AEM 617

Hydraulics

Part 2

A 320 vs B 737 Cockpit

tiny.cc/AEM617A320B737

A 320 flight computer

tiny.cc/AEM617A320FBW

Gulfstream G500 and G600 FBW

tiny.cc/AEM617G500

Dr. Oneill,

I might be able to shed some light on some airbus points, my dad has a type rating in the A320 so I'm pretty familiar with it.

The side stick doesn't have any force feedback or even a linear stick force to G relationship. Its kind of just a friction gradient. going from 0 to 1/4 aft stick is the same force as going from 3/4 to full aft stick. But it is a full displacement stick as opposed to the F-16. I've flown the sim for the Viper and its got 5 degrees or so of play in each axis, but very different from the airbus in the the amount of force. its a a hefty pull to get max g out of the jet where the airbus is just a light pull against the stop to have the jet give you either maximum aerodynamic g or structural g.

Also on that Air France crash, The A320 has a couple different flight control laws it can follow. There is normal, alternate, and direct law flight mode. Normally when flying the airplane it's in normal law, which has g, roll, pitch, and stall protection. When the airplane is getting close to stall it goes into ALPHAFLOOR mode which limits aft stick to prevent stall and advances power to maximum automatically. The pilots relied on that alphafloor system as they slowed the jet down for the low pass, Unfortunately when below as certain RADALT altitude the control laws revert to alternate law which does not have alphafloor protection and the jet can be stalled which put them in the left of the power curve situation they found themselves in.

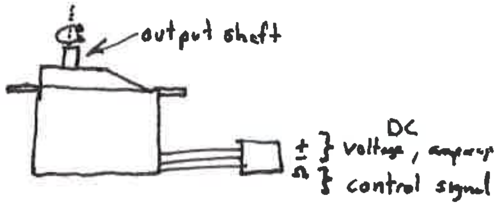
https://www.youtube.com/watch?v=IKBABNL-DDM

^04:15 - A good video of the envelope control system on the airbus

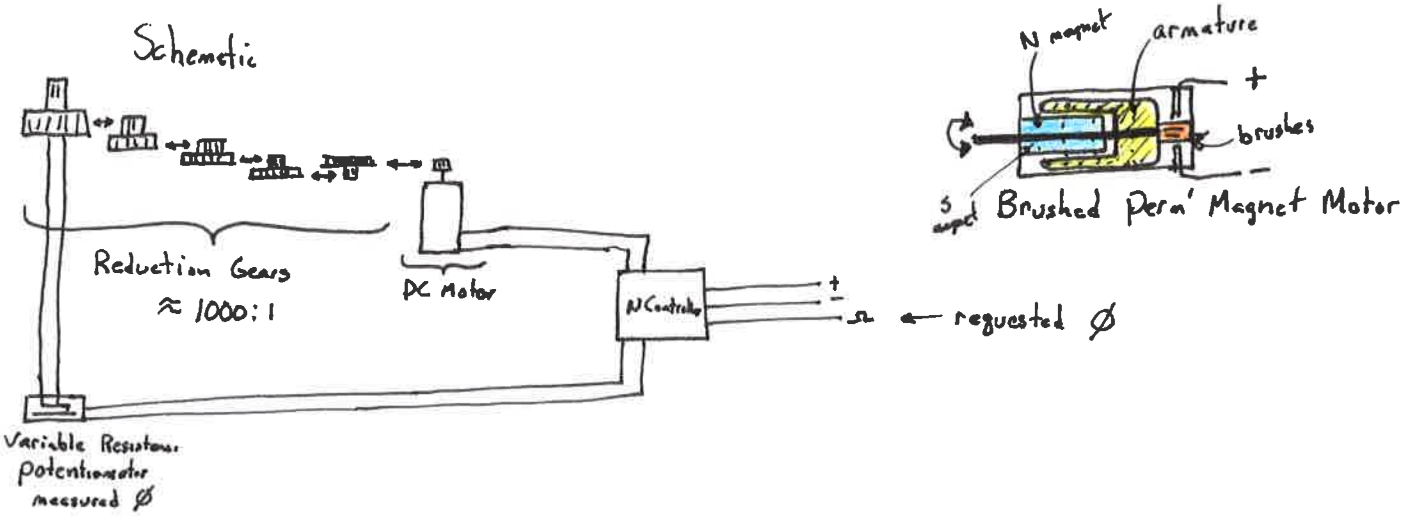
Just some insight that may be interesting to students in class.

<NAME>

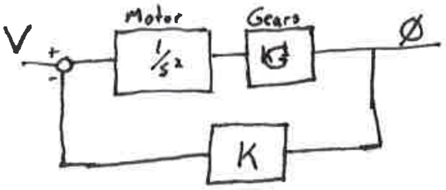
Analysis of the electromechanical servo.



Schematic



System Model



Simplifications:

- Gears have inertia
- Gears have play + friction
- Potentiometer is noisy (sometimes terrible on small servos)

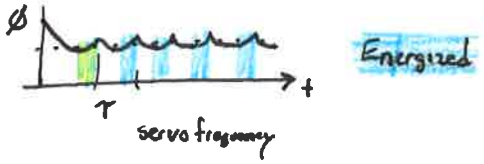
Limitations

- Holding torque: An actual application requires holding a position ∅ with an applied torque.

  The brushed motor can only provide torque when ~~the~~ current is applied.

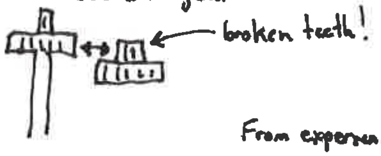
  And the motor torque is <u>more</u> than necessary to maintain position ∅. (i.e overshoot)

  This is the source of servo buzz.



- Gear strength

  The reduction drive requires high torque with a limited (very limited!) number of engaged teeth. High quality servos use metal gears or reinforced composites for the final reduction gear



A true weak link

From experience, don't buy nylon geared servos.

Hydraulics Symbols

Reservoirs:

vented

pressurized

line terminates *above* fluid

line terminates *below* fluid

Tangent topic!
Not yama or shan

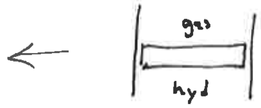
Accumulators:

unloaded

spring loaded

weighted

Gas charged



Pumps

Single direction

Bi directional

Variable displacement

Variable displacement pressure compensated

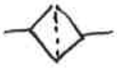
Motor

Single direction

Bi directional

Electric Motor

Filters / lines



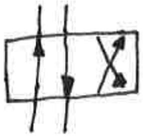
Filter

orifice
"fixed restriction"

Variable restriction

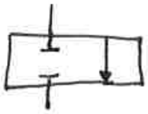
Manual
On-OFF valve

Valves



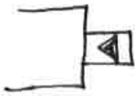
4 way, 2 position

6 way, 2 position
Solenoid operated



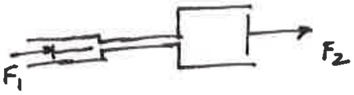
2 way, 2 position
normally closed

pressure valves

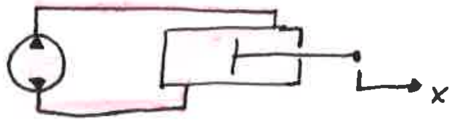


pilot pressure
internal

We previously considered the simple force transferring hydraulic system



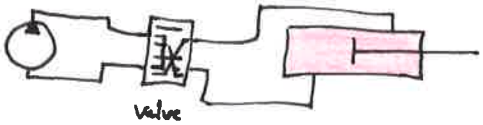
A logical development of this concept would be:

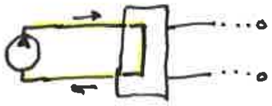


The actuator position depends on the volume of fluid pumped into each side of the cylinder.

This would require a hydraulic pump capable of both high $\dot{Q}$ and low $\dot{Q}$ at high pressures.

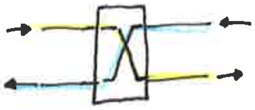
Variant seen in heavy duty equipment (e.g. farm, construction)





Centered

"up"



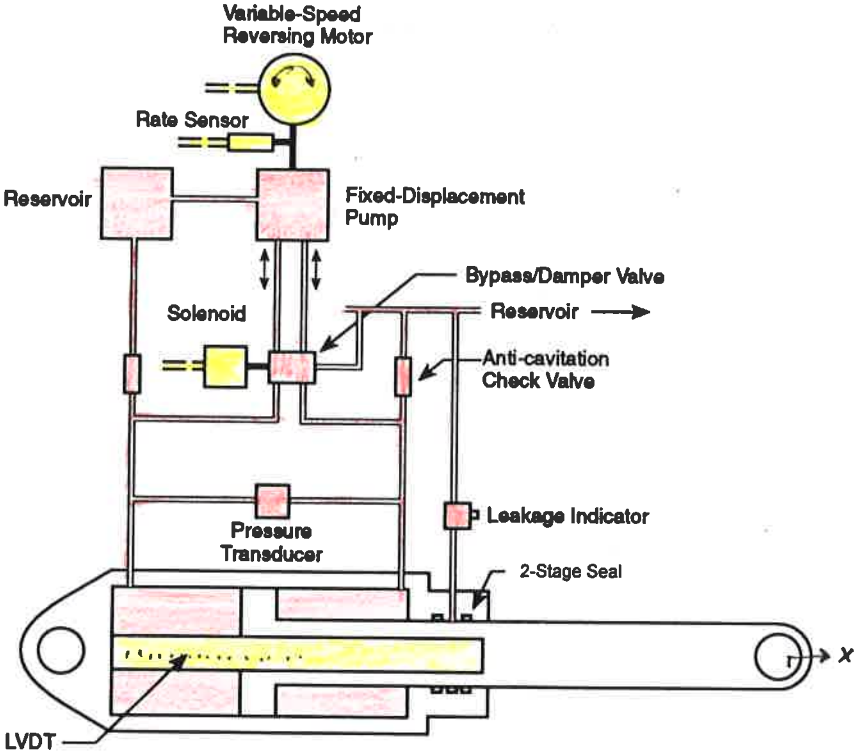
"down"

Not proportional, shock load, no leak/loss protection

Maybe ok for gear, but not for flight surfaces

Boeing tried a variant in the mid 1980s.

Electrohydrostatic actuator (EHA)



Linear Variable Differential Transformer: a position sensor

Figure 6-8. Servomotor IAP (EHA) Arrangement.

Rejected:

- Start-stop cycles on motor
- Motor controller overheating
- Response (frequency + lag)
- Reliability
- Stiffness requires pressure.

Integrated Hydraulic Actuator package (IAP)

Example

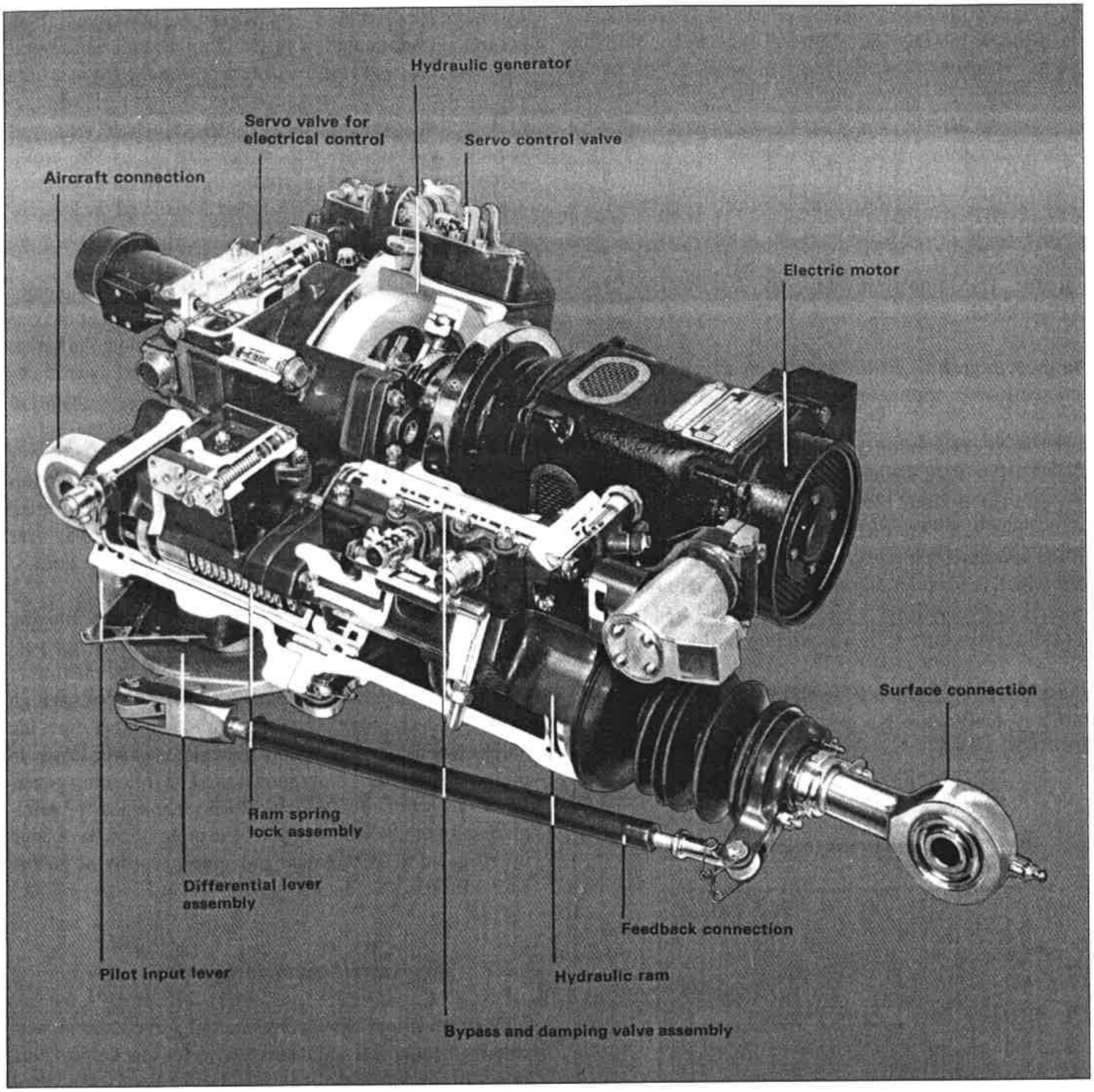


Figure 6-1. Dowty Boulton Paul Integrated Actuator Package. (Courtesy of Dowty Aerospace Wolverhampton)

Fluid Inertia (aka Inertor)

$$F = ma = m\frac{dv}{dt}$$

$$\underbrace{P}_{\text{pressure}} = \overset{\uparrow}{I}\,\frac{dQ_f}{dt}$$

Inertance

Change in flow rate wrt time

$I = \left(\frac{\rho}{A}\right)L$ for constant cross section

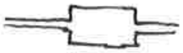
Changing the flow rate quickly creates large pressures.

Kinetic Energy

$$E = \frac{I}{2}Q_f^2$$

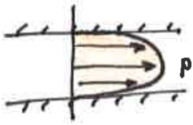
notice $\frac{I}{2}$ not $\frac{1}{2}$

Nonuniform flow velocity (variable area)



$$I = \int_0^L \frac{\rho}{A(x)}\,dx$$

Actual pipe flow has a velocity distribution (no flow at walls)



A laminar parabolic profile gives $I = 2\cdot\frac{\rho}{A}\cdot L$

Ex: Given the previous 73 hp system at 36 gpm at 3500 psi, how much Kinetic energy is contained in the ½ in ID pipe if the pipe is 30 ft long?

Specific gravity of hydraulic fluid ≈ 0.88 $\Rightarrow \rho_{hf} = 0.88\cdot\underbrace{1.94\,\frac{slug}{ft^3}}_{62.4\frac{lb}{ft^3}/32.2} = 1.71\,\frac{slug}{ft^3}$

$$\hat{I} = \frac{1.71\,slug}{} \left|\frac{4}{\pi\,0.5^2\,in^2}\right| \frac{30\,ft}{} \left|\frac{ft^2}{144\,in^2}\right. = 1.81\,\frac{slug}{in^4}$$

perfect pipe

$I = 2\hat{I}$ for parabolic profile $= 3.62\,\frac{slug}{in^4}$

$$Q_f = \frac{36\,gal}{min}\left|\frac{231\,in^3}{gal}\right|\frac{min}{60\,s} = 138.6\,\frac{in^3}{s}$$

$$E = \frac{I}{2}Q_f^2 = \frac{3.62\,slug}{in^4}\left|\frac{}{2}\right|\frac{138.6^2\,in^6}{s^2}\left|\frac{lbf\,s^2}{slug\,ft}\right|\frac{ft}{12\,in} = 2900^{\text{in-lbf}} \approx 300^{J}$$

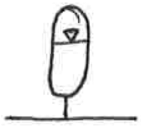
≈240 ft-lb

≈ 38 spl

≈ twice MLB pitch

≈ ½ 9mm

Accumulator



A pressurized tank useful for providing a supply of energy (pressure · volume) or as a pressure spike alleviation. (e.g. pressure transients, thermal expansion, etc)

$$\underbrace{Q_f}_{\text{flow rate}} = \underbrace{C_f}_{\substack{\text{tank}\\ \text{size}\\ \text{stiffness}}} \underbrace{\frac{dP}{dt}}_{\substack{\text{pressure}\\ \text{change}}}$$

analog of electrical capacitor $i = C\frac{de}{dt}$

Ex: Given a spring accumulator of piston area $5\ \text{in}^2$ and a spring constant of $100\ \text{lbf/in}$, what is the pressure rate of change wrt time if the flow rate is $1\ \text{in}^3/\text{s}$?

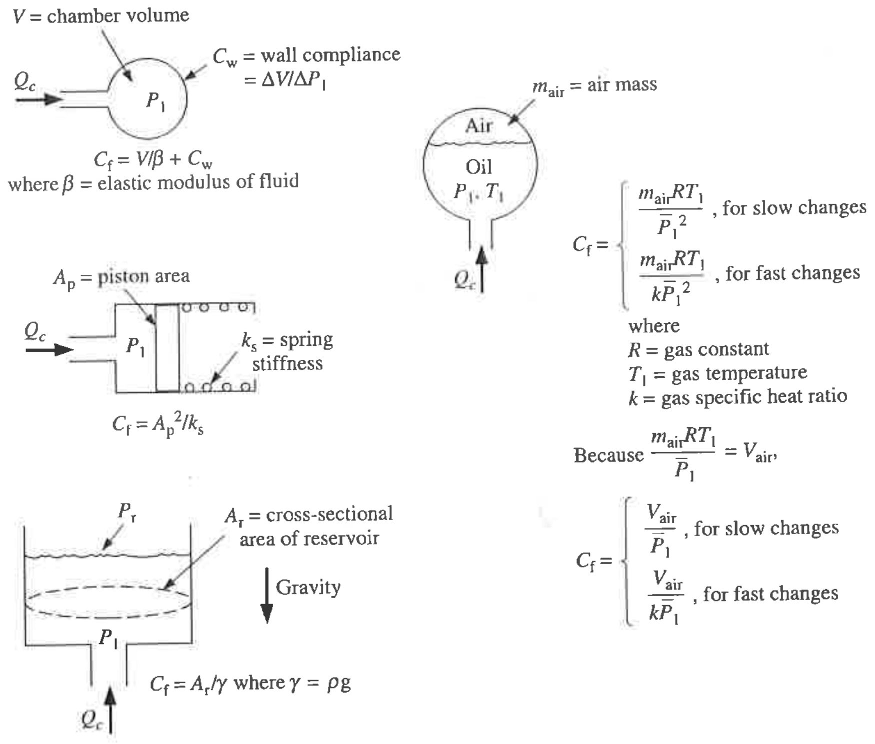


Figure 9.2 Typical fluid capacitors and their capacitances.

Source:
Dynamic Modeling and Control of Engineering Systems
Shearer, Kulakowski, Gardner

$$C_f = \frac{A_p^2}{K_s}$$

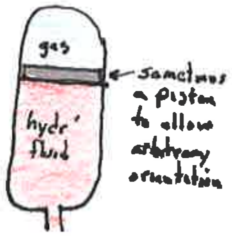
$$\frac{dP}{dt} = \frac{Q_f}{C_f} = \frac{Q_f \cdot k_s}{A_p^2} = \frac{1\ \text{in}^3}{\text{s}} \cdot \frac{1}{25\ \text{in}^4} \cdot \frac{100\ \text{lbf}}{\text{in}} = \boxed{4\ \frac{\text{psi}}{\text{s}}}$$

Ex: After 10 seconds, how much energy is stored? $P(t=0)=0$

$$P(10s) = 4\ \frac{\text{psi}}{\text{s}} \cdot 10\text{s} = 40\,\text{psi}$$

$$E = \frac{C_f}{2}P^2 = \frac{25\ \text{in}^4}{2} \cdot \frac{40^2\ \text{lbf}^2}{\text{in}^4} \cdot \frac{\text{in}}{100\ \text{lbf}} = 200^{\ \text{in lbf}} \approx 22^{\ \text{J}} \approx 60^{\ \text{watt}}\ \text{bulb for } 1/3 \text{ second.}$$

Accumulator (Gas charged)



1) Supply p and Volume to assist pump

$$V_s = V_e - V_f \quad \text{(gas volume not hyd!)}$$

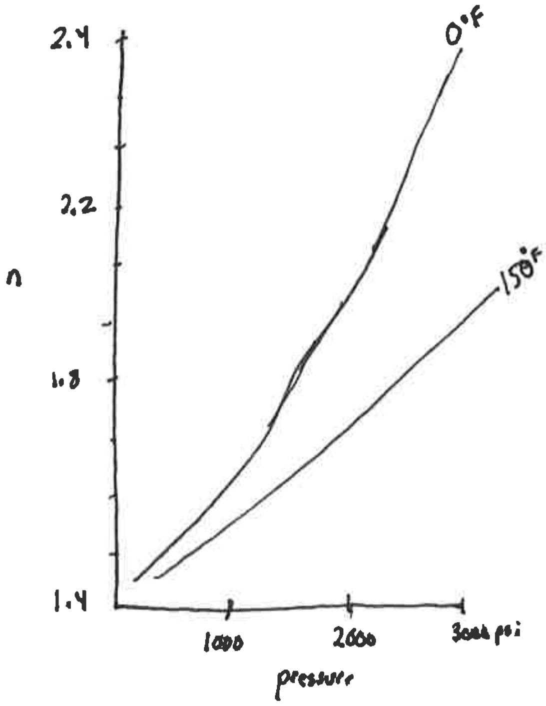
Polytropic gas

$$p_f V_f^n = p_e V_e^n$$

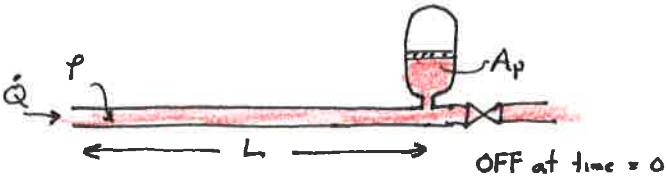
Substitute to give

$$V_e = V_s \cdot \frac{1}{1 - \left(\frac{p_e}{p_f}\right)^{1/n}} \beta \qquad \text{where } \beta \approx 1.1 \text{ piston}$$

n depends on process



2) To provide shock mitigation



Fluid motion → kinetic energy → compress gas

$$\frac{1}{2}\rho V_{volume}\cdot V^2 = -\int_{V_f}^{V_e} p\,dV$$

simplifies to

$$V_e = \frac{(n-1)\rho\dot{Q}^2 L}{2A_p p_e\left(\left(\frac{p_f}{p_e}\right)^{n-1/n}-1\right)}$$

Ex: What size of accumulator should be used for the previous 36 gpm 3500 psi 30 ft ½ in ID pipe? The max pressure rise is 500 psi. Hot system ≈ 150°F

$$\underbrace{V_e\cdot A_p}_{\text{we can choose area and volume}} = \frac{(n-1)\rho\dot{Q}^2 L}{2p_e\left(\left(\frac{p_f}{p_e}\right)^{\frac{n-1}{n}}-1\right)}$$

$\rho = 1.71\ \frac{slug}{ft^3}$

$\dot{Q} = 36$ gpm

150°F → $n \approx 2.0$

$$\frac{p_f}{p_e} = \frac{3500+500}{3500} = 1.1428 \Rightarrow \left(\frac{p_f}{p_e}\right)^{\frac{n-1}{n}} = 1.1428^{\frac{1}{2}} = 1.07$$

$$\left(\frac{p_f}{p_e}\right)^{\frac{n-1}{n}} - 1 = 0.07$$

$$V_e\cdot A_p = \frac{(2-1)}{}\left|\frac{1.71\ \cancel{slug}}{\cancel{ft^3}}\right|\frac{36^2\ \cancel{gal^2}}{\cancel{min^2}}\left|\frac{30\ \cancel{ft}}{}\right|\frac{1}{2}\left|\frac{in^2}{3500\ \cancel{lbf}}\right|\frac{}{0.07}\left|\frac{231^2\ \cancel{in^3}^2\ in^3}{\cancel{gal^2}}\right|\frac{\cancel{min^2}}{60^2\ \cancel{s^2}}\left|\frac{\cancel{lbf}\,\cancel{s^2}}{\cancel{slug}\,\cancel{ft}}\right|\frac{\cancel{ft^3}}{12^3\ \cancel{in^3}}$$

$$= 1.18\ in^5$$

- As the allowable Δp decreases (less pressure rise), the volume increases.

  at 100 psi rise $V_e\cdot A_p = 5.7\ in^5$

- Alternatively, if we rely on the flexibility of the hydraulic hose/pipe where $V_e\cdot A_p \approx 0$, the pressure rise is large.