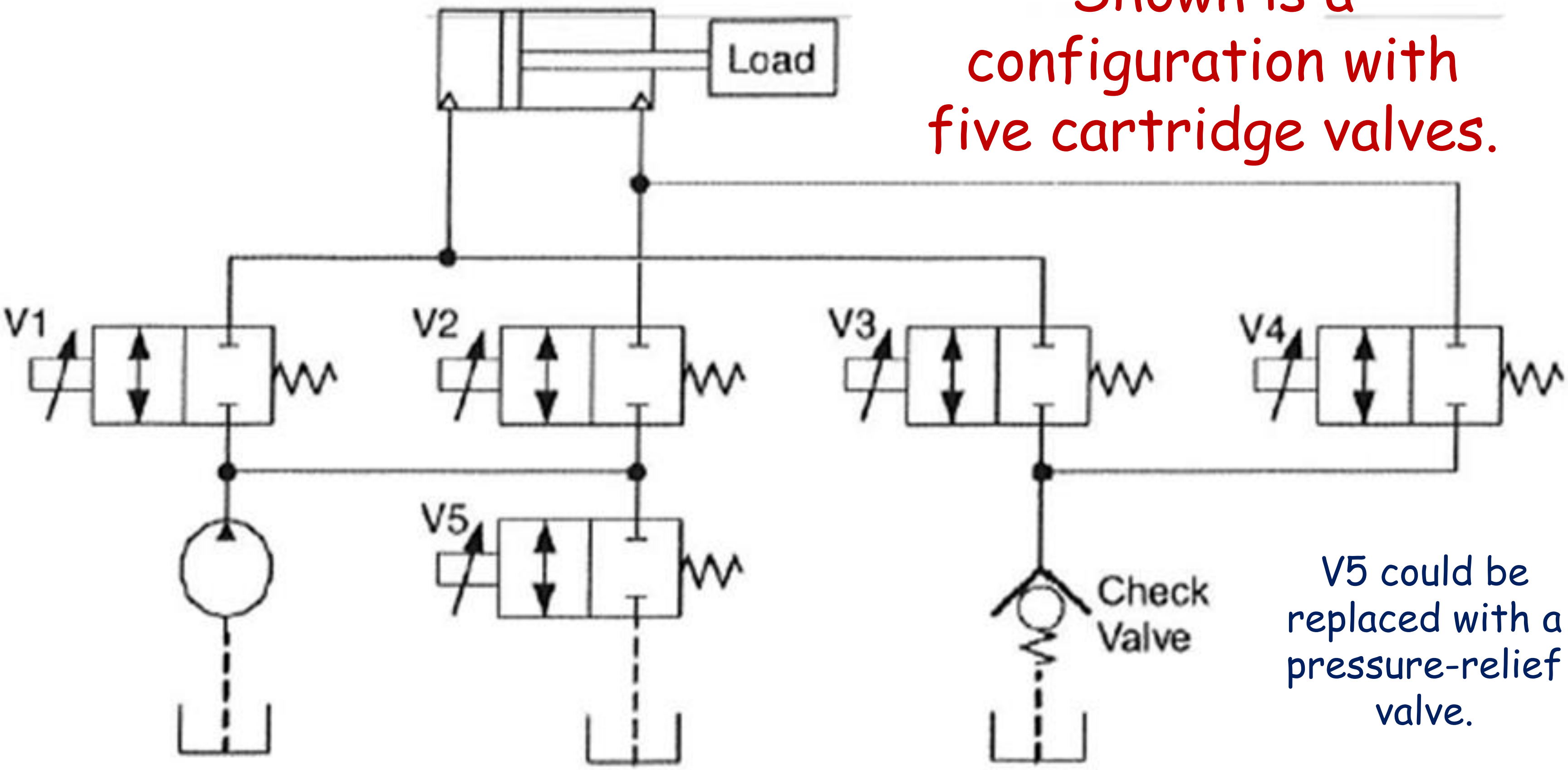
Individual metering fluid power systems: challenges and opportunities

B Eriksson* and **J-O Palmberg**
Department of Management and Engineering, Linköping University, Linköping, Sweden

Abstract: A review of recent and current research on individual metering fluid power systems is presented. An overview of different systems and their pros and cons is given. General challenges related to independent metering fluid power systems are discussed. The major choices in the design of these systems are the hardware layout and the control strategy. The evolution of existing independent metering fluid power systems from the 1970s until the present day is also presented.

Proc. IMechE Vol. 225 Part I: J. Systems and Control Engineering

Shown is a configuration with five cartridge valves.



V5 could be replaced with a pressure-relief valve.

The flow from the pump to each chamber and the flow from each chamber to the tank are controlled by four separate valves. A fifth valve connects the pump directly to the tank. This configuration can emulate many spool valve geometries simply by changing the control software.

- Much study of independent metering valves focuses on the ability of the configuration to operate as a “variable geometry” spool valve, i.e., it is able to recreate the flow characteristics associated with different geometries. Changes made in software allow the valve’s characteristics to be significantly altered, giving rise to names such as smart valve, multi-function valve, and programmable valve.
- From an efficiency standpoint, the more important property of independent metering valves is the ability to independently control the flows into and out of both cylinder chambers. The practical significance of this is independent control of the cylinder pressures.

- When such a configuration is used with a load-sensing pump, there is potential for energy savings.
  - Since the pressures can be independently specified, a given net force on the cylinder can be achieved in a variety of ways.
  - By setting one chamber pressure to a low level, the other chamber pressure can be used to achieve the required load force with as low a pressure as possible.
  - The result is a lower maximum chamber pressure.
  - For a load-sensing pump, the supply pressure (and thus the input energy) will be significantly reduced.

Regeneration Flow

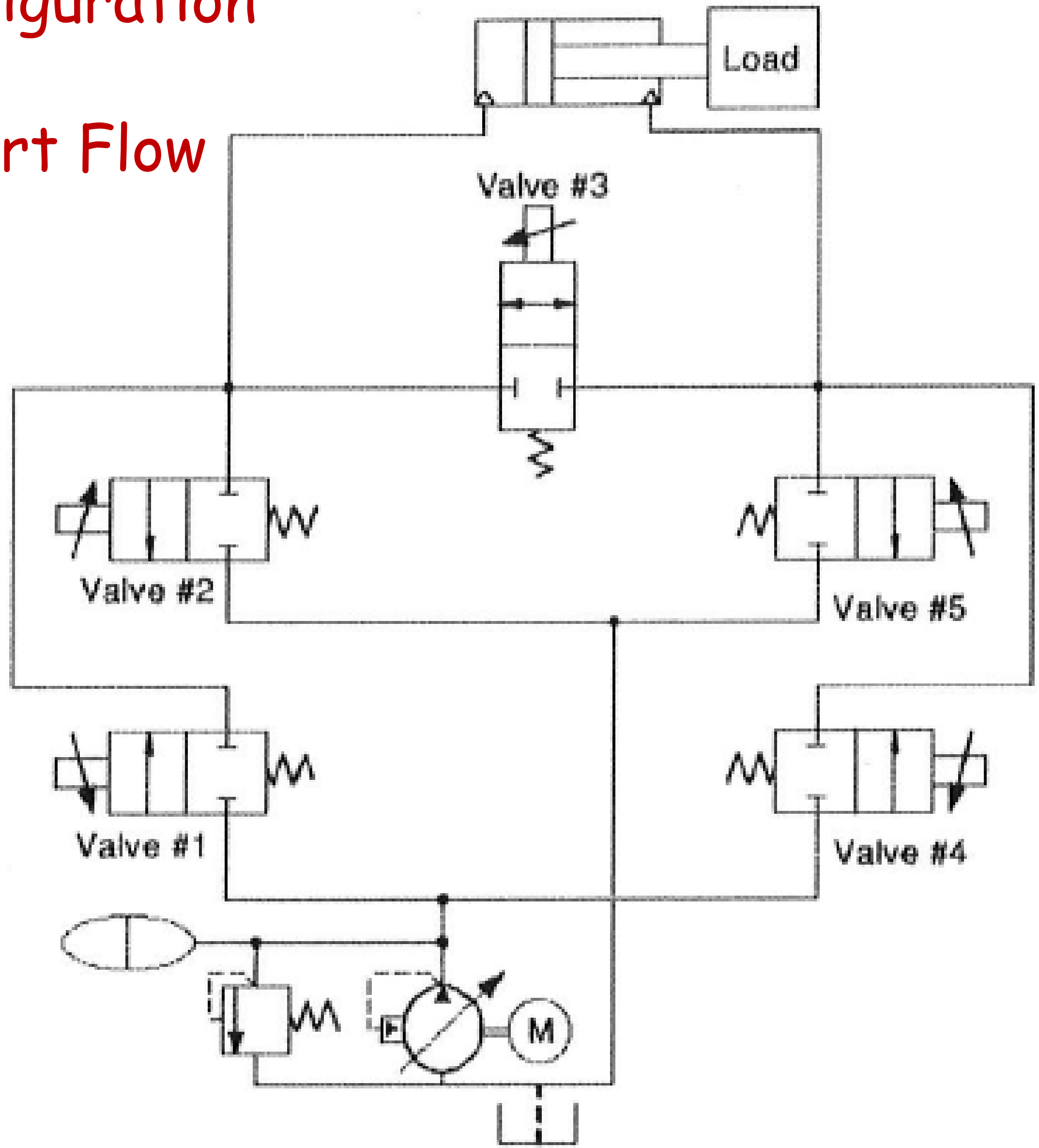
- If a constant pressure source is used, then no matter how the system pressures change, the energy supplied for two different situations will be equal if the flows are equal.
- The only way to save energy is to reduce the flow from the pump. This may be accomplished by recycling hydraulic fluid already delivered to the system.
- Regeneration is the process of directing flow from one chamber into the other without the use of the pump.

- Regeneration flow can be used whenever the cylinder chamber supplied by the pump actually has a lower pressure then the other chamber. This commonly occurs during deceleration periods or when large force is acting in the direction of motion, e.g., lowering a heavy load. This type of load is called an overrunning load.
- In the figure on the slide 34, flow could be regenerated from one chamber to the other through the return valves V3 and V4. This is referred to as low-side regeneration. This system has a check valve in the return line which supports a slight pressure drop before opening, thus maintaining positive pressure. The flow could be in either direction, depending on the direction of the overrunning load. Energy is saved because the required supply flow is reduced.

- A single-rod cylinder has a significant difference in the piston areas. When only a small positive load force is required (e.g., when extending the rod with nearly constant velocity), the pressure of the rod-end side can exceed the pressure of the head end.
- Thus, fluid can be regenerated from the rod chamber to the head chamber. This type of regeneration has been used with a four-valve configuration by allowing the flow to pass through the supply valves (fluid flows first through valve V2 and then valve V1 in the figure on slide 34.) In order for this to be possible, the rod chamber pressure must exceed the source pressure.

- This type of flow is called high-side regeneration. High-side regeneration can boost the maximum extension speed, but usually requires significantly higher source pressure (and hence greater energy consumption for a load-sensing pump) than normal operation.
- A 5-valve configuration which uses a valve to control flow directly between the cylinder ports has been shown to save energy. This is shown on the next slide. Valve #3 is the regeneration valve.
- The extra valve here connects the two cylinder ports directly and does not connect the pressure source and tank (although a relief valve is included which does provide this function.)

Five-Valve Configuration with True Cross Port Flow



- When true cross port flow is possible, the restrictions on the pressures during regeneration are less restrictive.
  - Thus, when an overrunning load is present, the pressure in the regeneration line can exceed the tank pressure significantly.
  - Similarly for constant velocity extension, the rod chamber pressure need not exceed the supply pressure to use regeneration. The only restriction is that the cylinder chamber supplied by the pump actually has a lower pressure than the other chamber.
  - The flow from the high pressure chamber can be used to either reduce the pump flow (thus saving energy) or increase the maximum possible flow (thus increasing the speed).

Energy Recovery Accumulator

- In fluid power systems, an accumulator often acts as an energy storage device.
- A hydro-pneumatic accumulator contains a quantity of pre-charged gas (often nitrogen is used) and a port through which oil can enter or leave the accumulator.
- The gas is separated from the oil by a barrier such as a piston, bladder, or diaphragm.
- When the port is exposed to high pressure, oil enters the accumulator, compressing the gas and storing energy.
- When the port pressure falls, oil will flow from the accumulator under the force of the pressurized gas.

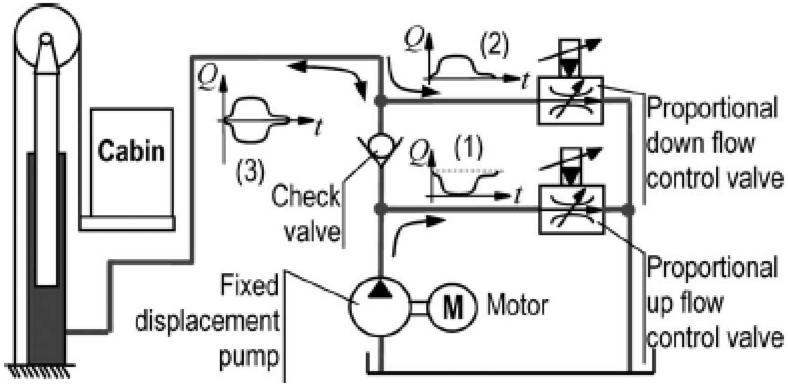
- Accumulators are able to integrate directly into hydraulic systems and have been used in many applications.
- An energy-recovery system for a hydraulic elevator has been proposed which incorporates a hydraulic accumulator and a four-quadrant motor. As the elevator traveled both upward and downward, oil was directed through the hydraulic motor. This enabled both capture and reutilization of the potential energy. A variable-frequency drive electric motor connected to the hydraulic motor allowed the hydraulic motor to supplement the energy provided to the load or to recover excess energy. (See next slide).

New Investigation in Energy Regeneration of Hydraulic Elevators

Huayong Yang, Wei Sun, and Bing Xu

IEEE/ASME TRANSACTIONS ON MECHATRONICS, VOL. 12, NO. 5, OCTOBER 2007

***Abstract*—In the conventional valve-controlled hydraulic elevator, when the cabin moves downwards, the entire potential energy of the cabin is wasted and converted into fluid heat by throttling. Thus, the energy consumption of a traditional hydraulic elevator is much higher than that of the traction elevator. To reduce the energy consumption and power installation requirements, the energy-regenerative hydraulic elevators have been developed since 1997. In this paper, different generations of the design are discussed for energy-regenerative system; in particular, a new generation is focused on in detail. Experimental studies of this new design are carried out to compare the energy consumption of different system designs of hydraulic elevator. Finally, a conclusion is drawn that the new generation of hydraulic elevator can achieve a significant energy-saving performance compared to the traditional elevators.**



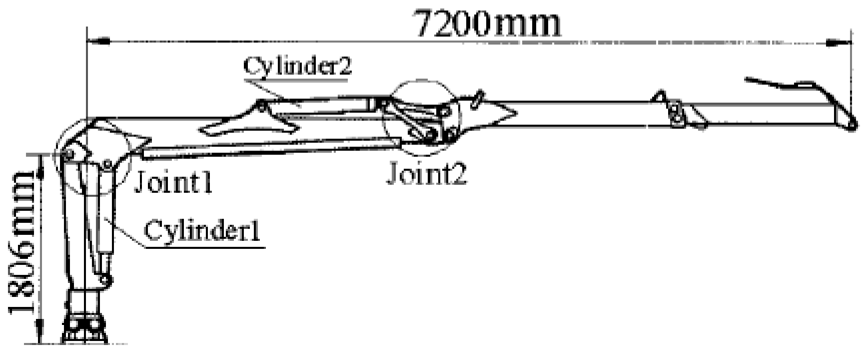
- An energy recovery system for a hydraulic crane has been developed (see slide 48). This system incorporates a pair of assistant cylinders mounted in a parallel force configuration with the main boom cylinder, thus sharing the load among all three cylinders. When the load is first raised, the assistant cylinders draw oil from the oil reservoir through a check valve. During this motion, all the force is provided by the main cylinder. When the cylinders retract while lowering the load, the force of the load forces oil from the assistant cylinders into an accumulator. The next time a load is lifted (and for all subsequent cycles) the pressure stored in the accumulator acts on the assistant cylinders.

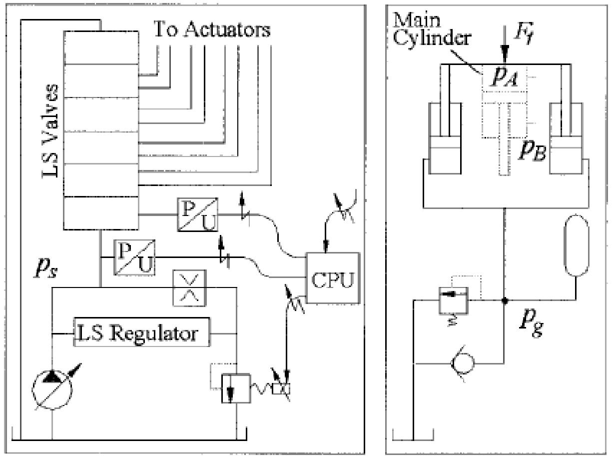
- This significantly reduces the load required from the main cylinder, allowing the load-sensing pump to supply flow at a lower pressure to the main cylinder. This results in significant energy savings.
- This energy recovery system is essentially decoupled from the original system since there is no flow directly between the main cylinder and the assistant cylinders. The disadvantage of this type of system is the requirement for one or two additional hydraulic cylinders.

An energy recovery system for a hydraulic crane

X Liang* and **T Virvalo**
Institute of Hydraulics and Automation, Tampere University of Technology, Tampere, Finland

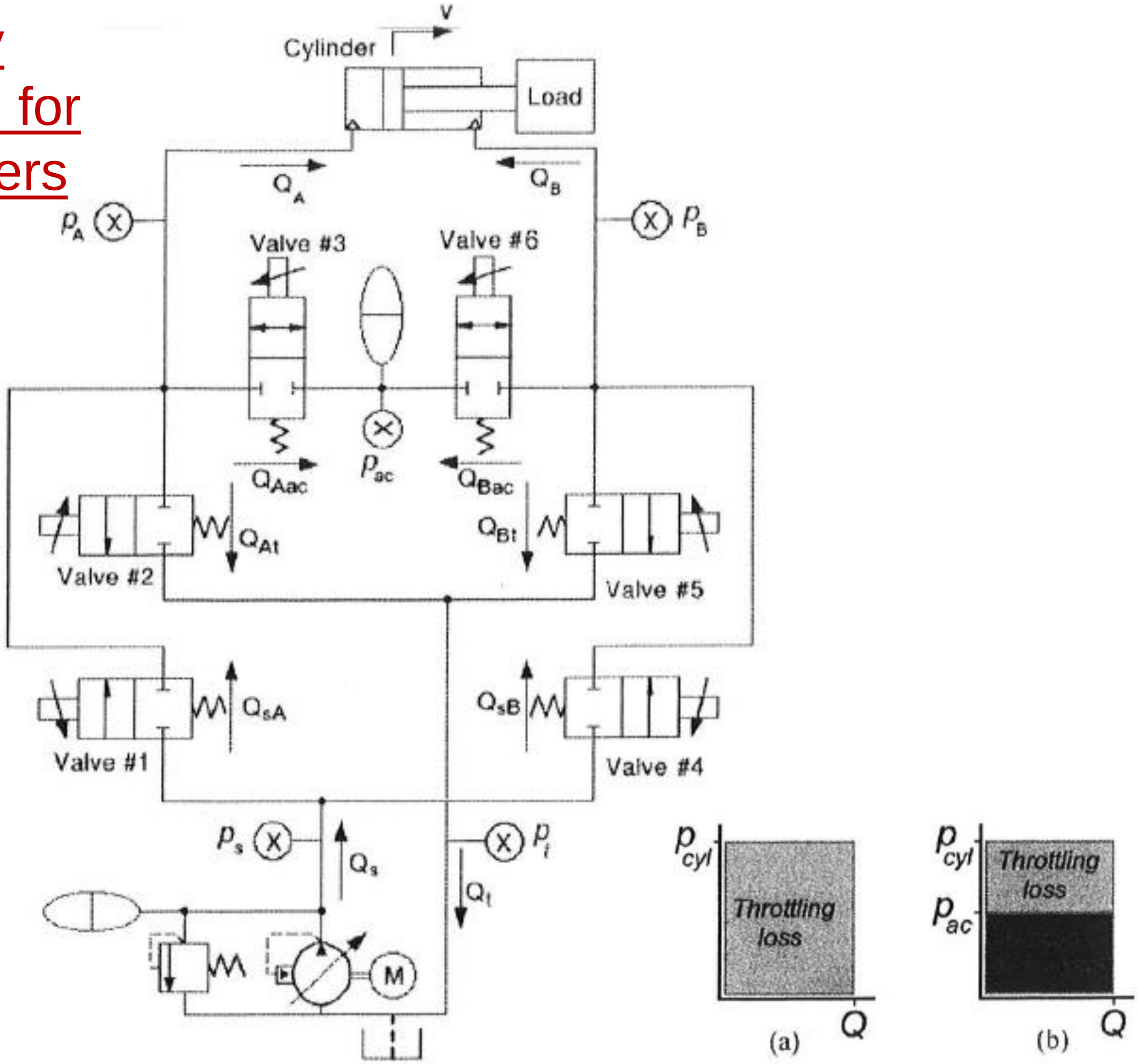
Abstract: In this paper an energy recovery system for a hydraulic crane is presented. An assistant system with an accumulator is used to drive one joint of an example crane together with an electro-hydraulic load-sensing (ELS) system. The practical system is tested. The hydrostatic analysis of energy transfer is based on the experimental process and an assumed typical duty cycle. The experimental and theoretical results show that the application of the assistant system with the ELS crane system can recover and reutilize energy, save pump supply energy and improve energy utilization of the crane system. The success of its design and test in an example hydraulic crane will encourage its extension to different commercial applications in other lifting machinery.





- A novel energy recovery system for hydraulic cylinders is shown on the next slide. The system consists of an accumulator and two control valves, through which the flows to and from the cylinder chambers are controlled. A regeneration flow path is created when both valves are opened.
- The effect of this modification is to decouple the regeneration flows, meaning the flow out of the high pressure cylinder need not equal the flow into the low pressure cylinder. This is the case for a simple regeneration path composed only of piping and valves, but when an accumulator is used, excess flow from one cylinder can be stored in the accumulator or extra flow may be supplied from the accumulator to the cylinder.

Novel energy recovery system for hydraulic cylinders



- The accumulator may be charged whenever there is flow out of a cylinder with pressure higher than that of the accumulator, and flow from the accumulator can replace pump flow whenever there is flow into a cylinder chamber at a lower pressure than the accumulator.
- The accumulator reduces throttling losses by acting as a second flow source or sink. High-pressure flow out of a cylinder can be directed to the accumulator rather than simply throttled to the tank. The throttling loss is lower because of the lower pressure drop.
- The figure illustrates the situation where a flow Q from a cylinder at pressure p_{cyl} is throttled to the reference pressure and the case when it flows into an accumulator with pressure p_{ac}.

- The shaded areas in the diagram represent the amounts of energy lost due to throttling and recovered by the accumulator. It is noted that although energy is stored in the accumulator, it could only be reused immediately if another cylinder pressure is below the pressure p_{ac}. Then, the accumulator could act as a secondary pressure source to supply flow to the cylinder at low pressure, replacing the flow from the pump.
- Of course, for a very slight pressure drop, the maximum achievable flow through a valve may be less than the required amount. In such cases, the accumulator can act as a source or sink in parallel with the main pressure source or the tank, which still provides a portion of the benefits seen when the accumulator is used to provide the entire flow.