Design and Modeling of Fluid Power Systems

ME 597/ABE 591 - Lecture 13

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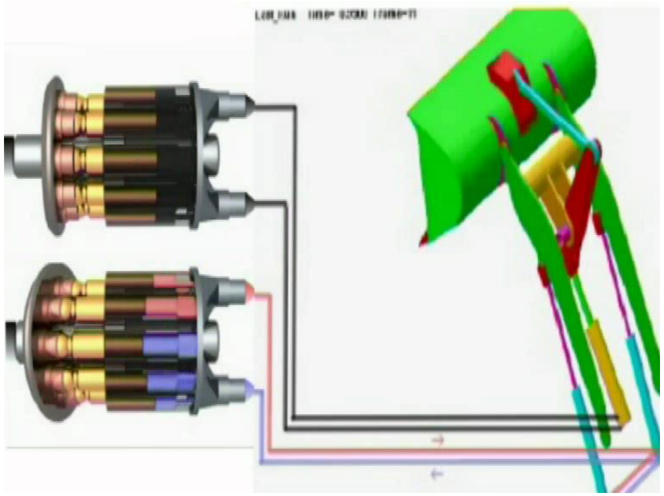
Displacement Controlled Systems

- Design and modeling of electrohydraulic pump control system

- Displacement controlled linear actuator

- Displacement controlled rotary actuator

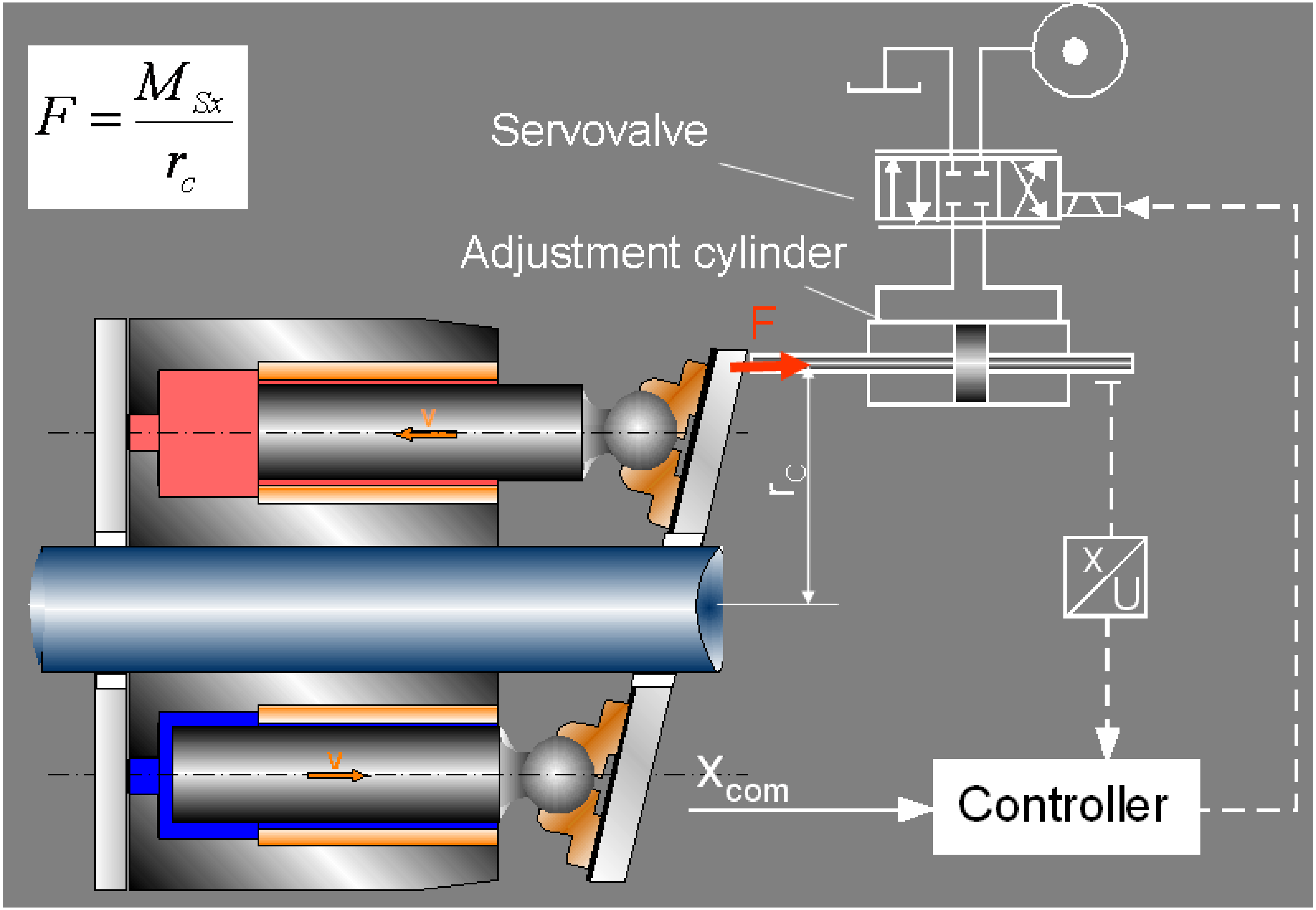
- Secundary controlled actuator



Swash plate control system

Electrohydraulic swash plate control system

$$F = \frac{M_{Sx}}{r_c}$$



Swash plate moment M_{Sx}

$$M_{Sx} = \frac{R}{\cos^2\beta} \cdot \sum_{i=1}^{z} F_{AKi} \cos\varphi_i$$

$f(p_i)$ instantaneous cylinder pressure

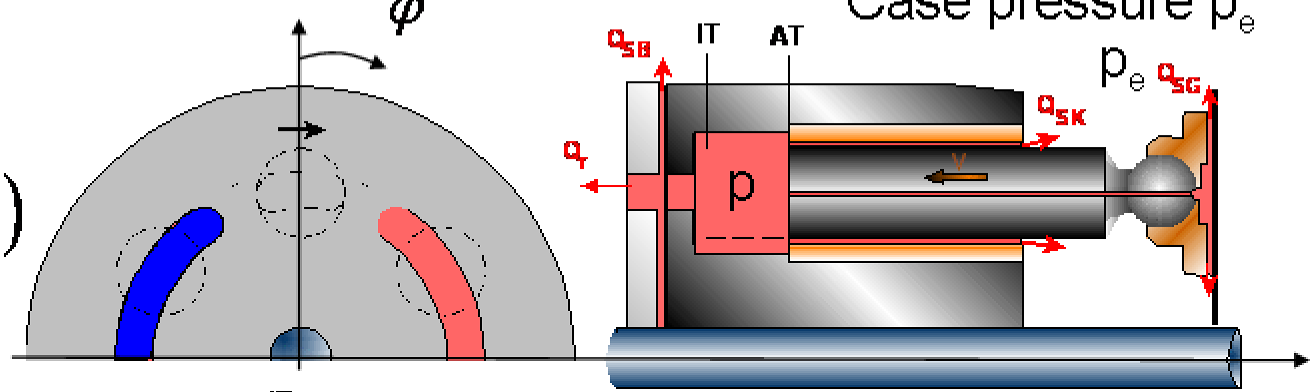
$$Q_S = f(\Delta p, n, V_i, \mu)$$

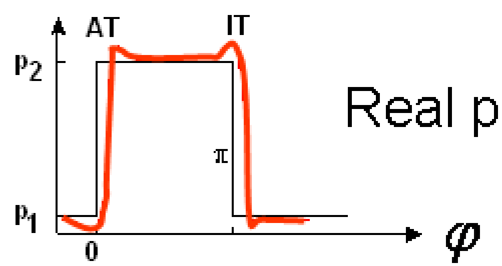
$$F_{AK} = F_{DK} + F_{aK} + F_{TK}$$

$$\frac{dp}{d\varphi} = -\frac{K}{\omega}\left(\frac{v_K \cdot A_K - Q_r - Q_{SK} - Q_{SB} - Q_{SG}}{V_0 - s_K \cdot A_K}\right)$$

$$F_{DK} = A_K \cdot (p - p_e)$$

$$M_{Sx} = f(\Delta p, n, V_i, \mu)$$



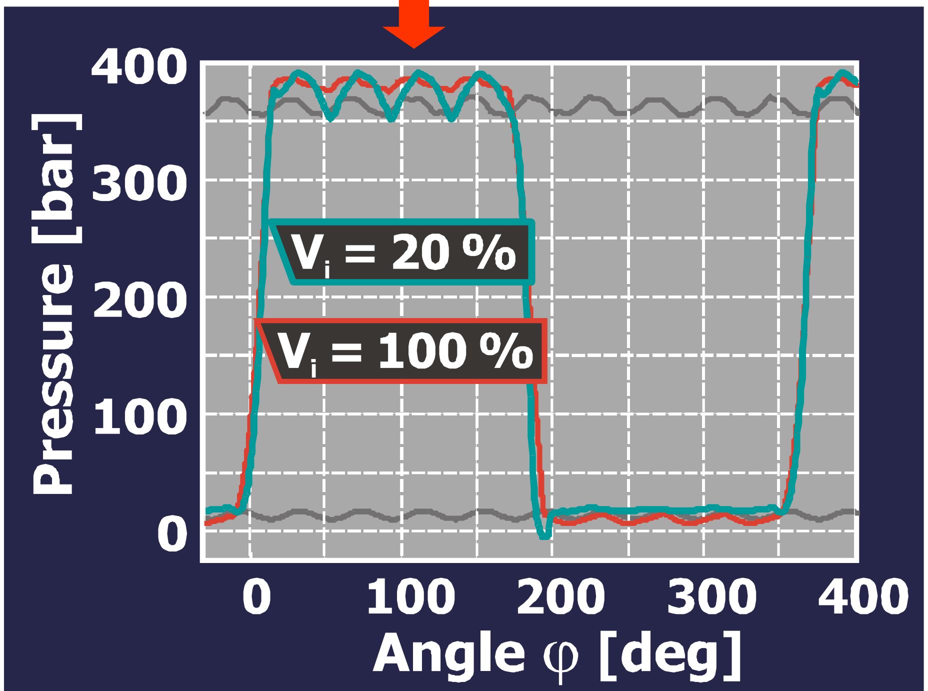


Real profile

$$d\varphi = \omega \cdot dt$$

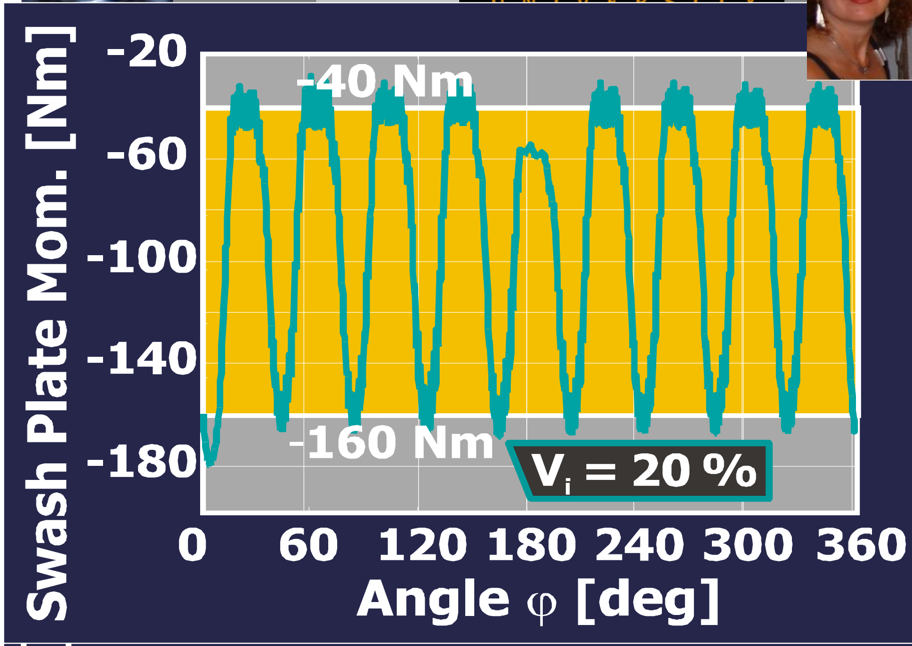
Swash plate moment M_{Sx}

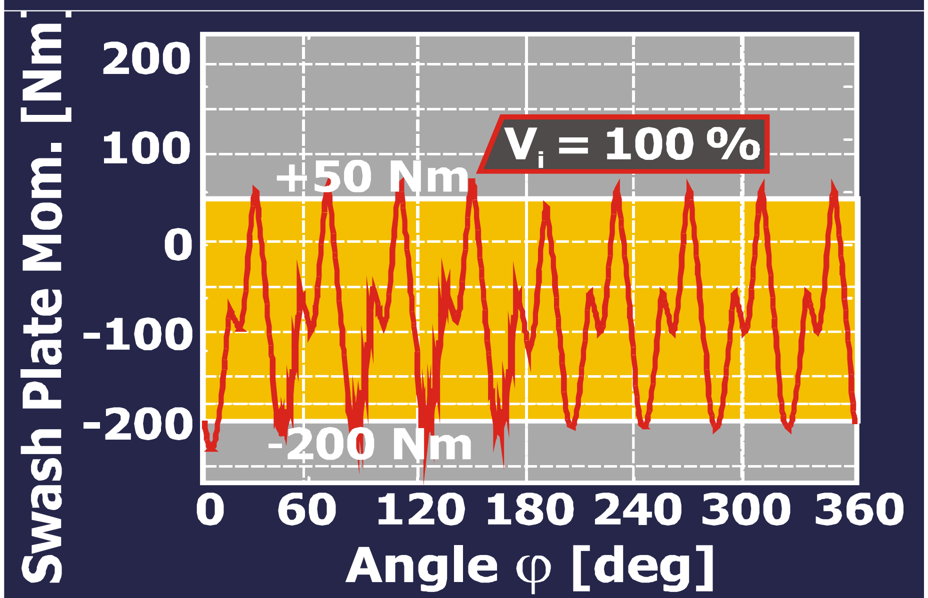
$$M_{Sx} = \frac{R}{\cos^2 \beta} \cdot \sum_{i=1}^{z} F_{AKi} \cdot \cos \varphi_i$$



Δp = 300 bar
n = 2000 rpm

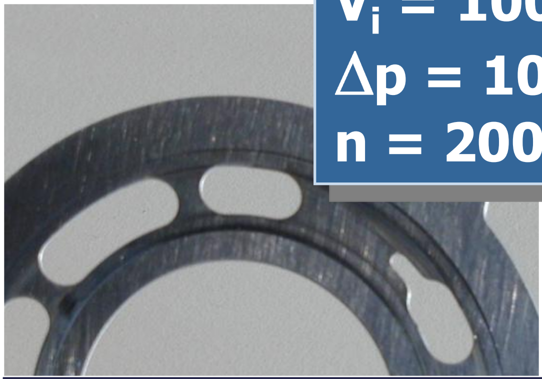
$$M_{Sx} = f(\Delta p, n, V_i, \mu)$$

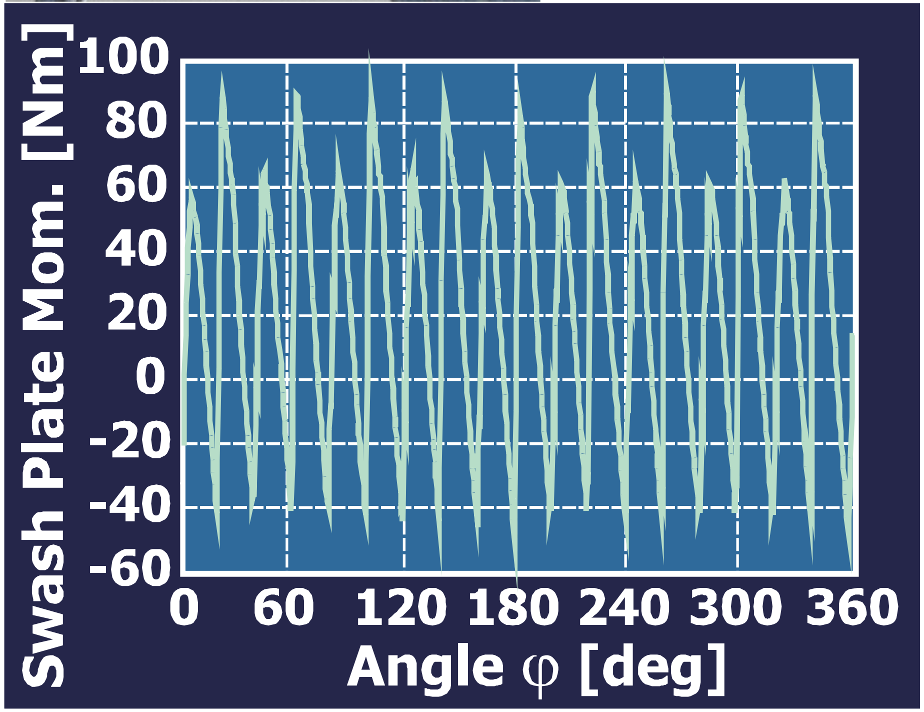


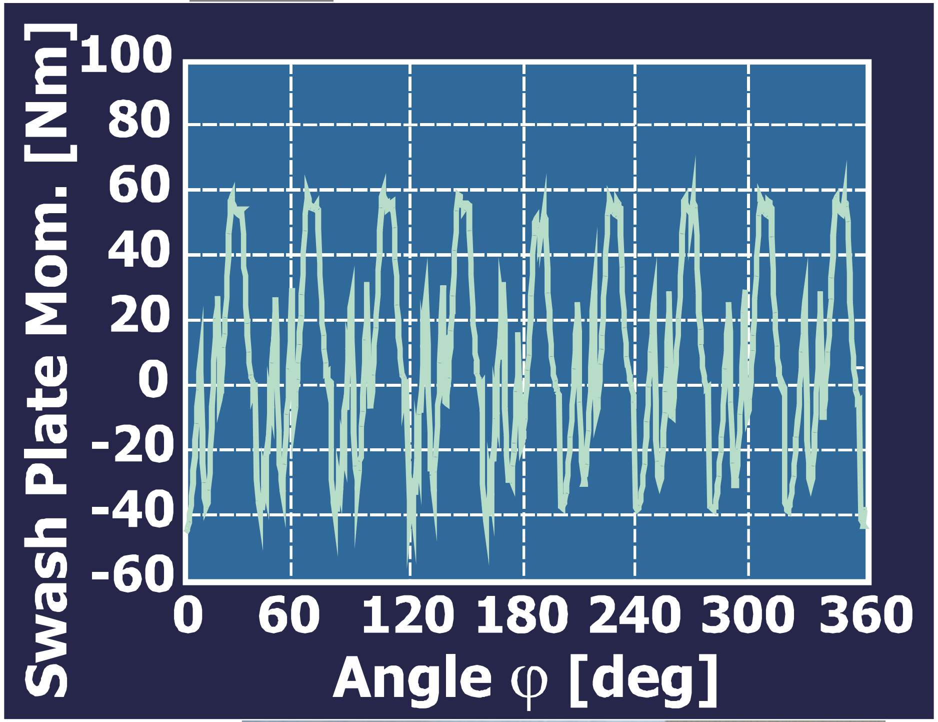


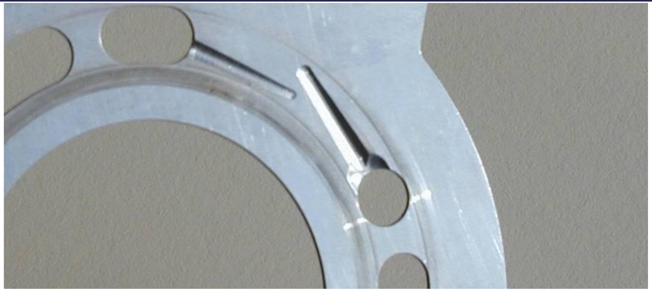
Influence of the Valve Plate Design

V_i = 100 %
Δp = 100 bar
n = 2000 rpm







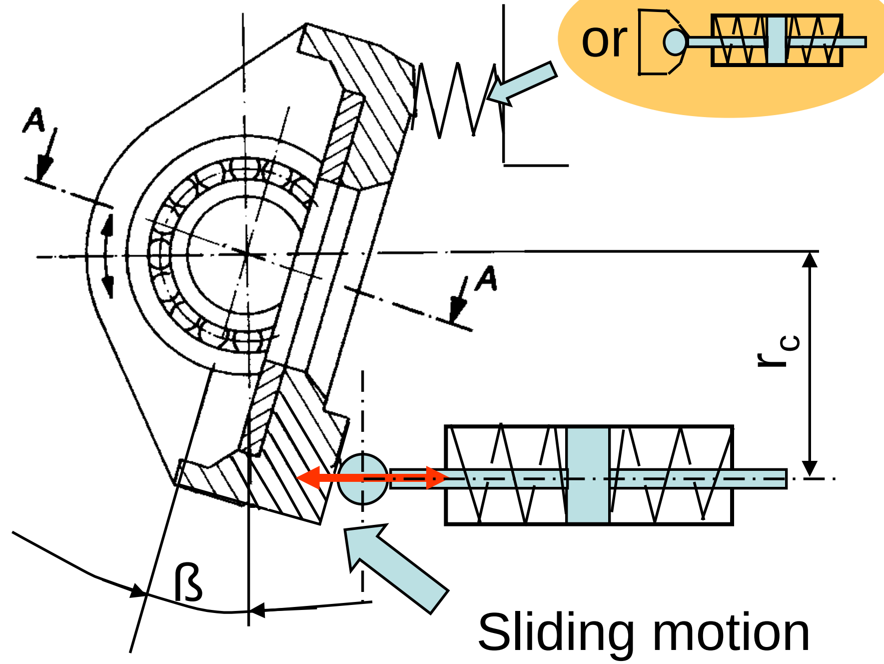


$$M_{Sx} = f(\varphi)$$

Swash Plate Control Mechanism

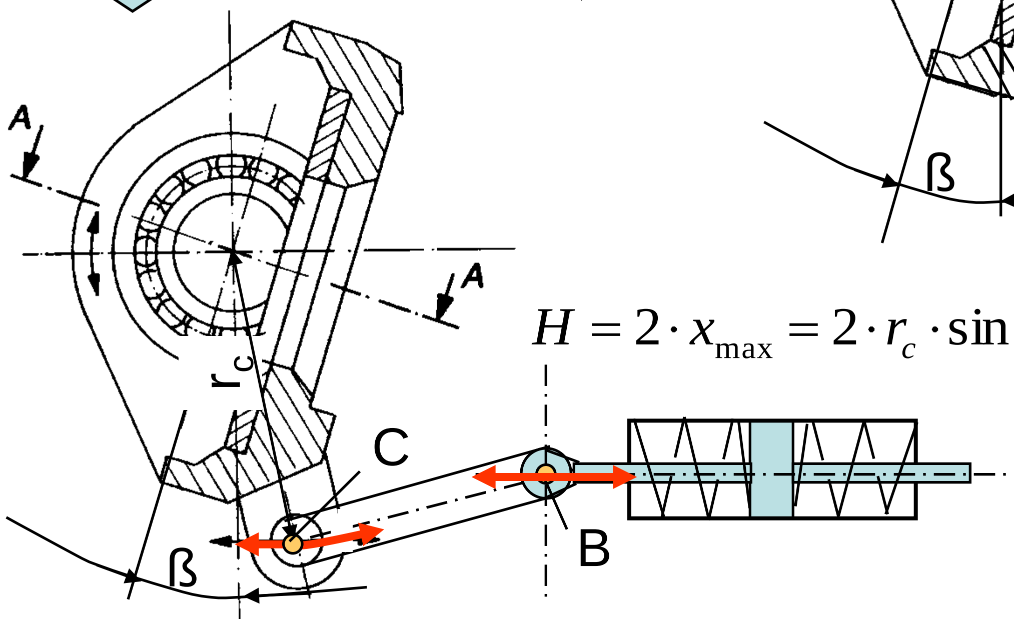
$$H = 2 \cdot x_{\max} = 2 \cdot r_c \cdot \tan \beta_{\max} \qquad (2)$$

Connecting rod less



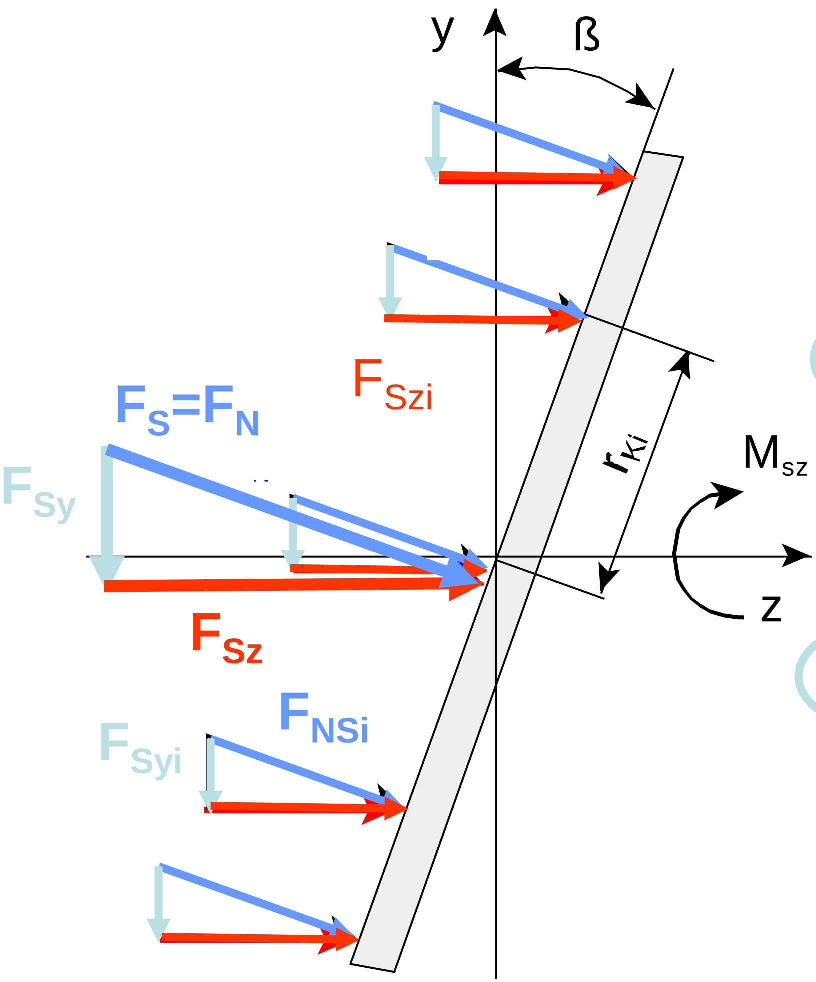
Sliding motion

Connecting rod joint



$$H = 2 \cdot x_{\max} = 2 \cdot r_c \cdot \sin \beta_{\max} \qquad (1)$$

Swash Plate Bearing Design Examples



$$F_S = \sqrt{F_{Sy}^2 + F_{Sz}^2} = F_N = \sum_{i=1}^{z} F_{NSi}$$

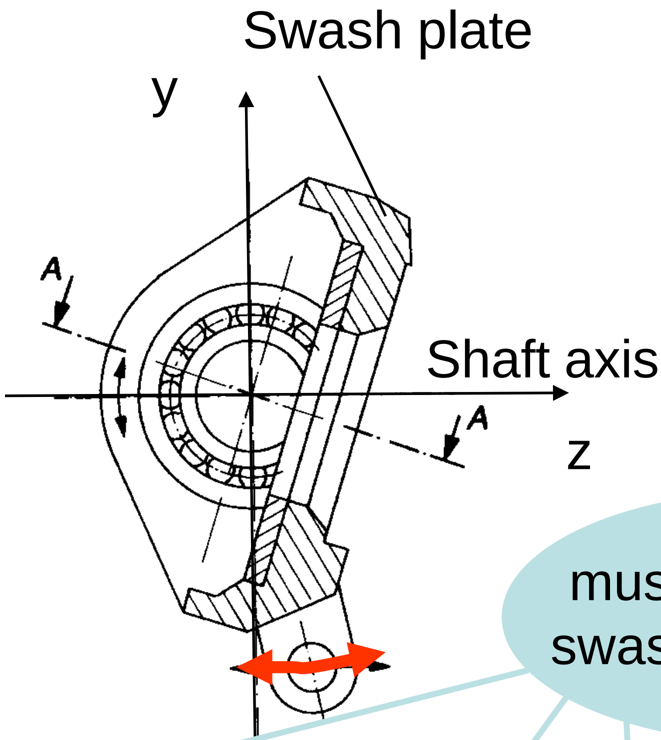
$$M_{Sy} = -\sum_{i=1}^{z} F_{Szi} \cdot x_{Si} = -R \cdot \sum_{i=1}^{z} F_{AKi} \cdot \sin\varphi_i$$

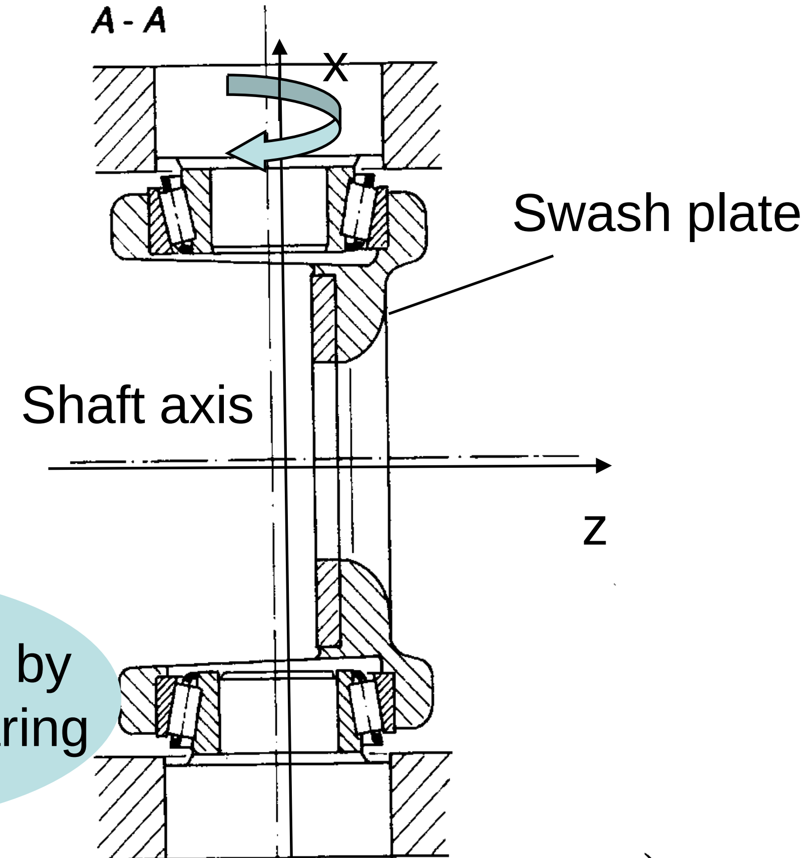
$$M_{Sz} = -R \cdot \left(\sum_{i=1}^{z} F_{TSi} + \tan\beta \cdot \sum_{i=1}^{z} F_{AKi} \cdot \sin\varphi_i \right)$$

must be carried by swash plate bearing

Swash Plate Bearing – Design Examples

In case of variable displacement pumps very often radial bearings are used





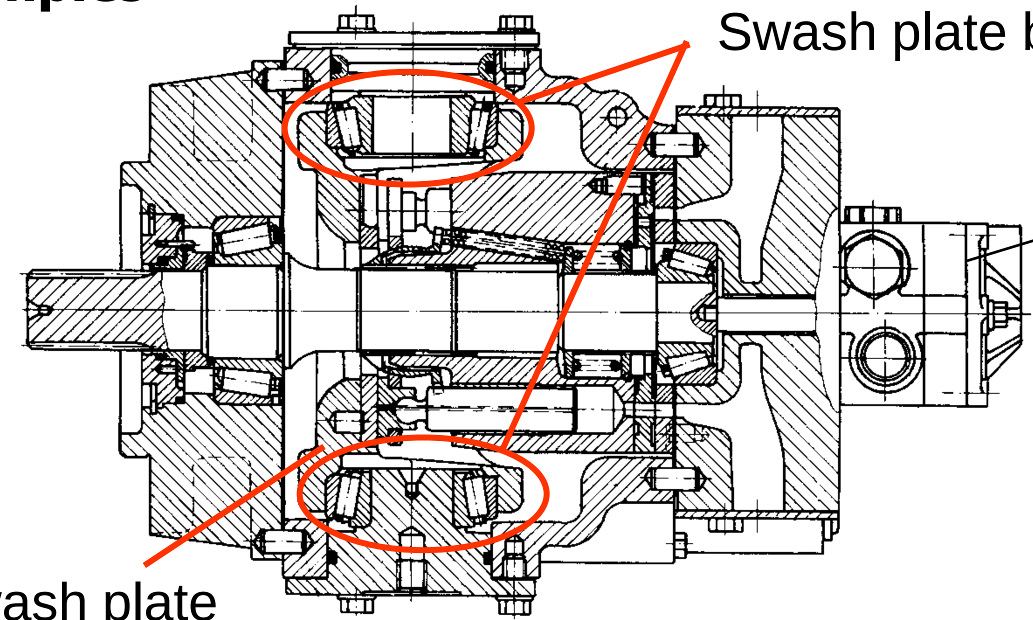
must be carried by swash plate bearing

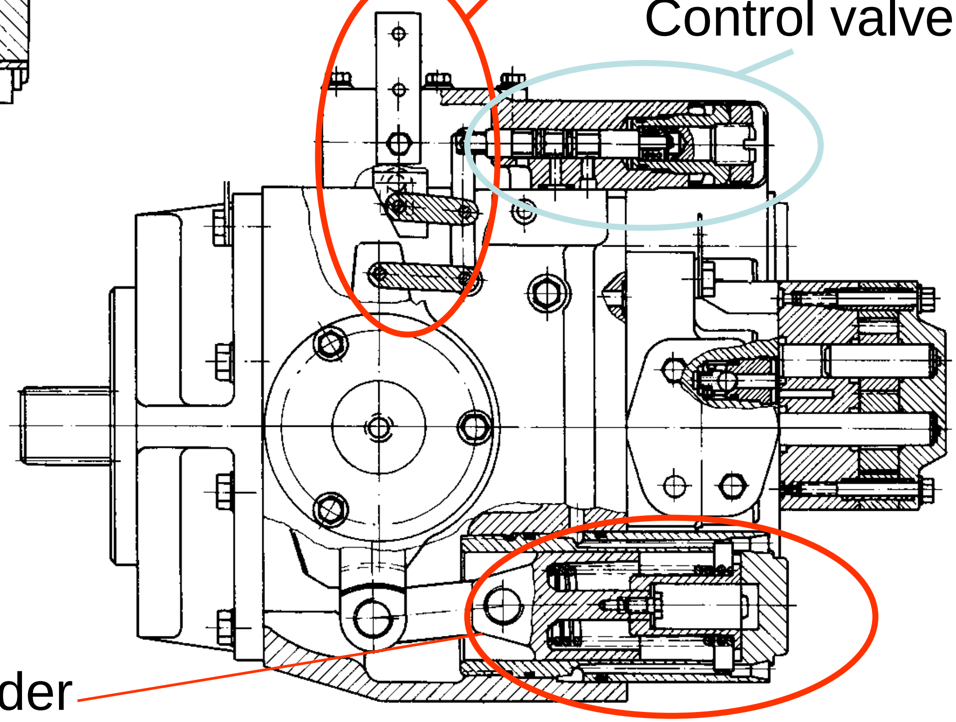
$$F_S = \sqrt{F_{Sy}^2 + F_{Sz}^2}$$

$$M_{Sz} = -R \cdot \left(\sum_{i=1}^{z} F_{TSi} + \tan\beta \cdot \sum_{i=1}^{z} F_{AKi} \cdot \sin\varphi_i \right)$$

$$M_{Sy} = -\sum_{i=1}^{z} F_{Szi} \cdot x_{Si} = -R \cdot \sum_{i=1}^{z} F_{AKi} \cdot \sin\varphi_i$$

Swash Plate Bearing – Design Examples





Design of swash plate control system

Swash plate control cylinder $H = 2 \cdot x_{max} = 2 \cdot r_c \cdot \tan \beta_{max}$

m… moveable mass

$$m \cdot \frac{d^2 x}{dt^2} + d \cdot \frac{dx}{dt} + F + c \cdot x = A(p_A - p_B)$$

$$F = \frac{M_{Sx}}{r_c}$$

d… coefficient of viscous friction

c… spring constant

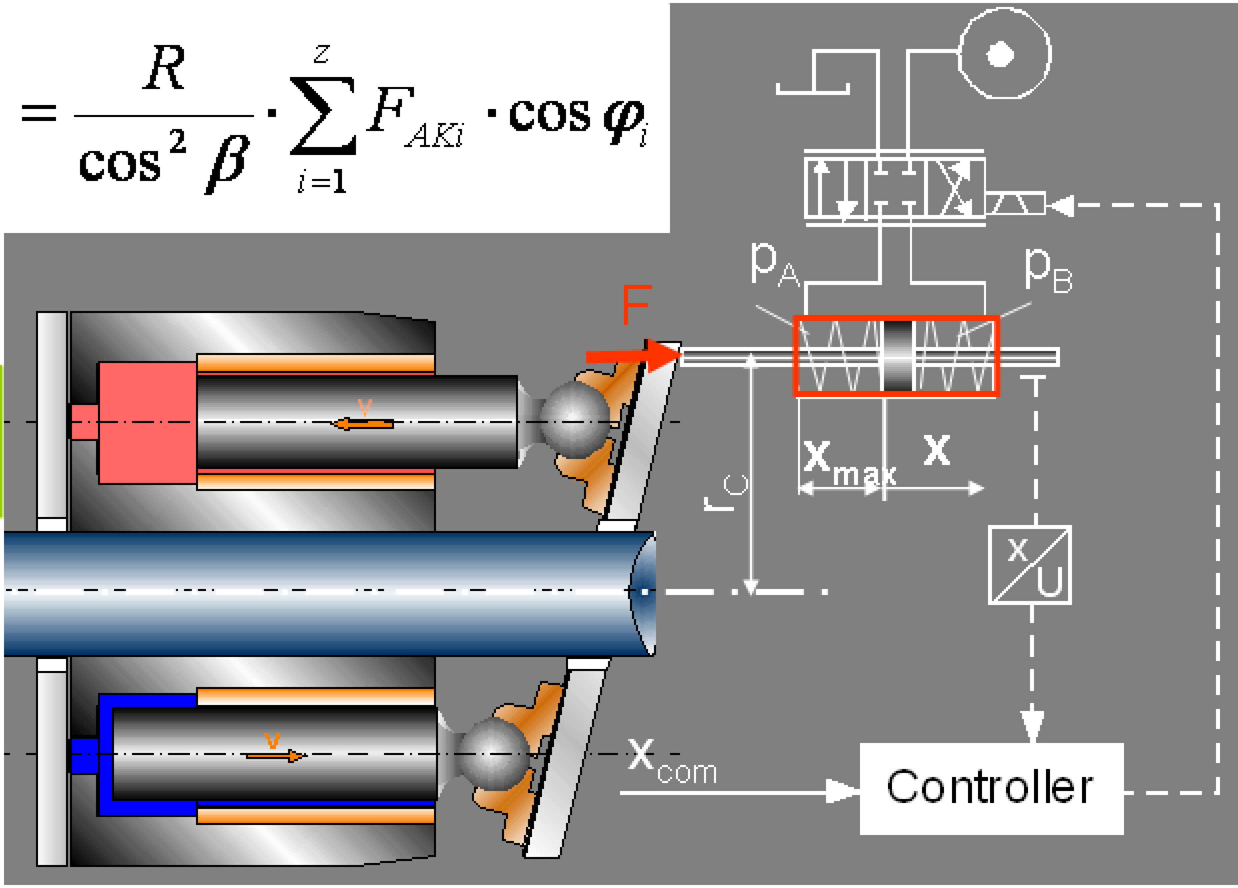
$$M_{Sx} = \frac{R}{\cos^2 \beta} \cdot \sum_{i=1}^{z} F_{AKi} \cdot \cos \varphi_i$$

$$\frac{dp_A}{dt} = \frac{1}{C_{HA}} \left(Q_A - Q_{Si} - A \cdot \frac{dx}{dt} \right)$$

$$\frac{dp_B}{dt} = \frac{1}{C_{HB}} \left(-Q_B + Q_{Si} + A \cdot \frac{dx}{dt} \right)$$

$$C_{HA} = \frac{V_{deadA} + A \cdot (x_{max} + x)}{K}$$

$$C_{HB} = \frac{V_{deadB} + A \cdot (x_{max} - x)}{K}$$



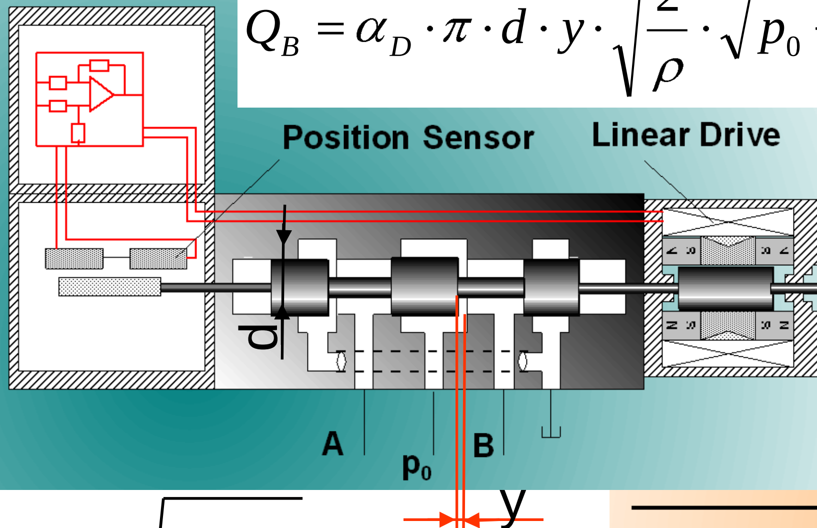
Selection of control valve

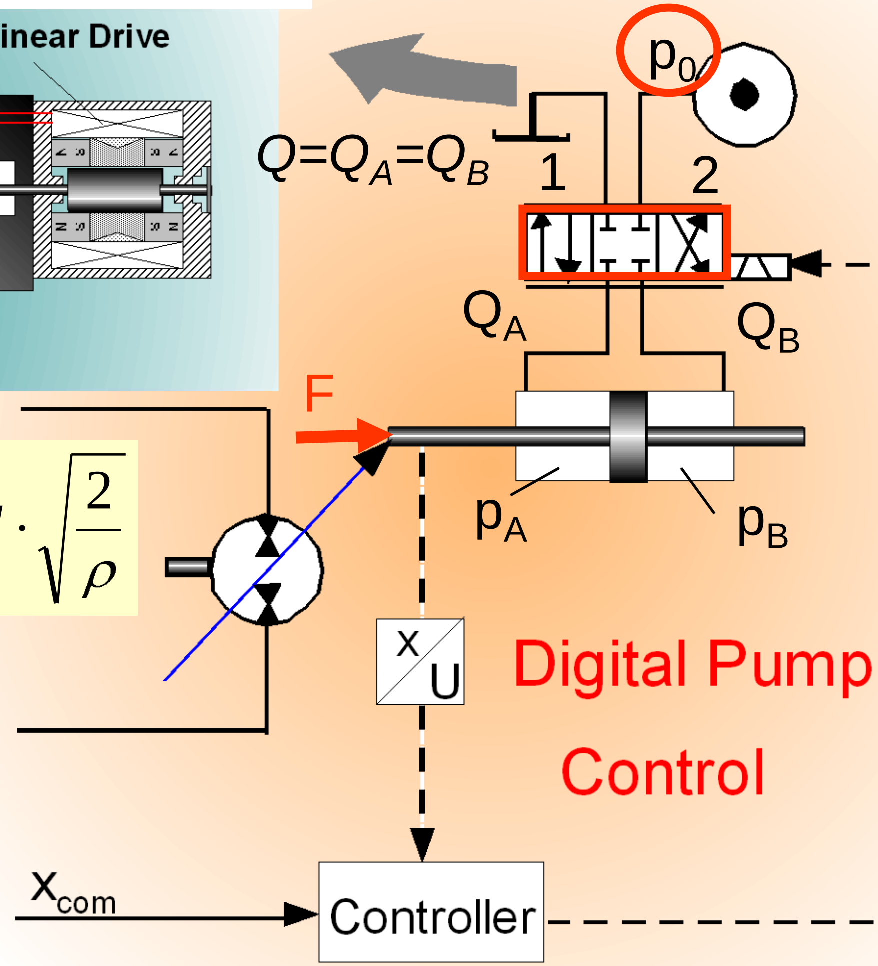
p_{nomSV}=70 bar

$$A \approx \frac{F}{0.3 \cdot p_0}$$

$$Q_B = \alpha_D \cdot \pi \cdot d \cdot y \cdot \sqrt{\frac{2}{\rho}} \cdot \sqrt{p_0 - p_B}$$

$$p_L = p_A - p_B$$





Valve size:

$$Q_{nom} = B \cdot y_{\max} \sqrt{\frac{p_{nom}}{2}}$$

$$B = \alpha_D \cdot \pi \cdot d \cdot \sqrt{\frac{2}{\rho}}$$

Required flow rate:

$$Q = A \cdot \dot{x}_{\max} \qquad \dot{x}_{\max} = \frac{2 \cdot x_{\max}}{t_{\min}}$$

$$Q_{nom} = Q \cdot \sqrt{\frac{p_{nom}}{p_0 - p_L}}$$

Design of swash plate control system

Main design requirements:

$$Q_{nom} = Q \cdot \sqrt{\frac{p_{nom}}{p_0 - p_L}}$$

Maximal flow rate

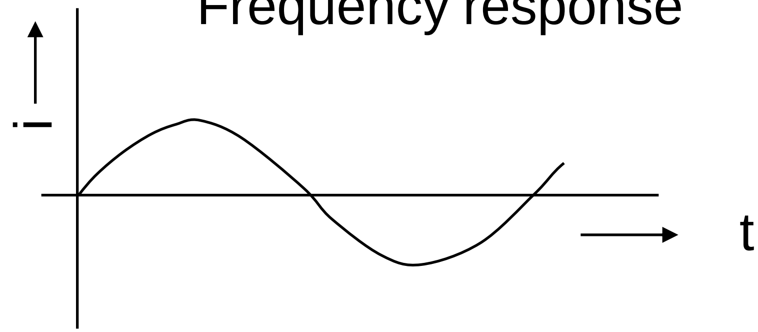
- Low power
- High bandwidth
- Min time t_{min} for $ß=0$ to $ß=ß_{max}$ (20 ms up to 200 ms)

$$x = \hat{x}\cdot\sin(\omega\cdot t)$$

$$\dot{x} = \hat{x}\cdot\omega\cdot\cos(\omega\cdot t)$$

$$\ddot{x} = -\hat{x}\cdot\omega^2\cdot\sin(\omega\cdot t)$$

Frequency response



Servovalve input current

$$i(t) = \hat{i}\cdot\sin(\omega\cdot t)$$

Design of swash plate control system

Pressure required to realized necessary acceleration

$$A \cdot p_{\ddot{x}} = m \cdot \ddot{x} \qquad \rightarrow \qquad p_{\ddot{x}} = \frac{m}{A} \cdot \hat{x} \cdot \omega^{2} \cdot \sin(\omega \cdot t)$$

Pressure required for velocity

$$A \cdot \dot{x} = Q_{nom} \cdot \sqrt{\frac{p_{\dot{x}}}{p_{nom}}} \qquad p_{\dot{x}} = \frac{p_{nom}}{Q_{nom}^{2}} \cdot \left(\hat{x} \cdot \omega \cdot \cos(\omega \cdot t)\right)^{2} \cdot A^{2}$$

Maximal value $\boxed{\dfrac{dp_{dyn}}{dt} = 0}$

$$\frac{dp_{dyn}}{dt} = \frac{p_{nom}}{Q_{nom}^{2}} \cdot \hat{x}^{2} \cdot \omega^{3} \cdot 2 \cdot \cos(\omega \cdot t) \cdot \sin(\omega \cdot t) \cdot A^{2} + \frac{m}{A} \cdot \hat{x} \cdot \omega^{3} \cdot \cos(\omega \cdot t) = 0$$

Maximal value $\Rightarrow$ $\boxed{\sin(\omega \cdot t) = \dfrac{m \cdot Q_{nom}^{2}}{2 \cdot p_{nom} \cdot \hat{x} \cdot A^{3}}}$ $\rightarrow$ $\boxed{Q_{nom}}$

Design of swash plate control system

Swash plate moment M_{Sx}

Supply pressure p_0

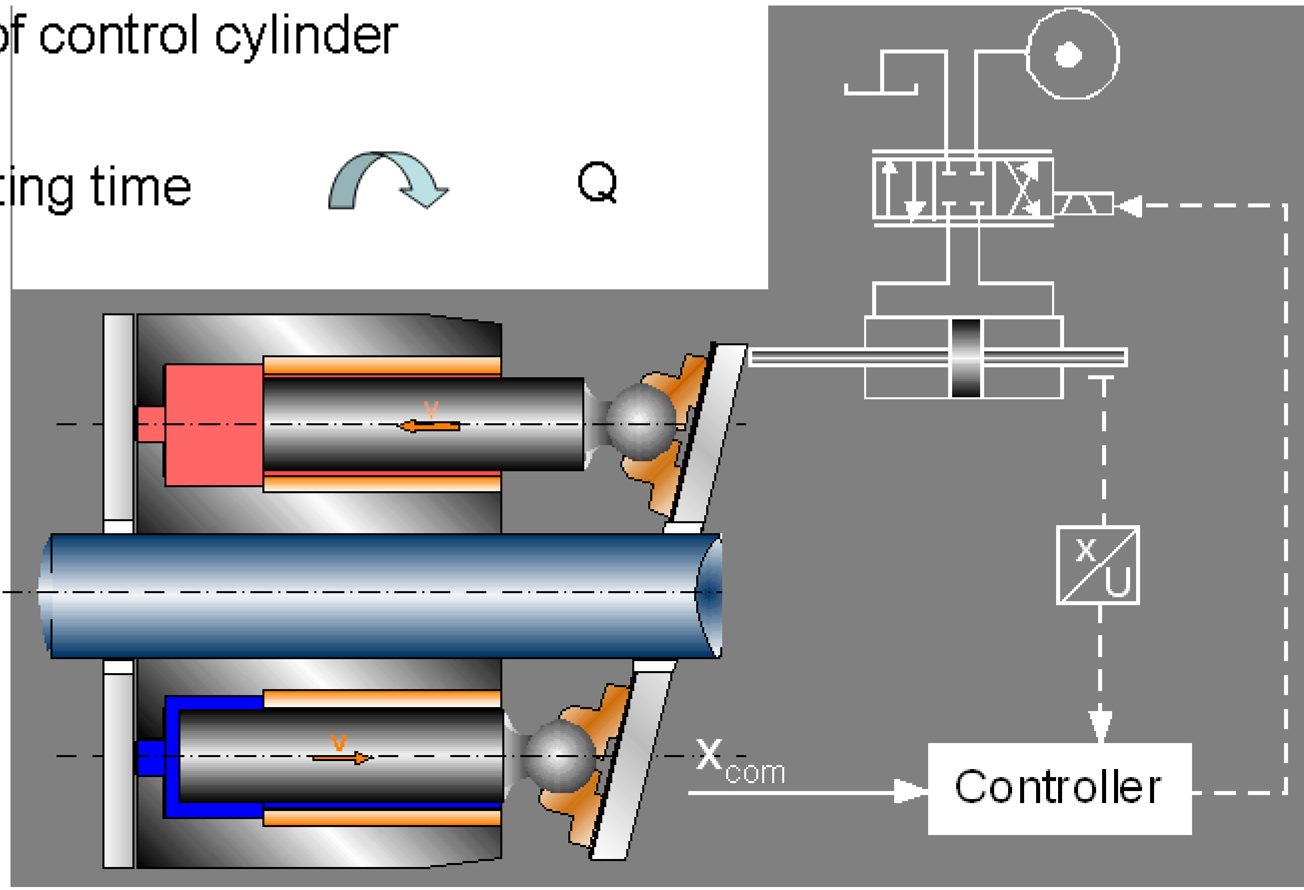
$$F = \frac{M_{Sx}}{r_c}$$

Effective piston area of control cylinder

Response time, adjusting time Q

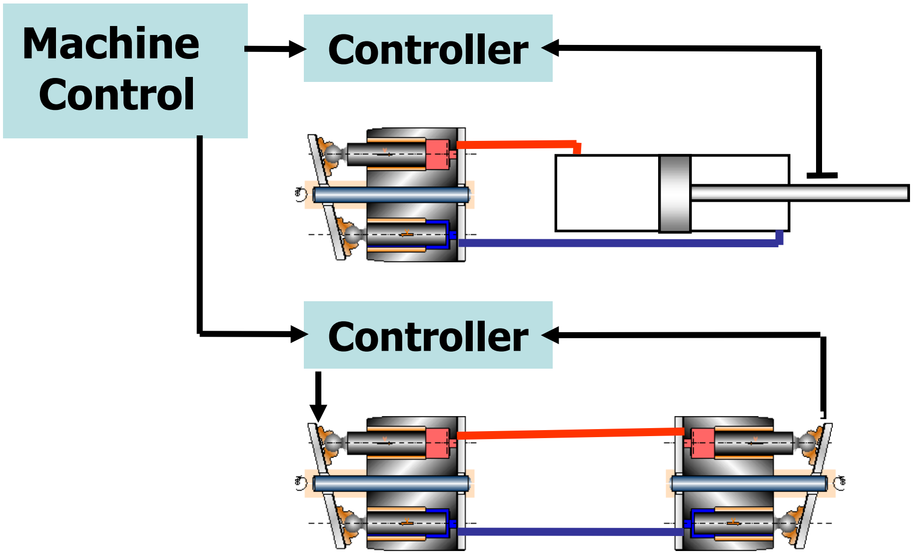
Servovalve or
proportional valve size

Swash plate controller

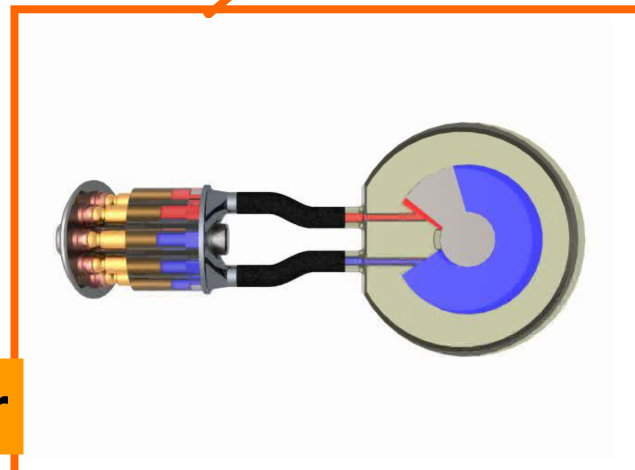


Displacement controlled actuator

- **Advanced Energy Saving Actuator Technology**



Successfully tested on laboratory test rig



New Valveless Rotary Actuator

Advanced Energy Saving Actuator Technology

Pump control instead valve control

What are the advantages ?

New Linear Actuator

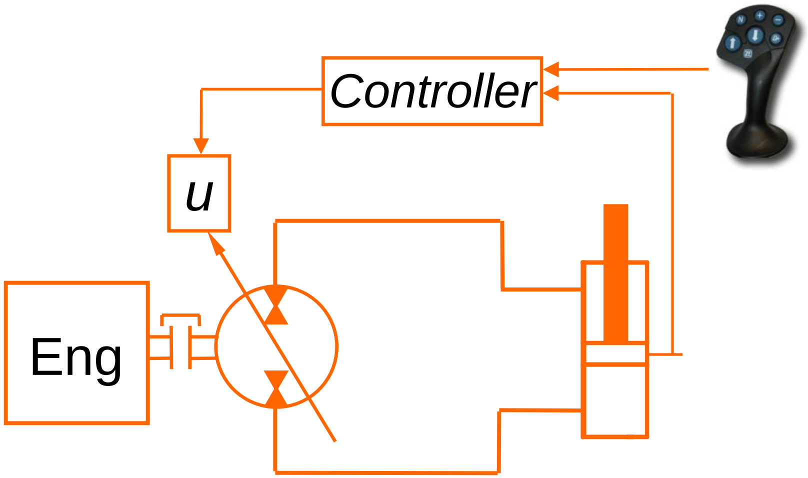
Better Utilization of Primary Energy

Energy Recovery

Easier to Control

System Simplification

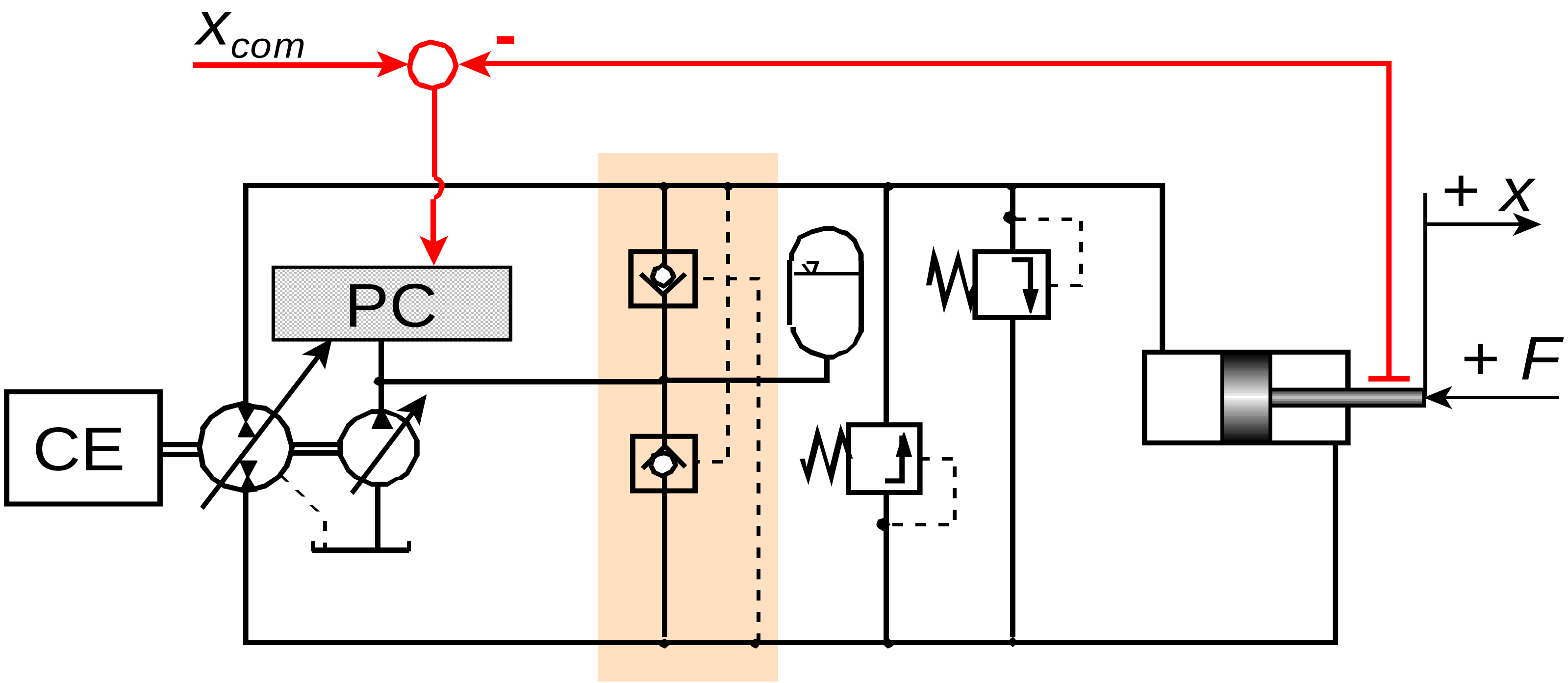
Less weight & space



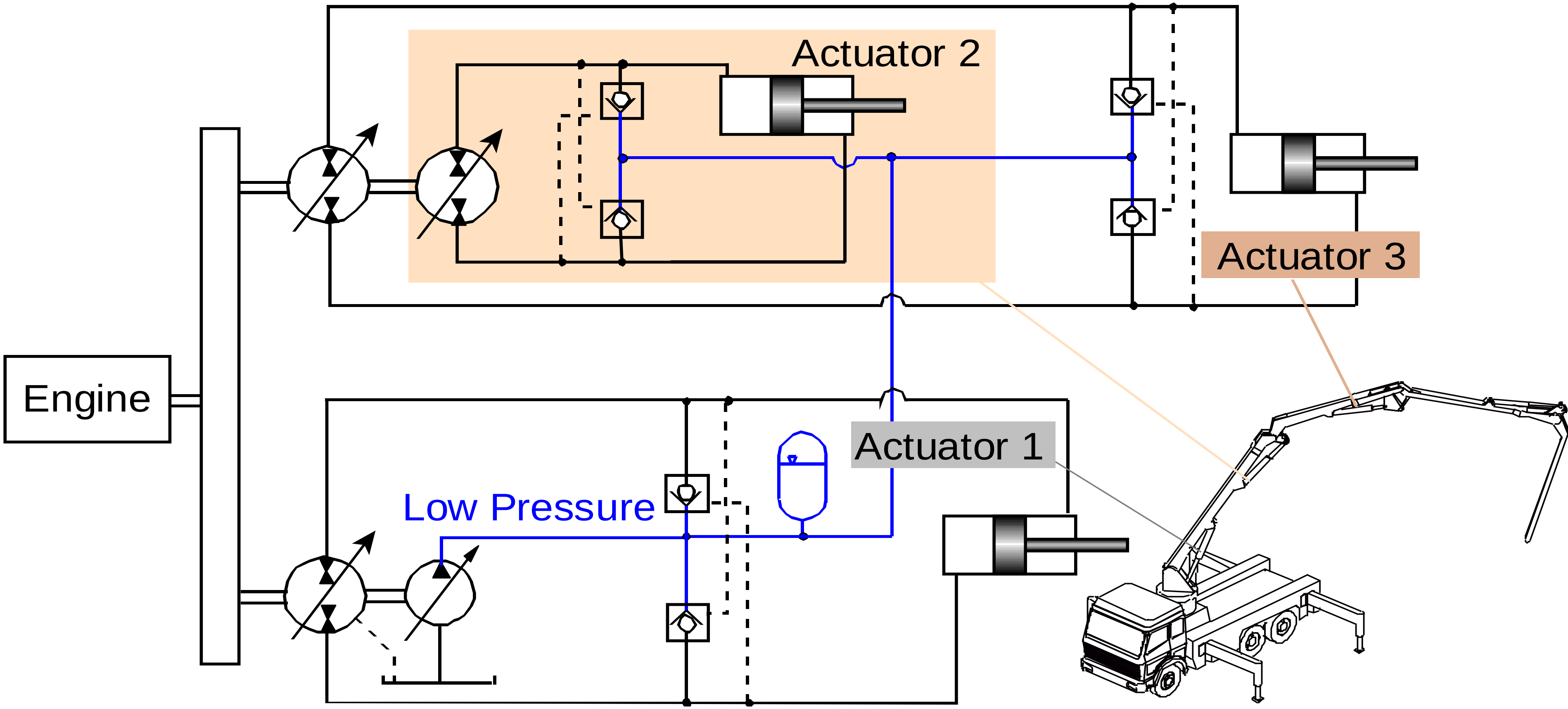
Less fuel consumption

Displacement Controlled Linear Actuator

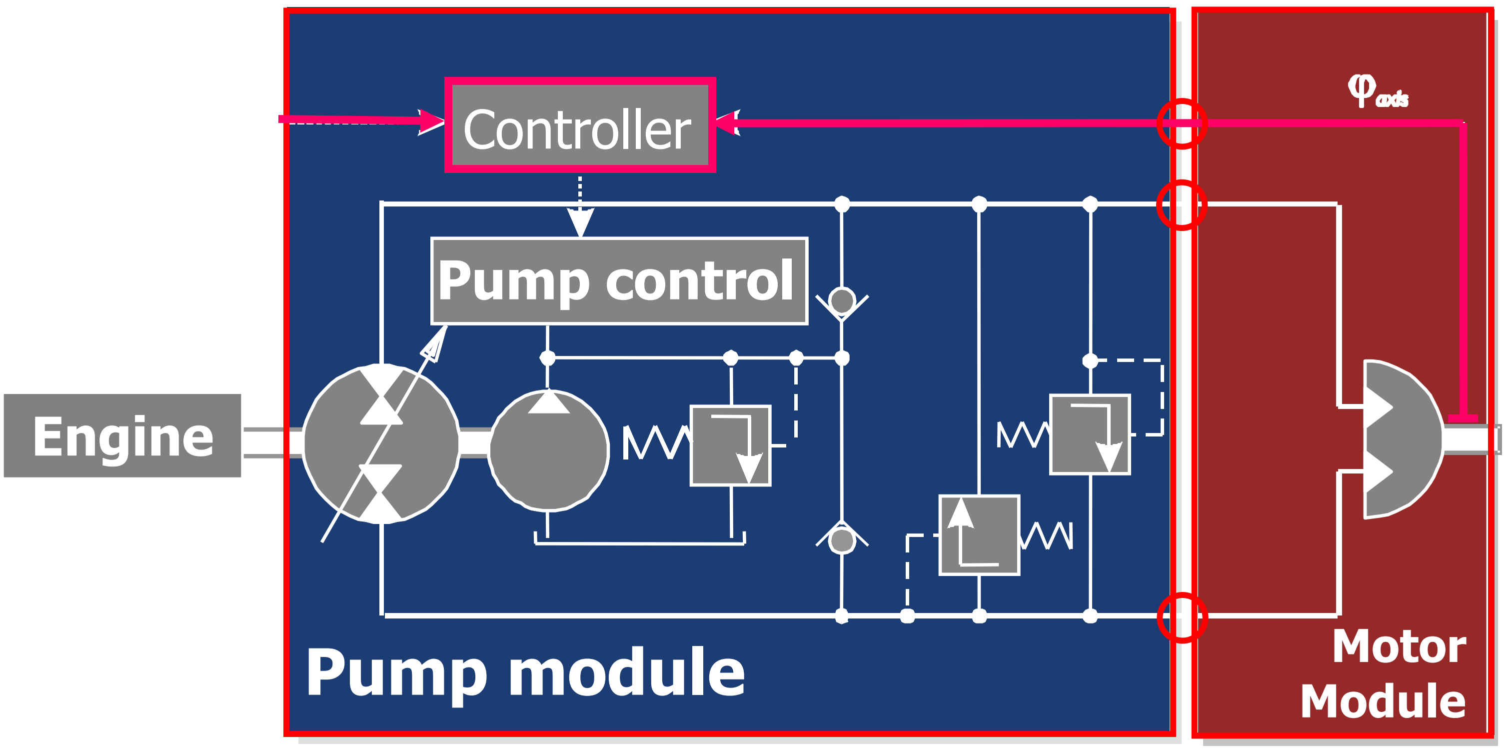
Using a Differential Cylinder



Joint Low Pressure Net for Connected Actuators



Displacement controlled rotary actuator



Possible applications?

Steering System

Stabilizer – roll control

Vehicle Suspension System – active damping control

Power Split Drive Technology using Hydrostatic Transmission

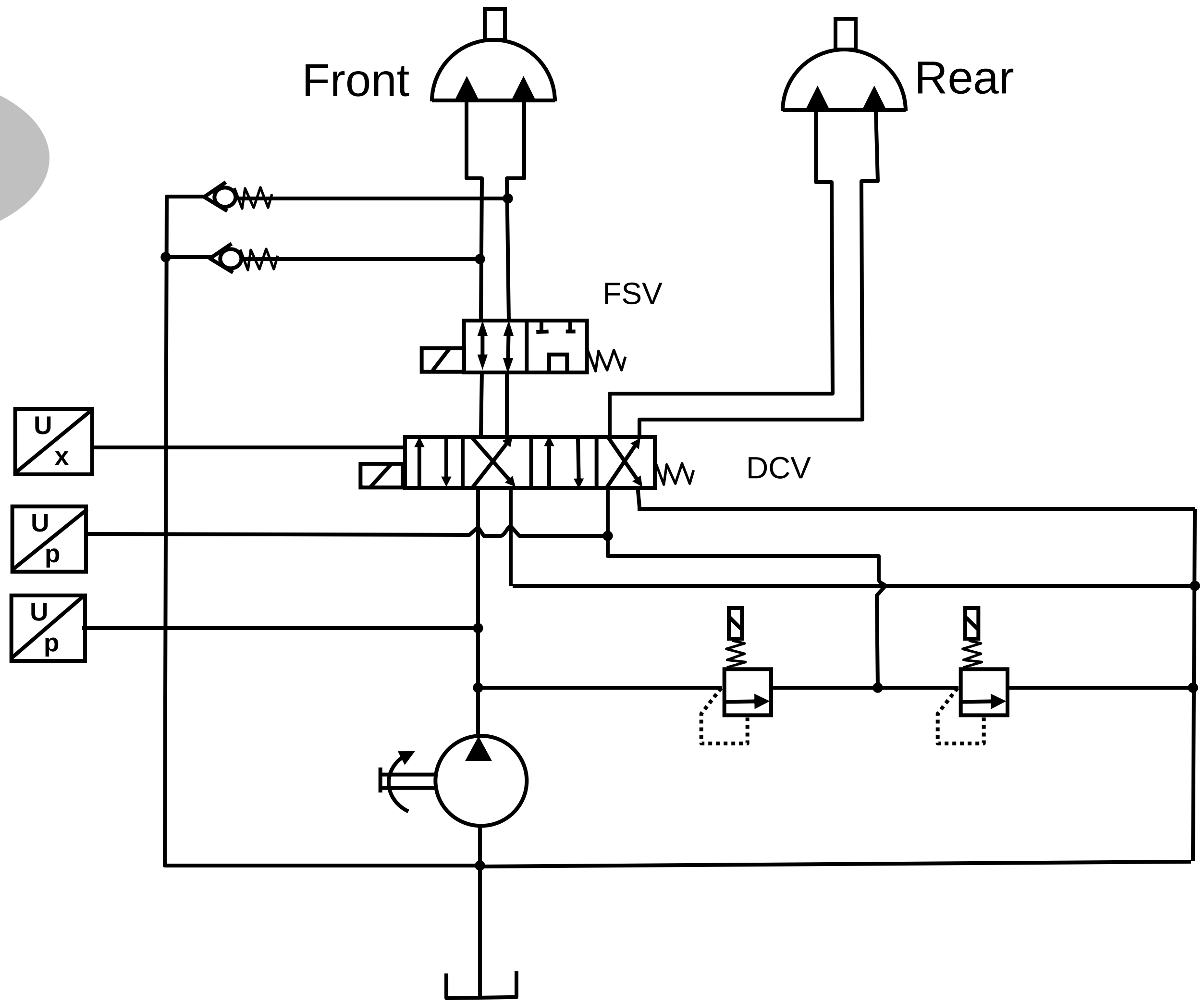
Cabriolet Roof Actuation and others

Systems with linear or rotary actuator movement

Active roll stabilization

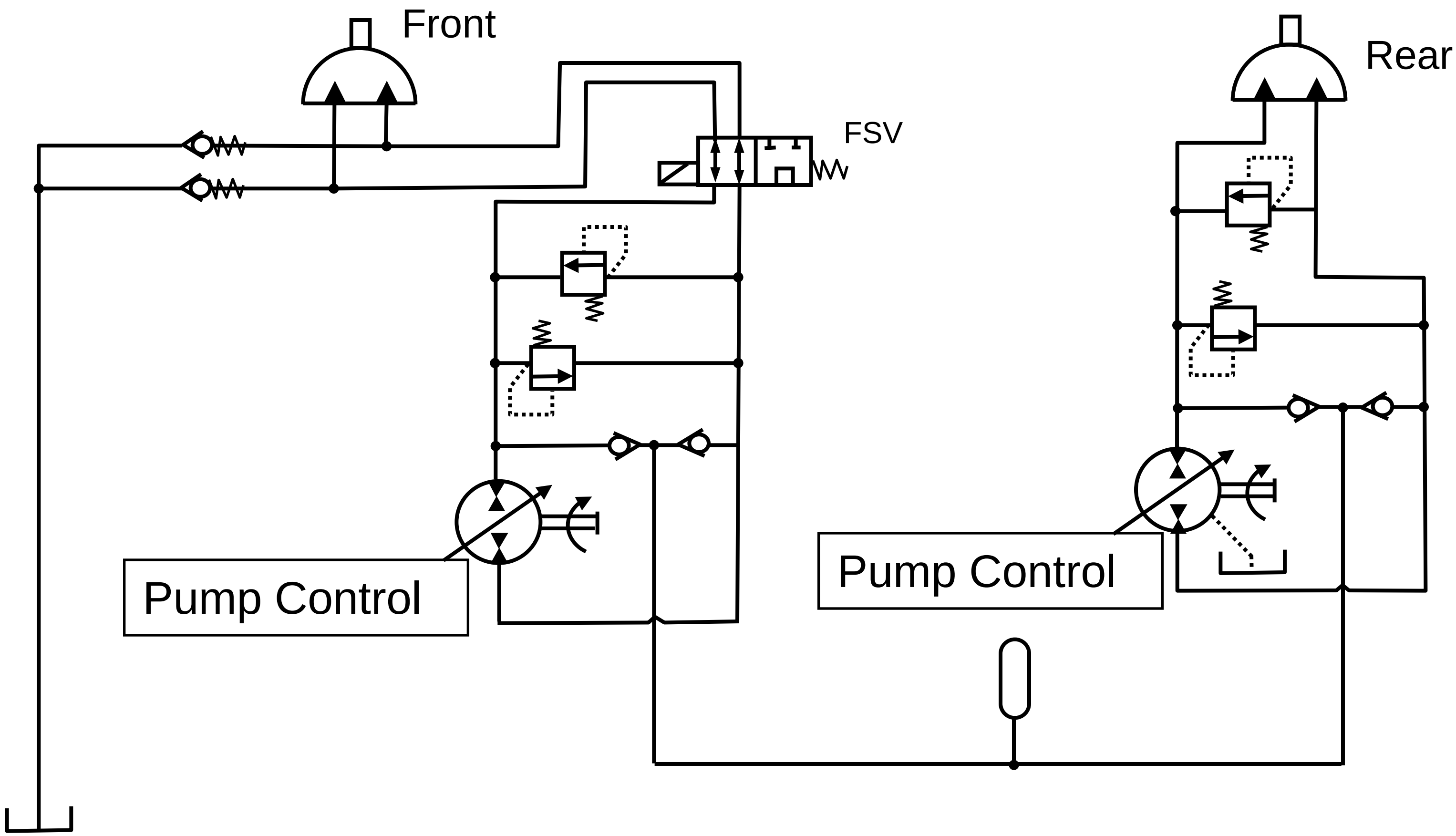
Valve controlled system

Today



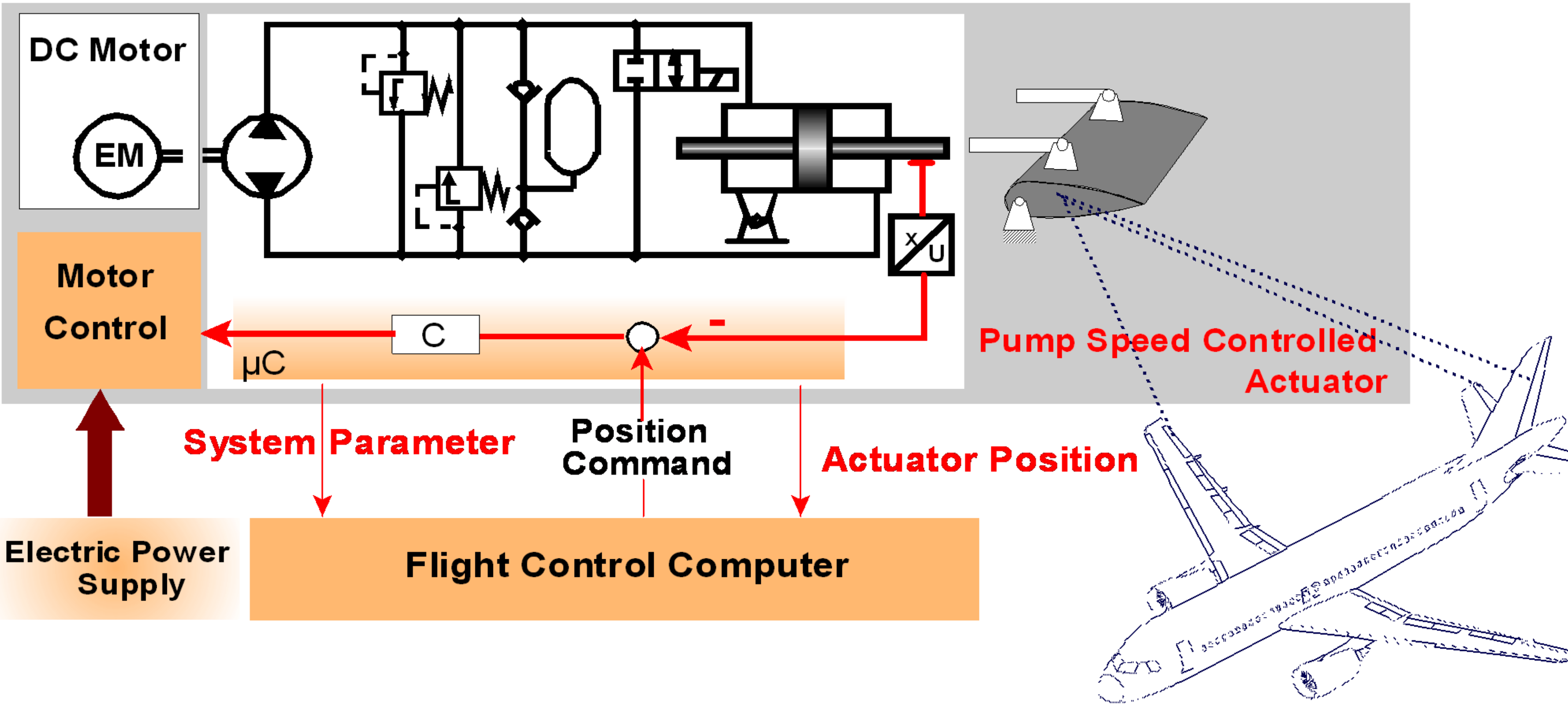
Active roll stabilization

Alternative Solution

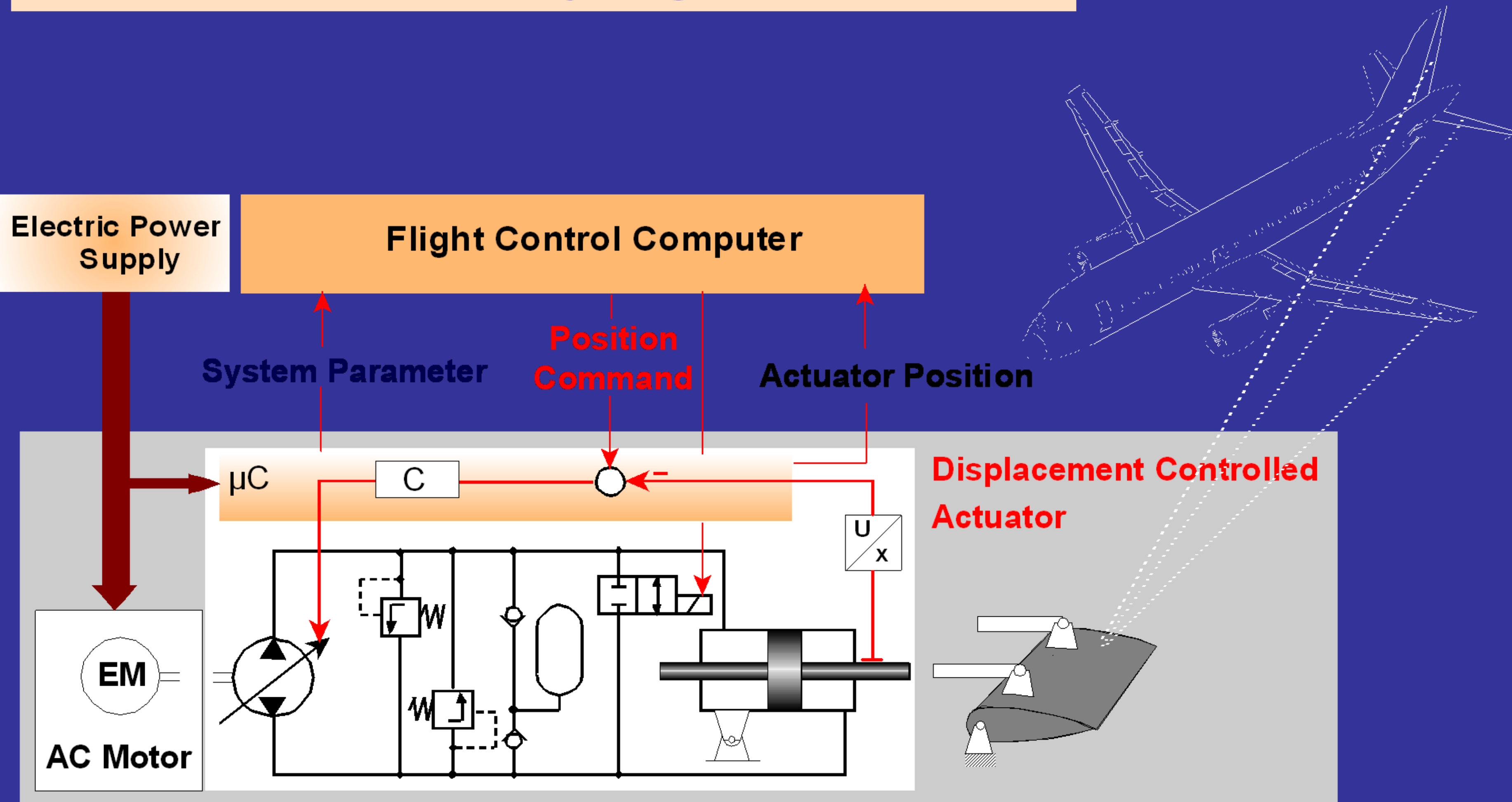


Fuel savings by valveless actuator technology

Examples - Primary Flight Control



Examples - Primary Flight Control



Secondary controlled actuator

Requires constant pressure supply

$$M_{Sx} = \frac{R}{\cos^2 \beta} \cdot \sum_{i=1}^{z} F_{AKi} \cdot \cos \varphi_i$$

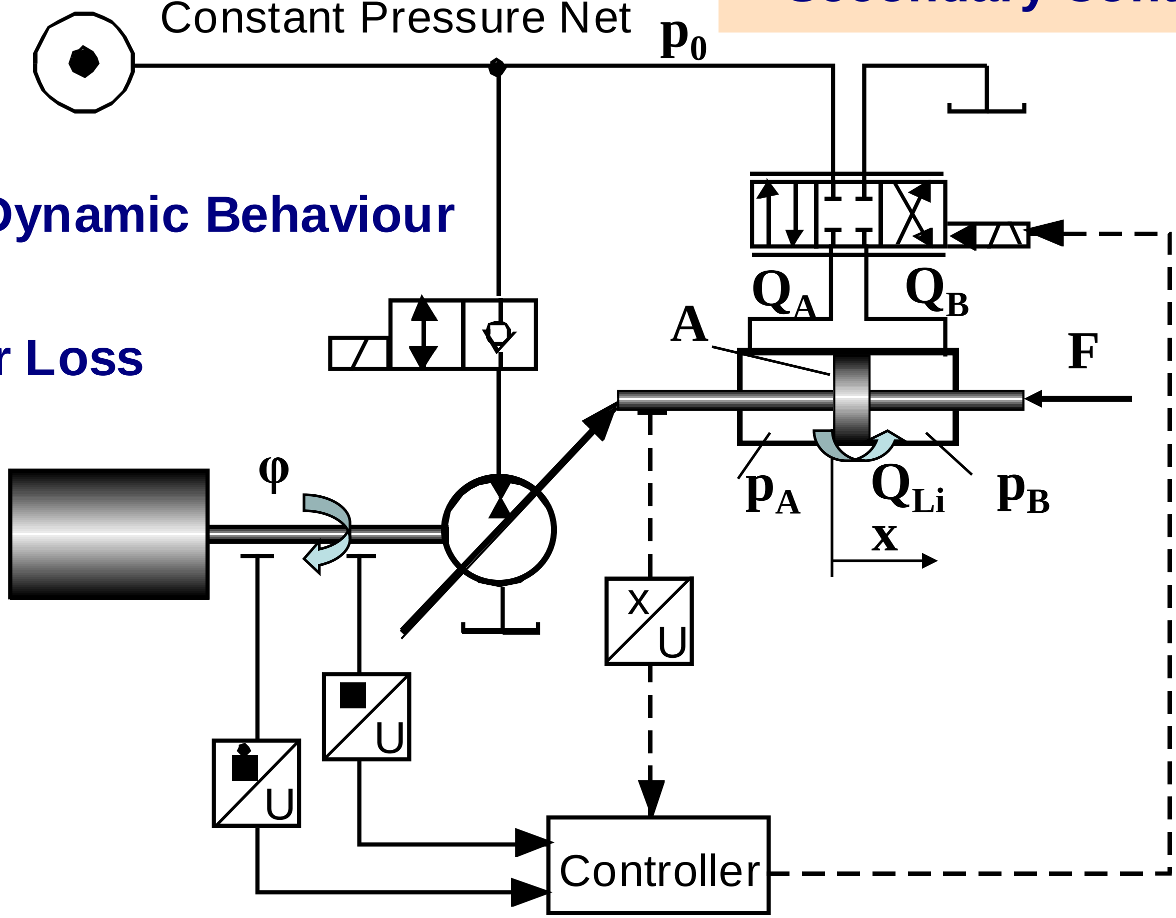
Displacement Controlled Rotary Drive

Secondary Controlled Drive

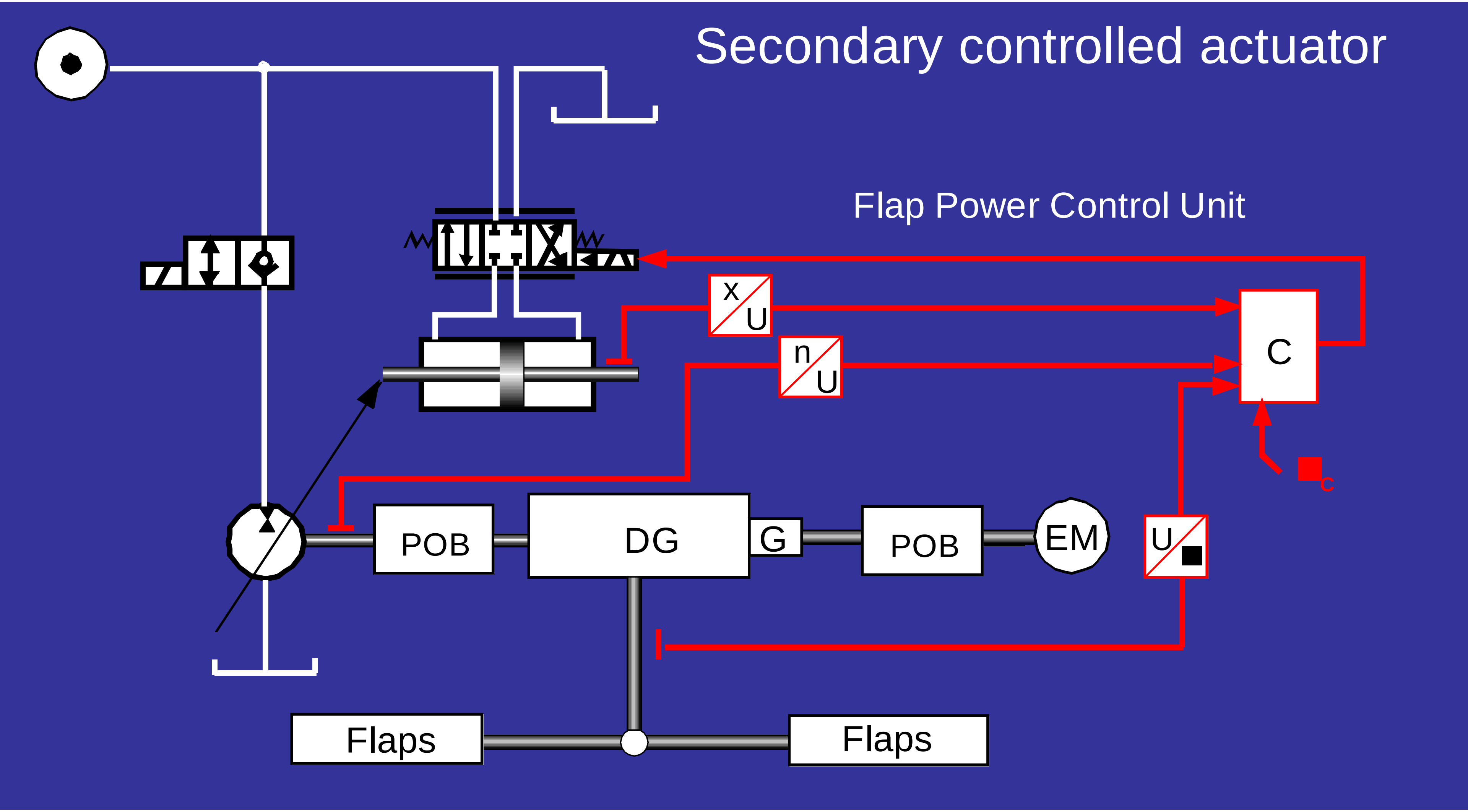
Excellent Dynamic Behaviour

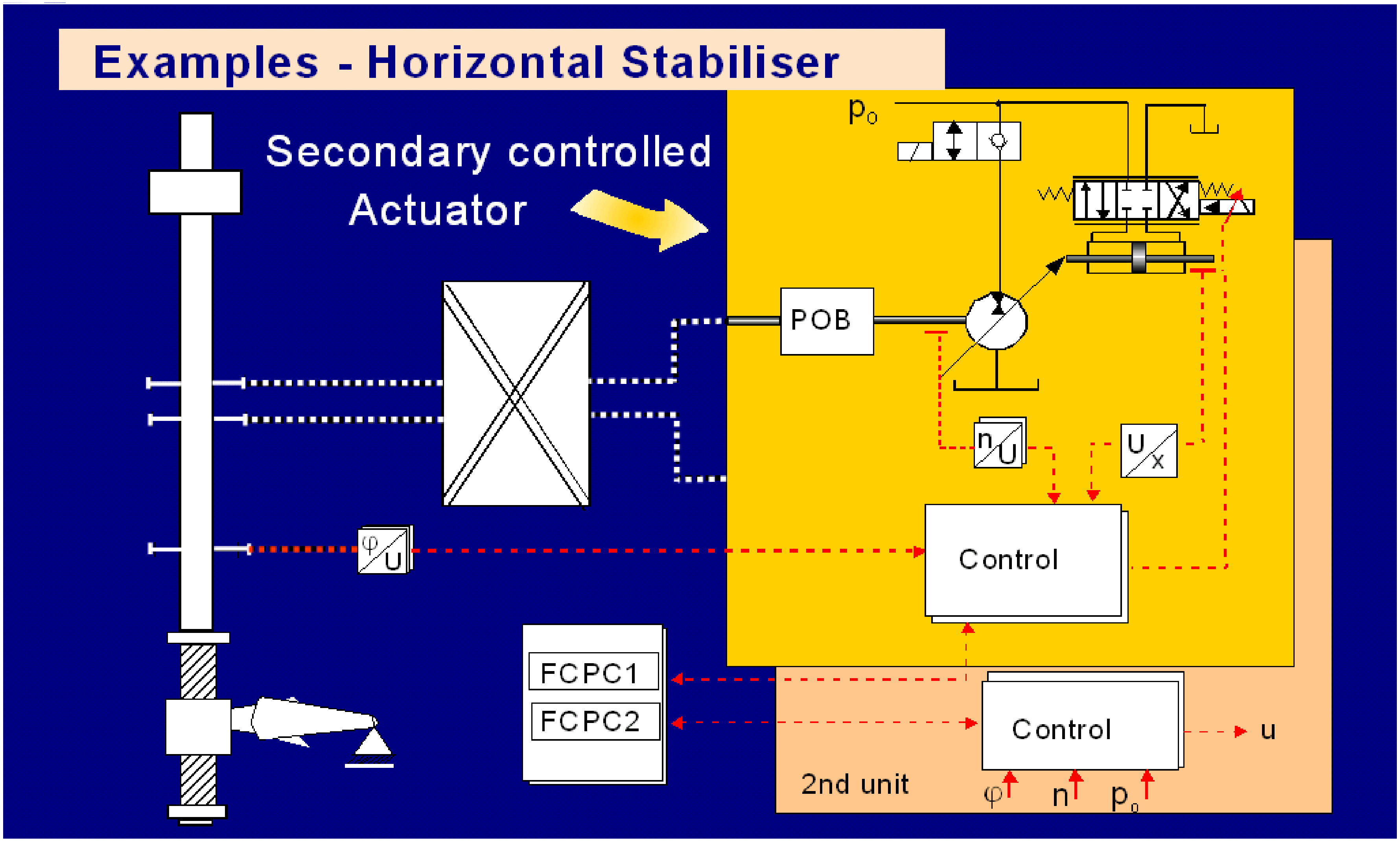
Low Power Loss

$$F = \frac{M_{Sx}}{r_c}$$

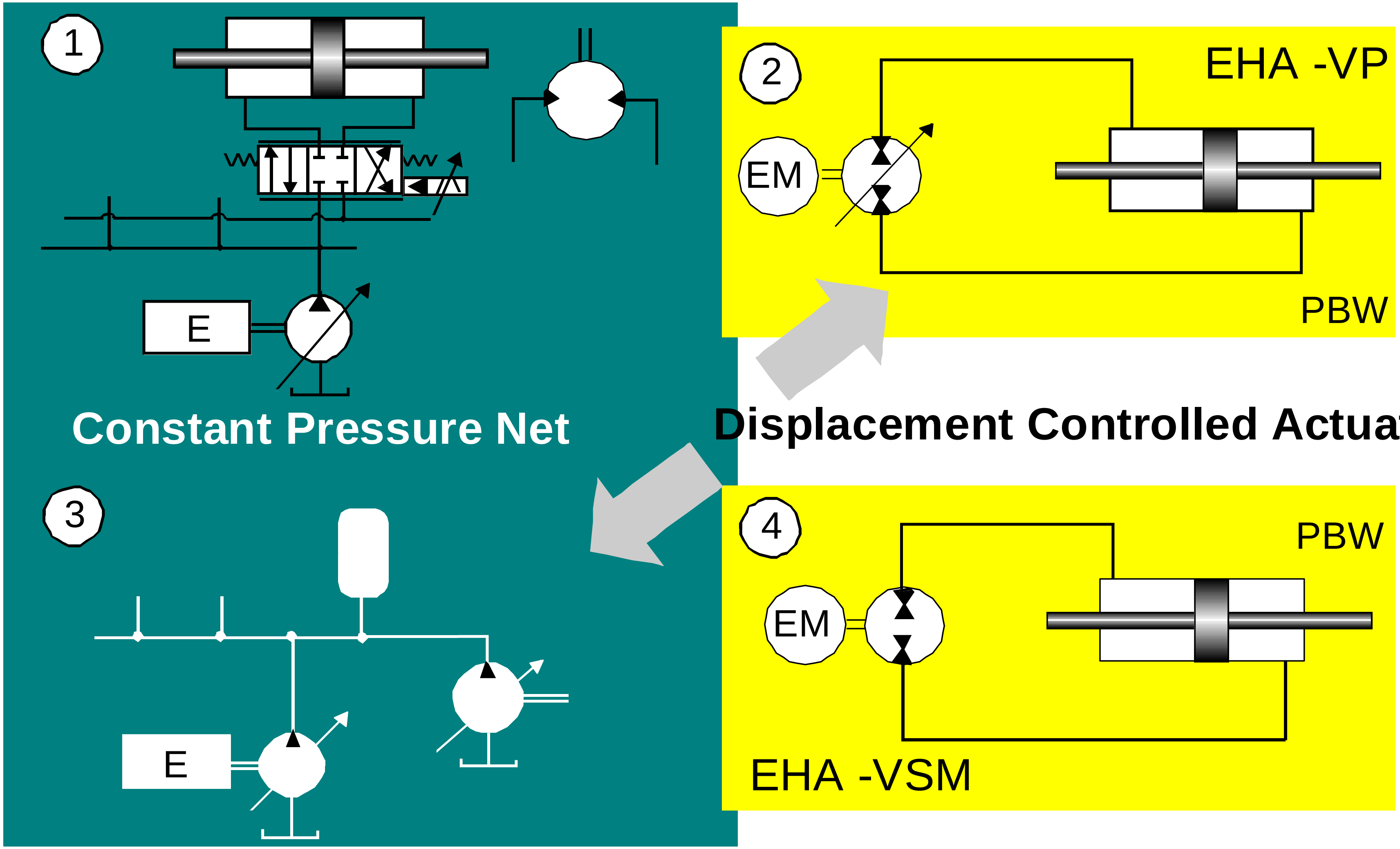


Examples - High Lift System

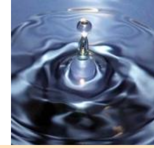



Examples - Horizontal Stabiliser
Secondary controlled
Actuator
p0
POB
n/U
U/x
Control
φ/U
FCPC1
FCPC2
Control
u
2nd unit
φ
n
p0

Hydraulic actuator principles

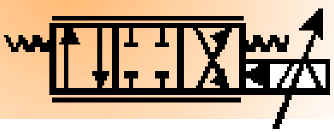


Hydraulic actuator principles



Classification & Main System Properties

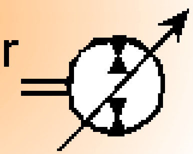
1 Valve Controlled Actuator



- Proportional valve
- Servovalve
- LS- valve

- Energy Dissipation
- High Bandwith
- Central Pressure Supply
- ~~Energy Recover~~

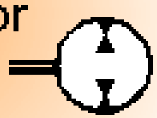
2 Displacement Controlled Actuator



- Pump Control
- Secondary Control

- ~~Central Pressure Supply~~
- Energy Recover
- High Bandwith
- Central Pressure Supply

3 Speed Controlled Actuator



- Electric Motor Control

- ~~Central Pressure Supply~~
- Energy Recover