WIEFP 2022

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Sizing and Design of Hydraulic Positioning Systems

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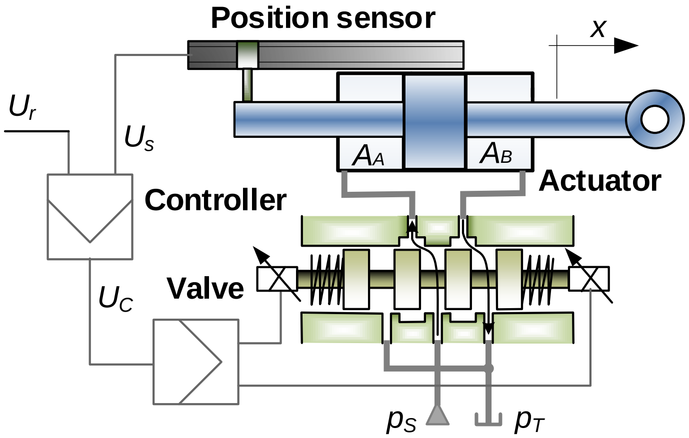
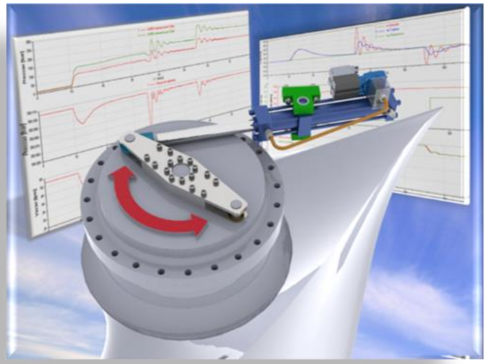
Agenda

- 1. Electro-Hydraulic Positioning Systems
- 2. Design Methodology of Technical Systems
- 3. Design Method of Hydraulic Positioning Systems
  - 3.1. Step 1: Static and Dynamic Sizing
  - 3.2. Step 2: Catalog Data Conversion
  - 3.3. Step 3: Dynamic Modelling and Simulation
- 4. Using the Design Method
- 5. Particularities:
  - 5.1. Cylinder and Valve Matching
  - 5.2. Influence of the Valve Flow Coefficient
  - 5.3. Correlation between Natural Frequencies

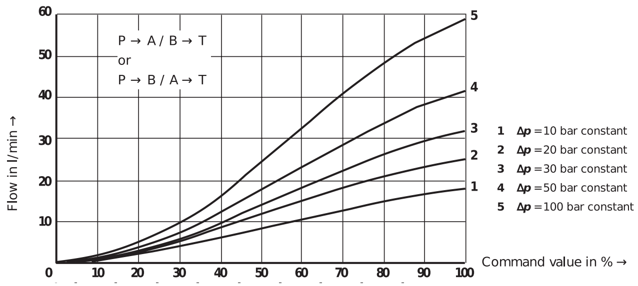
- Electro-hydraulic positioning systems (EHPS):
  - For driving and controlling high loads with reliability, speed, and accuracy
- Main components:
  - Electrically modulated hydraulic control valve
    - Servovalves, Proportional valves
  - Cylinder
  - Controller





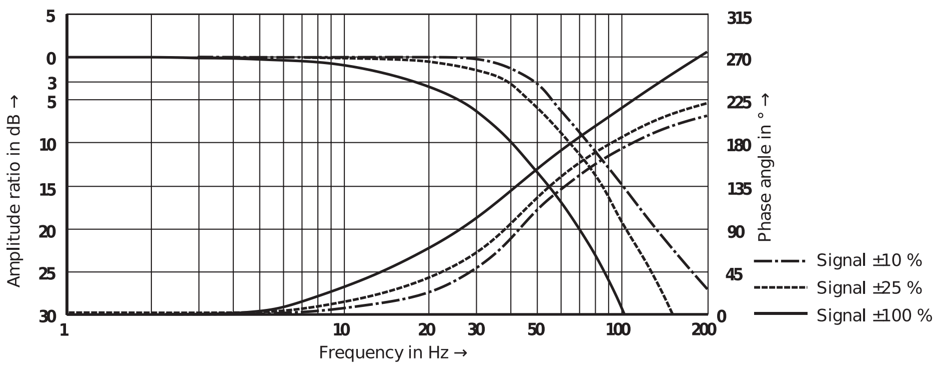
Control valves:

- Pressure dependence of the valve flow rate
- Dead zone

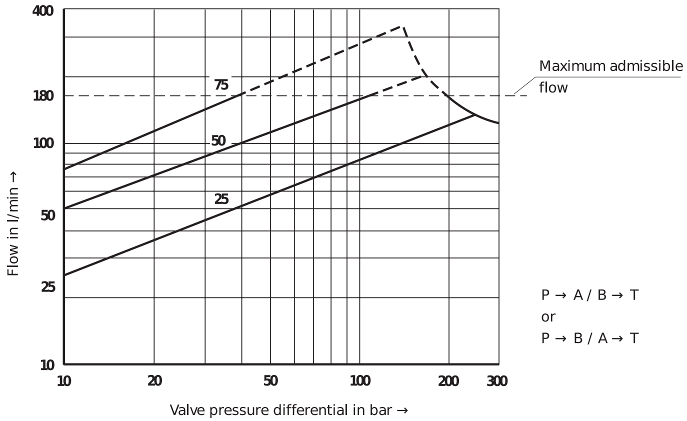


- Variable dynamic behavior

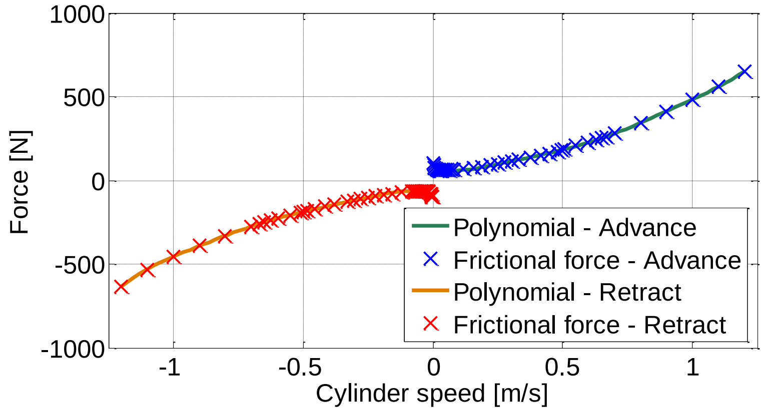
Frequency response (with V control spool)

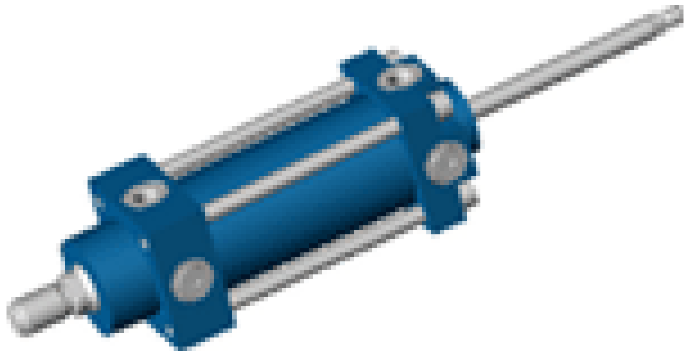


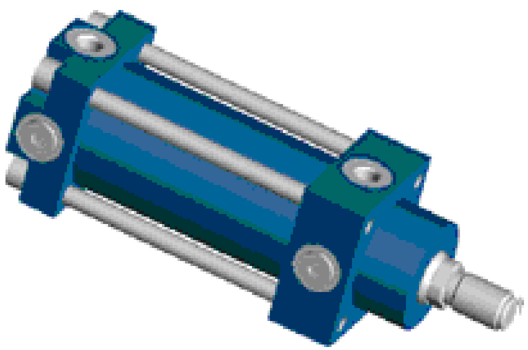
- Operating limits



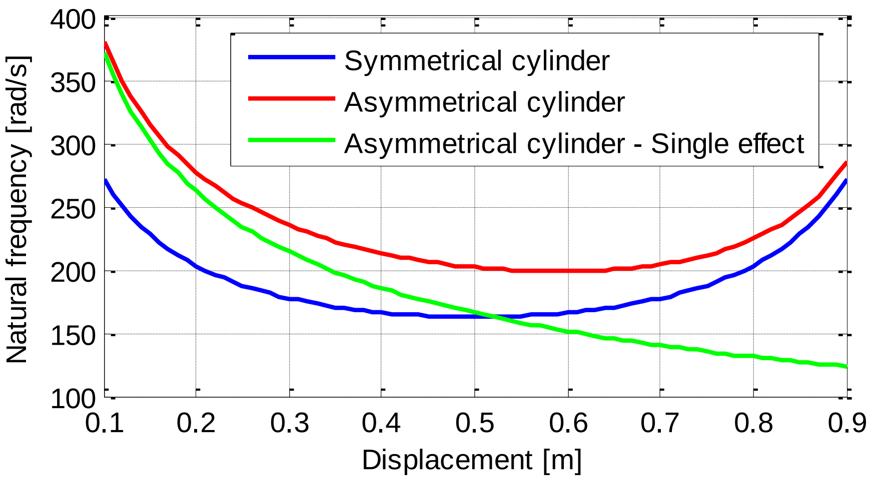
- Cylinder + load
  - Non-linear friction



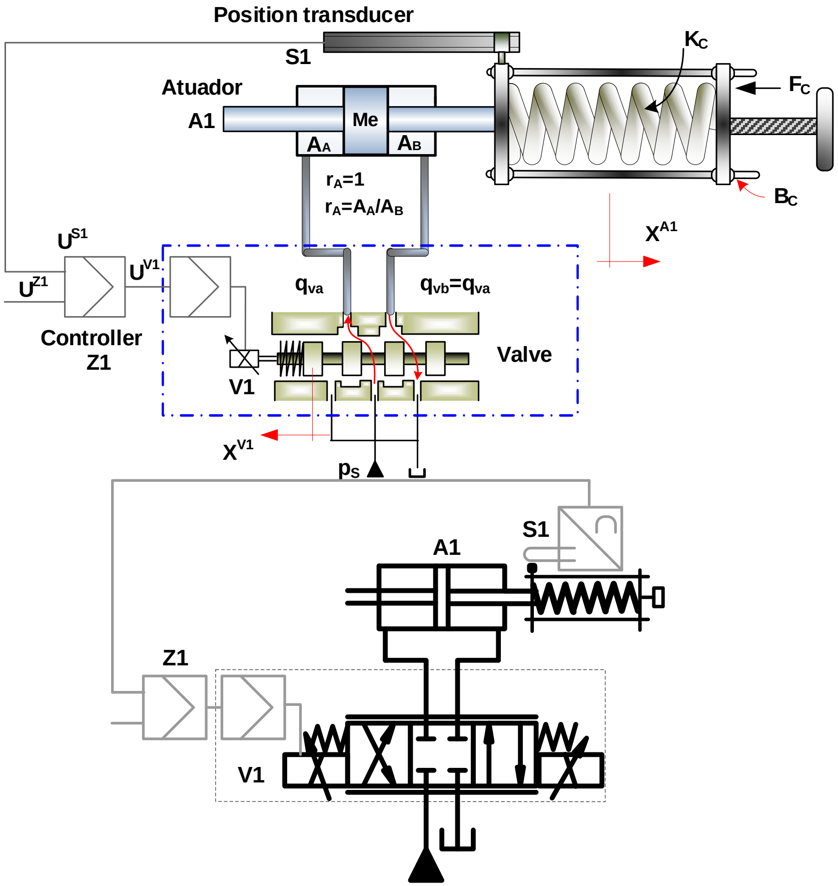




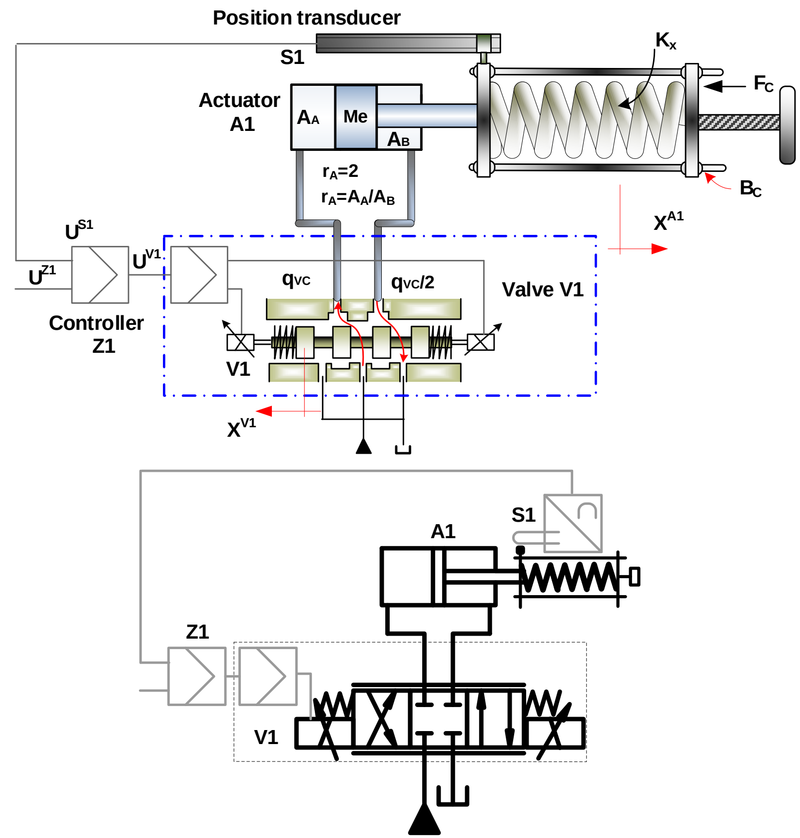
- Dynamic behavior function of the cylinder position



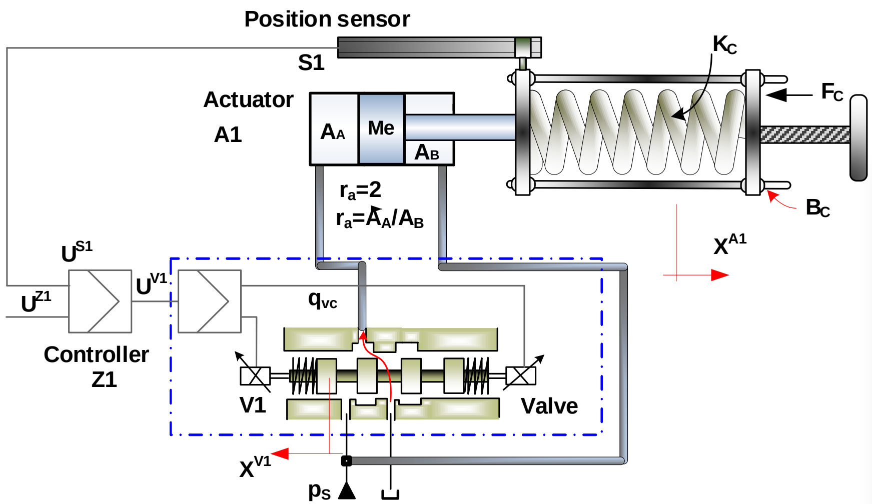
- System configurations:
  - Symmetrical cylinder + symmetrical valve

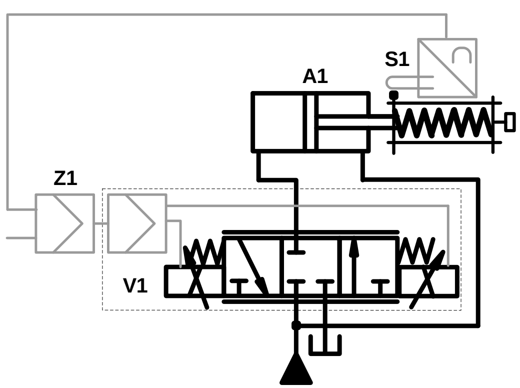


- Asymmetrical cylinder + asymmetrical valve
  - It is also usual: Asymmetrical cylinder + symmetrical valve



- System configurations:
    - Asymmetrical cylinder (single effect) + 3 port valve.

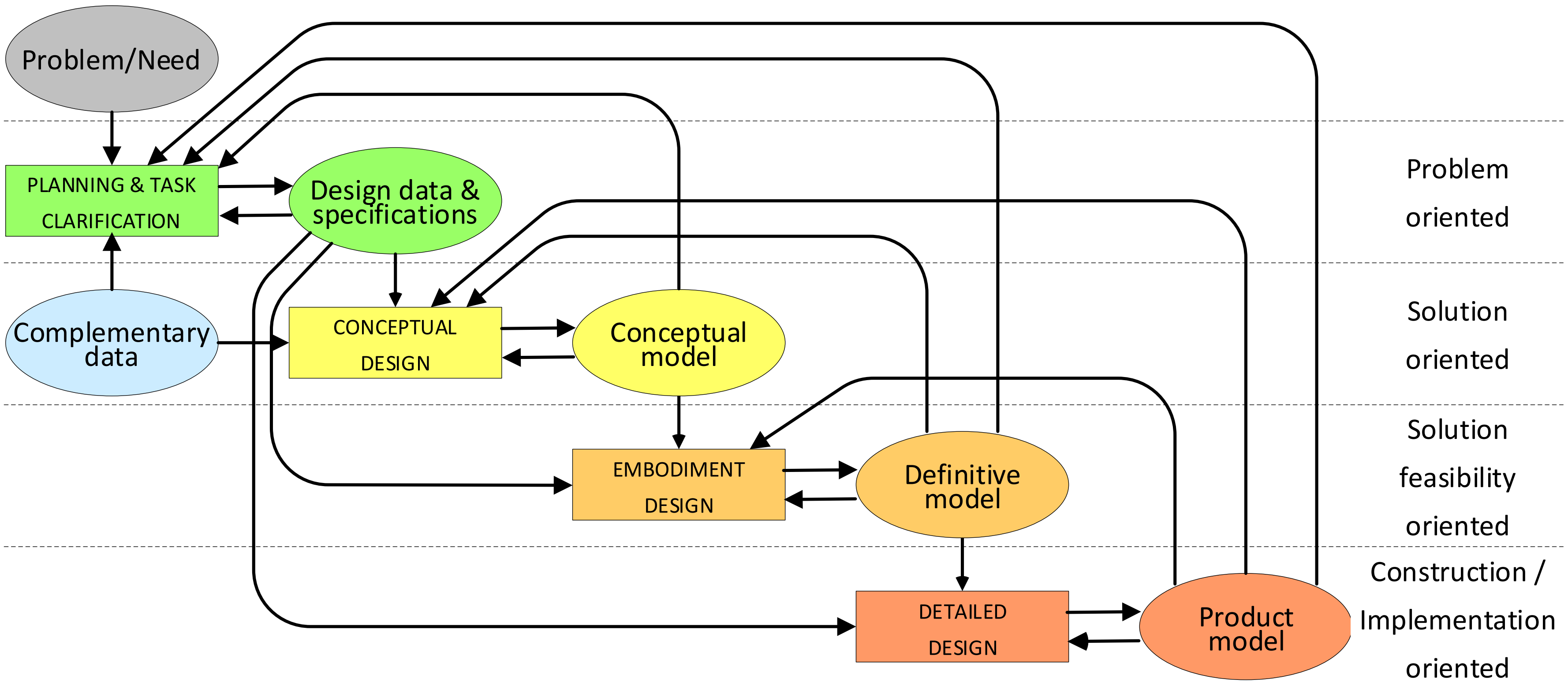




- Poorly designed system:
  - Oscillations, noise, and inadequate static and/or dynamic responses
  - Cause a negative understanding about the EHPS applicability

- Well-designed system:
  - Consistent, reliable, durable, smooth, and noiseless system

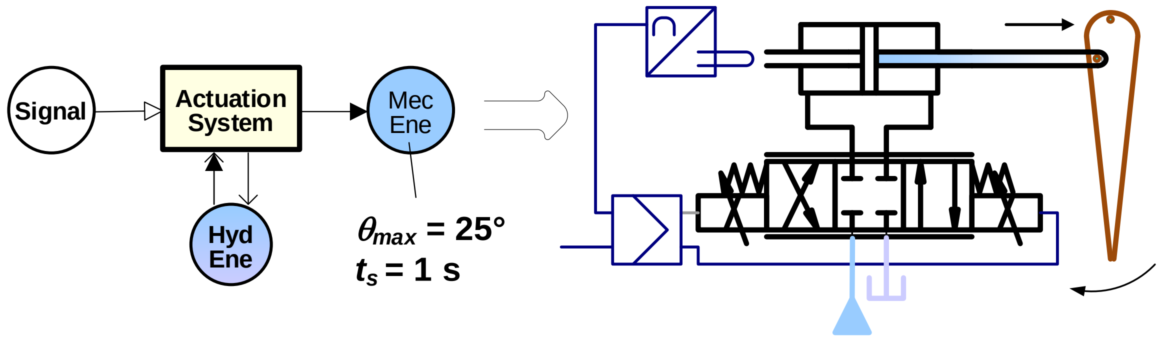
- Design of a Electro-Hydraulic Positioning System Controlled by Valve
  - Definition of the system configuration:
    - Symmetrical cylinder + symmetrical valve
    - Asymmetrical cylinder + asymmetrical valve
    - Asymmetrical single effect cylinder + 3 port valve
  - Component sizing
    - Valve, cylinder
  - Control strategy and controller synthesis


Problem/Need
PLANNING & TASK CLARIFICATION
Design data & specifications
Complementary data
CONCEPTUAL DESIGN
Conceptual model
EMBODIMENT DESIGN
Definitive model
DETAILED DESIGN
Product model
Problem oriented
Solution oriented
Solution feasibility oriented
Construction / Implementation oriented

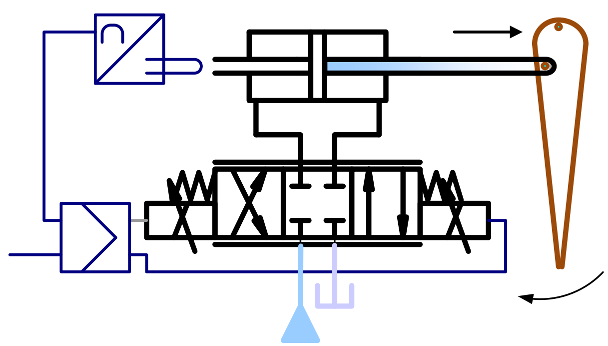
PLANNING & TASK CLARIFICATION

FUNCION = Rudder positioning
Maximum angle = 25°
Settling time = 1 s

CONCEPTUAL DESIGN



EMBODIMENT DESIGN



Controller gains: K_P = 2; K_I = 0,1

Position sensor: K_s = 300 V/m

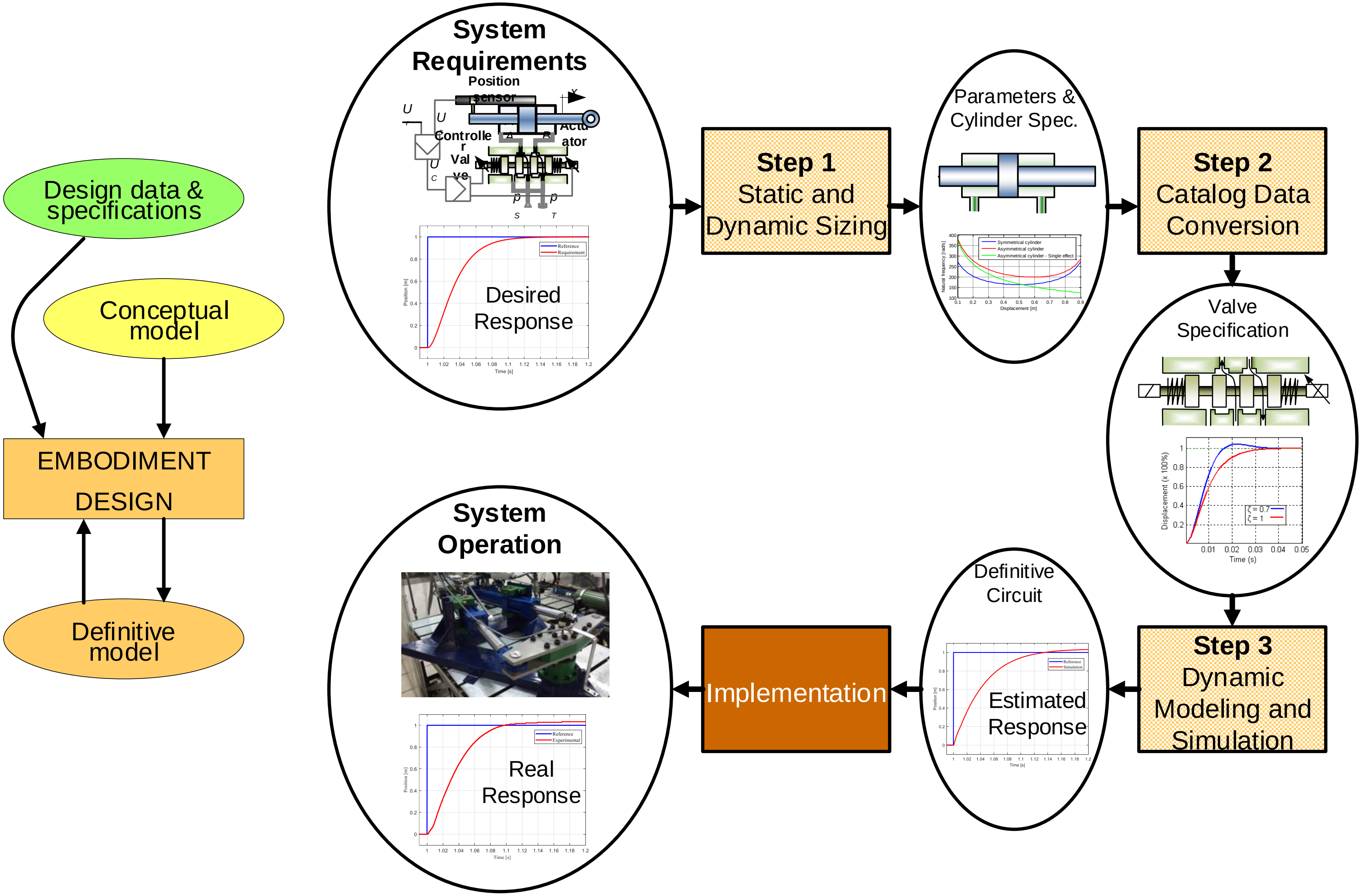
Cylinder: d = 50 mm

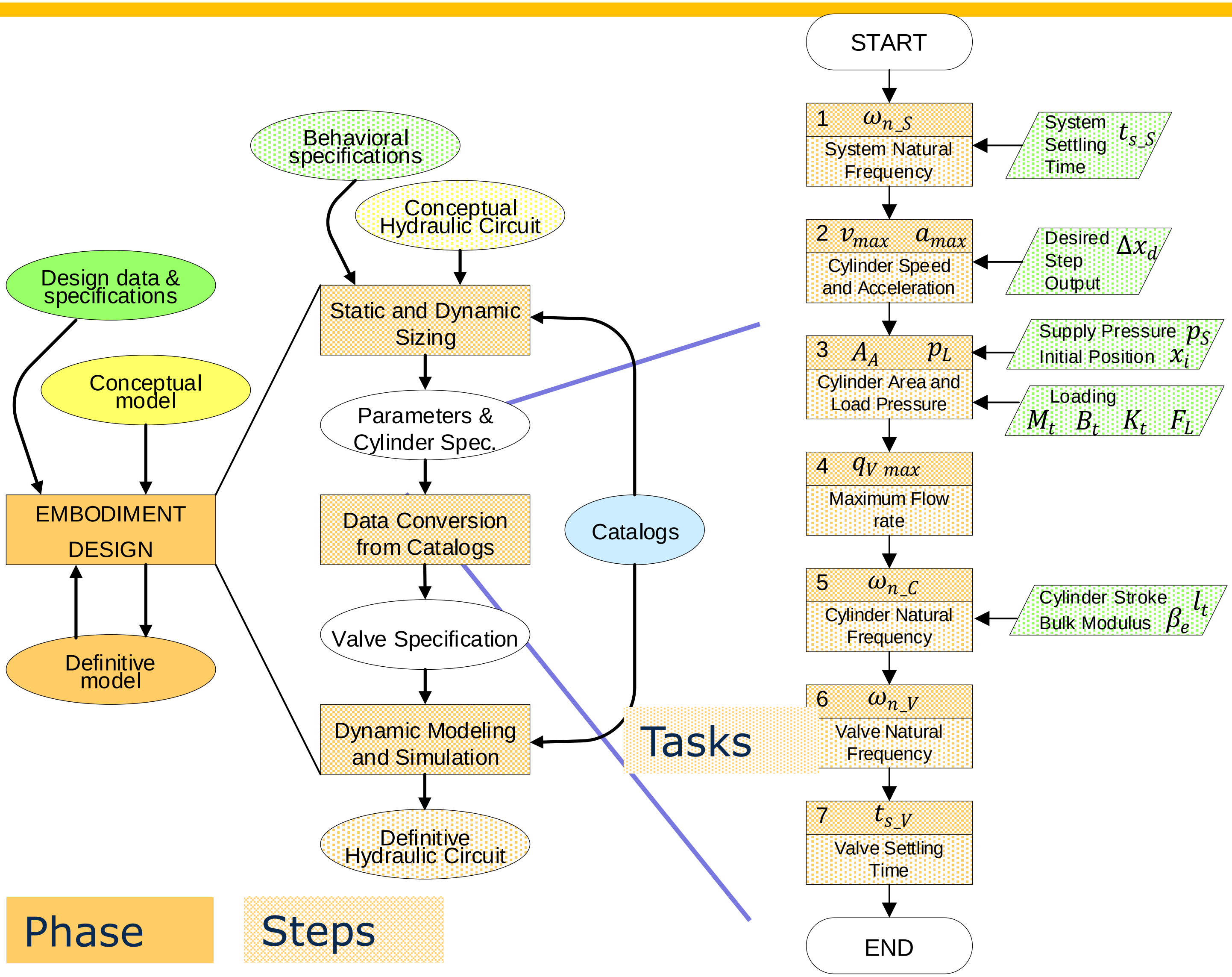
Valve: q_{Vn} = 10 L/min
(@ Δp = 10 bar)

DETAILED DESIGN

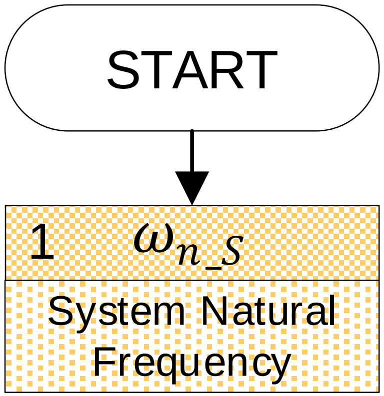
SENSOR = Type HDY0905
CONTROLLER = Type DSP1312
Cylinder = Type DXY09
Valve = Type 4QE600

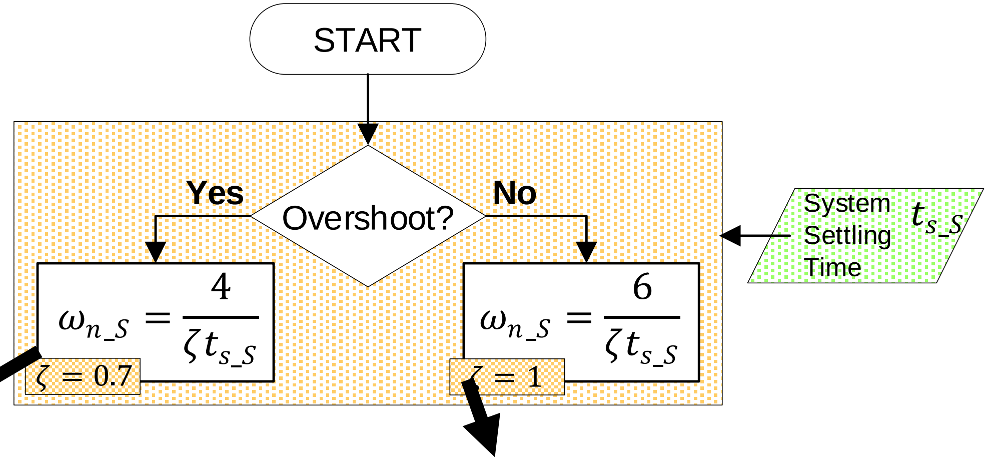
Design Method of Hydraulic Positioning Systems





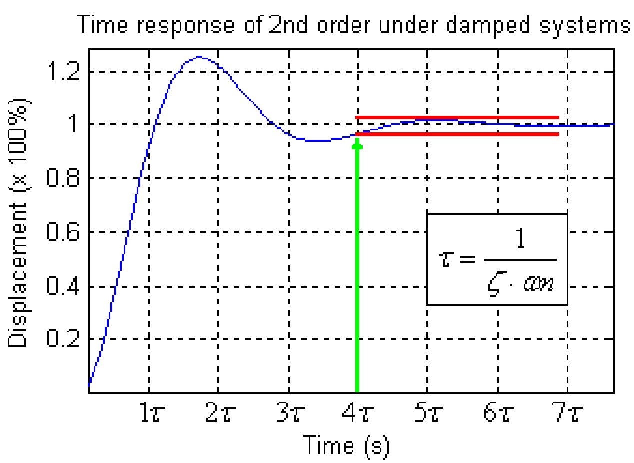
Behavioral specifications
Conceptual Hydraulic Circuit
Design data & specifications
Conceptual model
EMBODIMENT DESIGN
Definitive model
Static and Dynamic Sizing
Parameters & Cylinder Spec.
Data Conversion from Catalogs
Valve Specification
Dynamic Modeling and Simulation
Definitive Hydraulic Circuit
Catalogs
Tasks
Phase
Steps
START
1 ω_{n_S} System Natural Frequency
System Settling Time t_{s_S}
2 v_{max} a_{max} Cylinder Speed and Acceleration
Desired Step Output Δx_d
3 A_A p_L Cylinder Area and Load Pressure
Supply Pressure p_S
Initial Position x_i
Loading M_t B_t K_t F_L
4 $q_{V\ max}$ Maximum Flow rate
5 ω_{n_C} Cylinder Natural Frequency
Cylinder Stroke l_t
Bulk Modulus β_e
6 ω_{n_V} Valve Natural Frequency
7 t_{s_V} Valve Settling Time
END

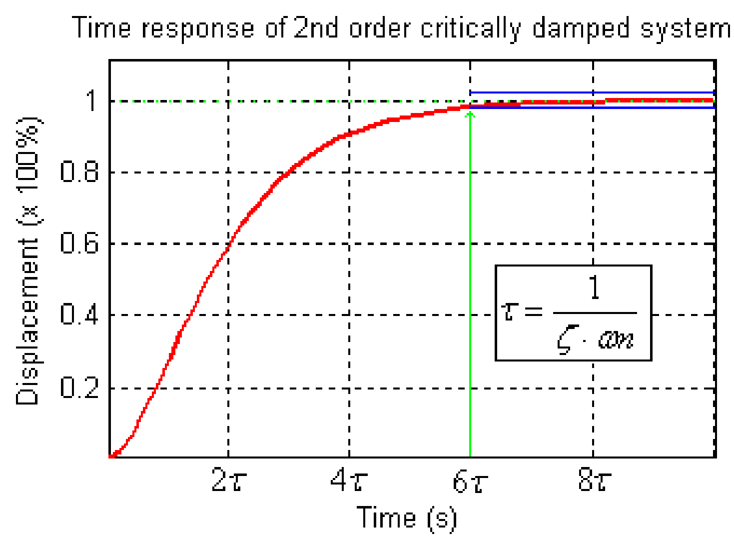




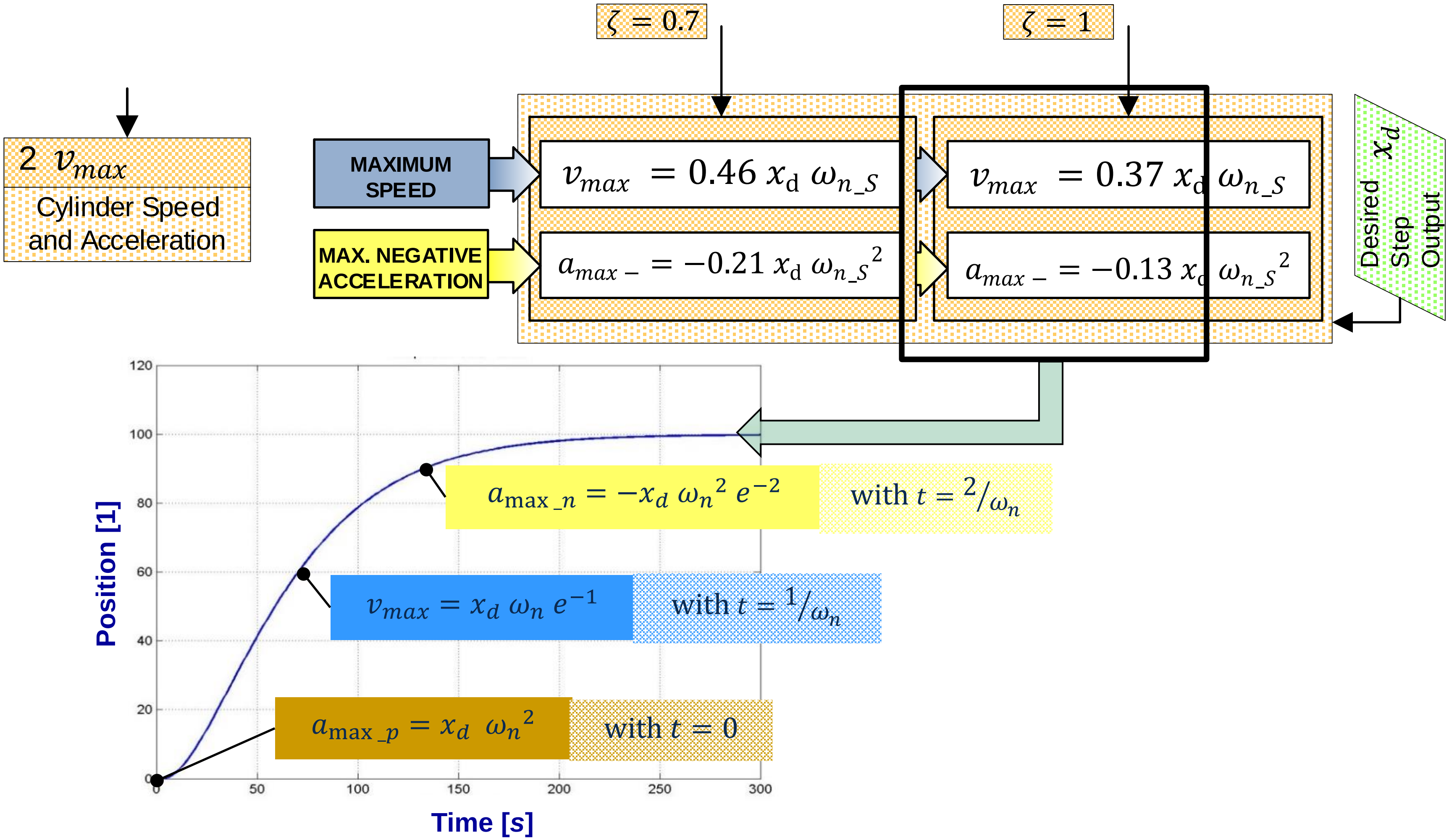
2nd order underdamped system response

2nd order critically damped system response

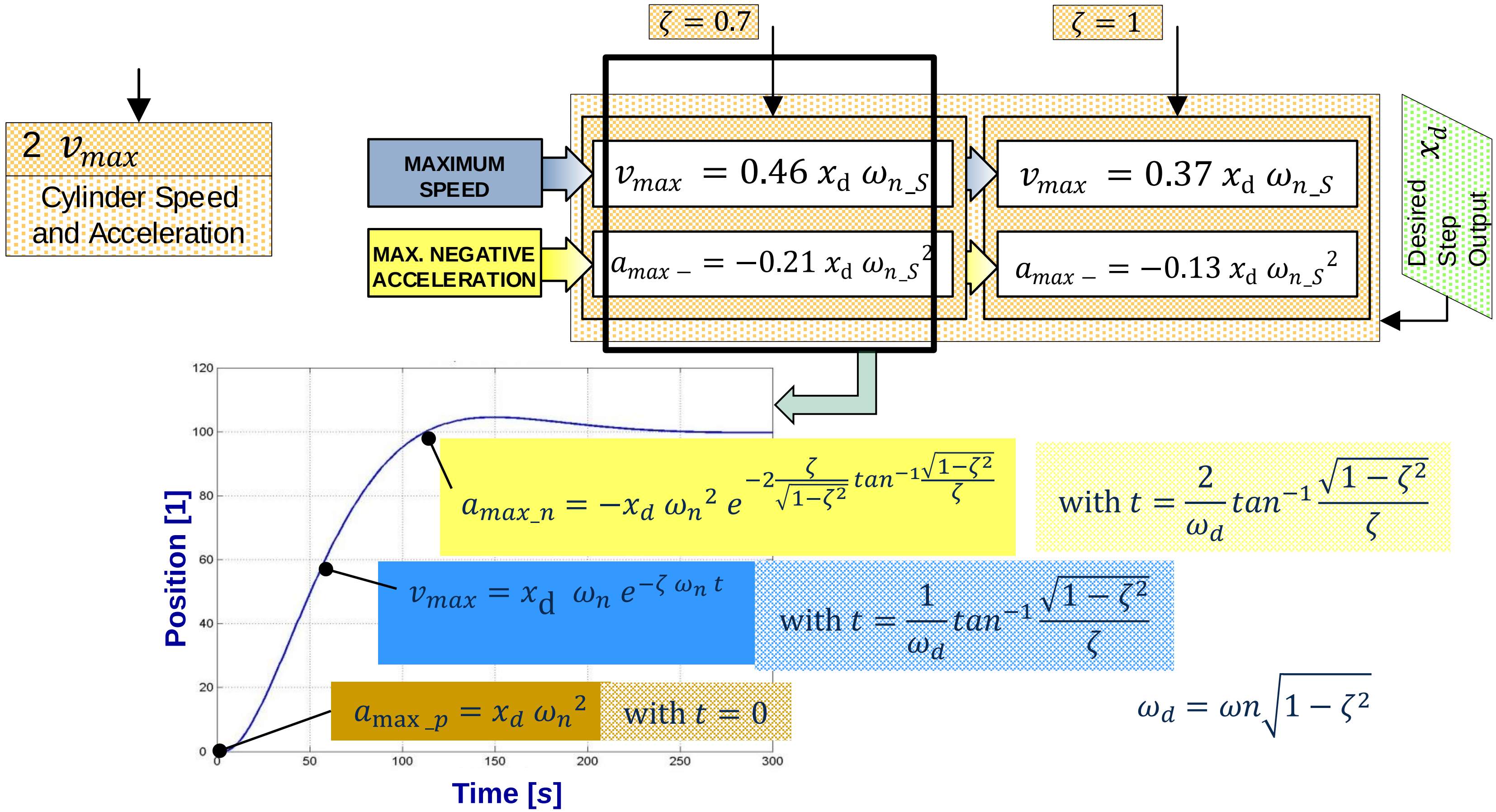




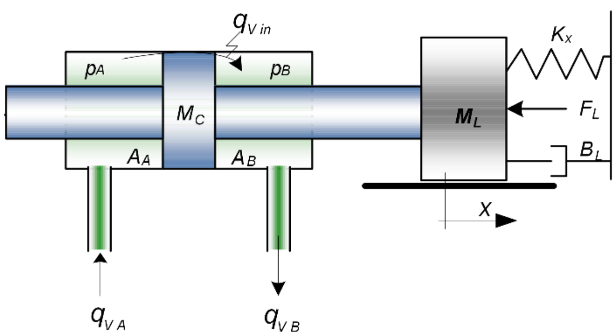
Static and Dynamic Sizing

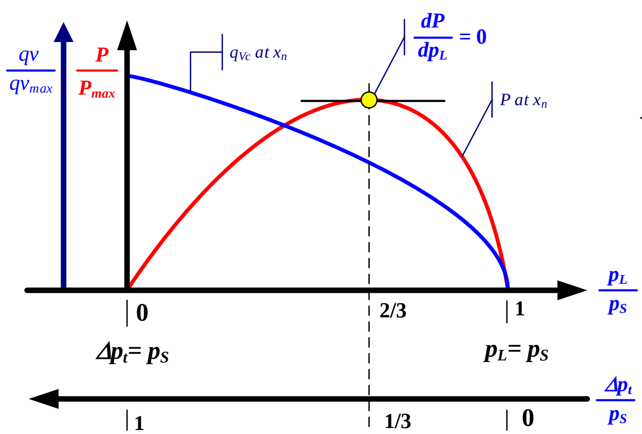


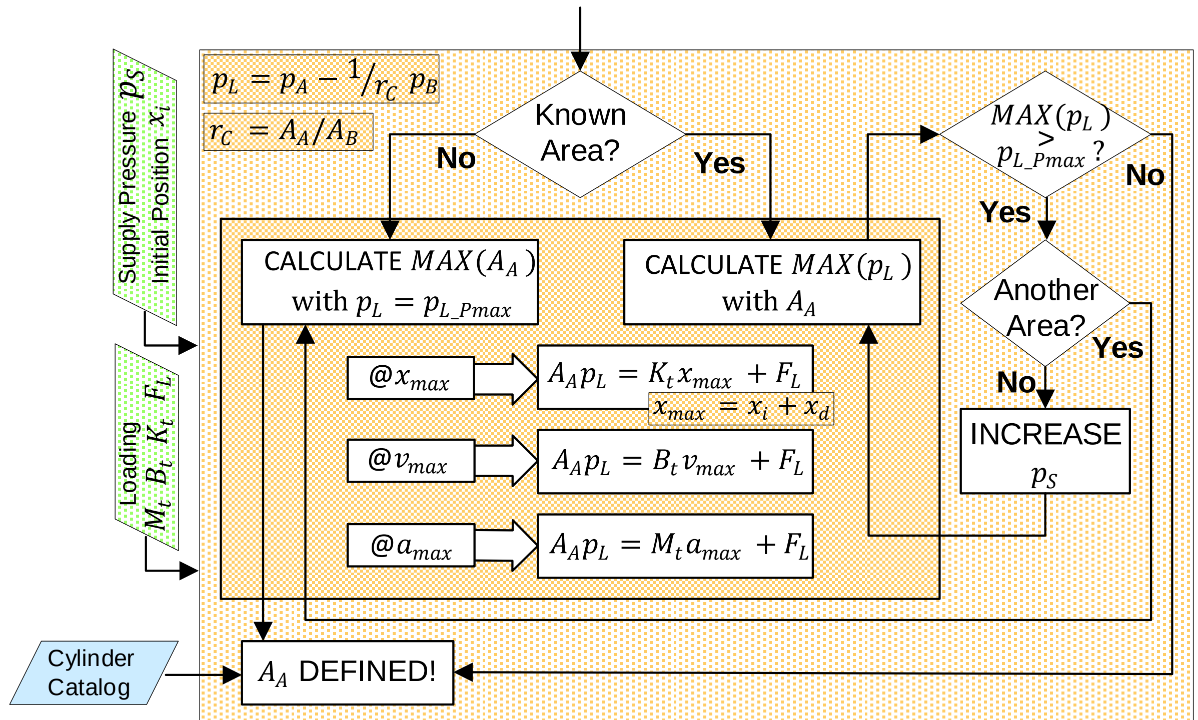
Static and Dynamic Sizing



3 A_A Cylinder Area and Load Pressure







LOAD PRESSURE @ MAXIMUM POWER

4 Ports Sym ⇒ $p_{L\_Pmax} = 2/3P_s$

3 Ports Asym ⇒ $p_{L\_Pmax} = 1/3P_s$

4 Ports Asym ⇒ $p_{L\_Pmax} = 4/3P_s$ (Extend)

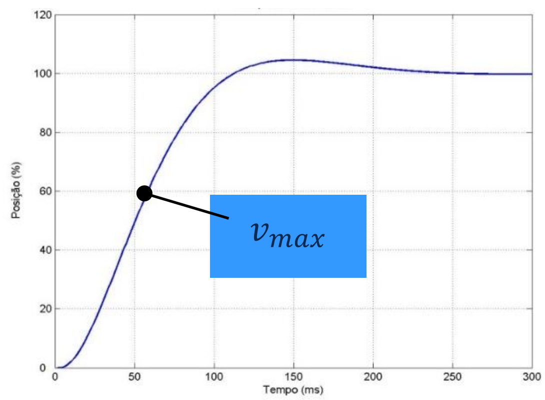
4 Ports Asym ⇒ $p_{L\_Pmax} = 1/3P_s$ (Retract)

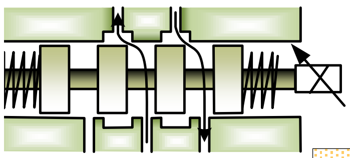
4 $q_{V\,max}$

Maximum Flow rate

5 ω_{n_C}

Cylinder Natural Frequency

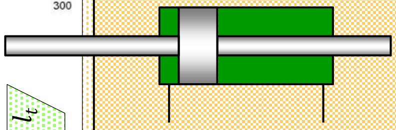




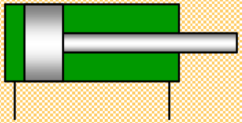
$$q_{V\,max} = A_A\, v_{max}$$

Cylinder Stroke l_t

Bulk Modulus β_e

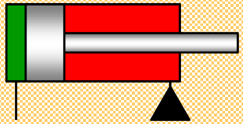


$$\omega_{n\_min} = \left(\frac{4\,\beta e\, A_A{}^2}{M_t\, V_t}\right)^{1/2}$$

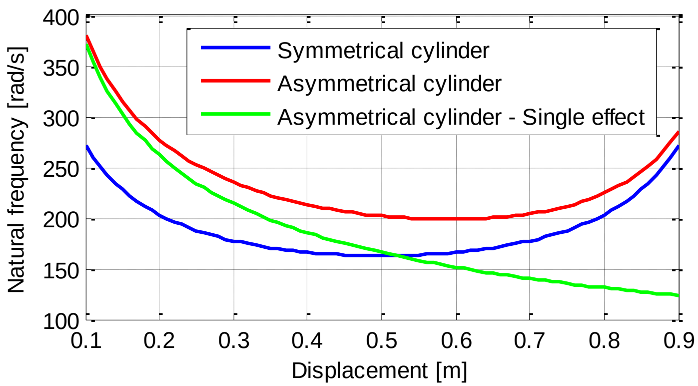


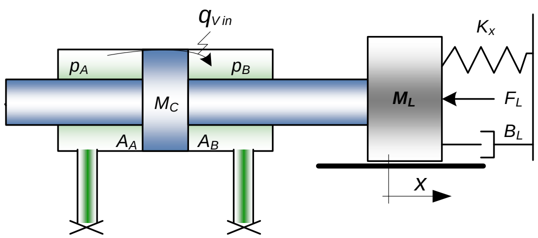
$$\omega_{n\_min} = \left[\frac{\beta e}{M_t}\left(\frac{A_A}{l_A} + \frac{A_B}{l_t - l_A}\right)\right]^{1/2}$$

$$\text{with } l_A = \frac{A_A\, l_t}{A_A - A_B}\left[1 - \left(\frac{A_B}{A_A}\right)^{1/2}\right]$$



$$\omega_{n\_min} = \left(\frac{\beta_e\, A_A{}^2}{M_t\, V_A}\right)^{1/2}$$





6 ω_{n_V} Valve Natural Frequency

7 t_{s_V} Valve Settling Time

END

$\omega_{n\_A} \geq 5\,\omega_{n\_S}$?

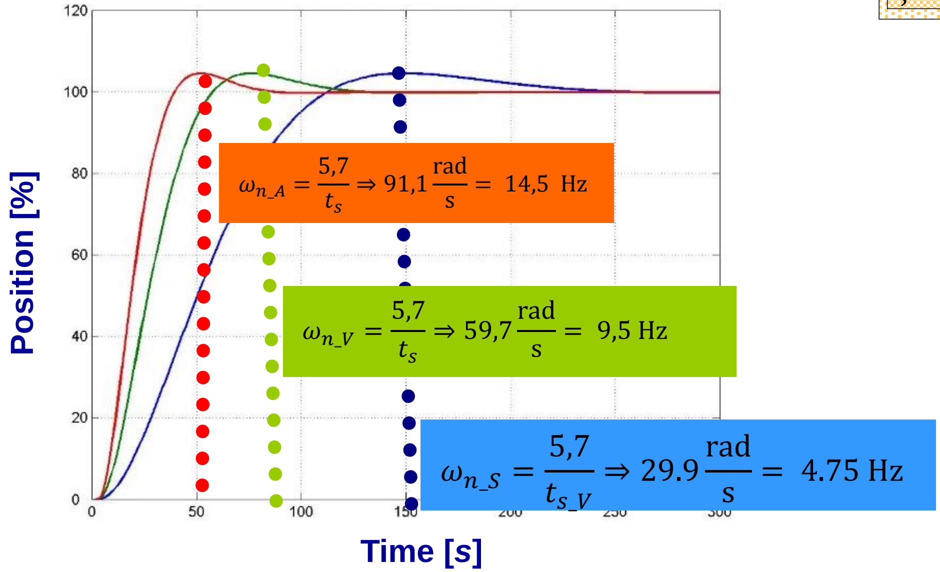
Yes: $\omega_{n\_V} \geq 5\,\omega_{n\_S}$

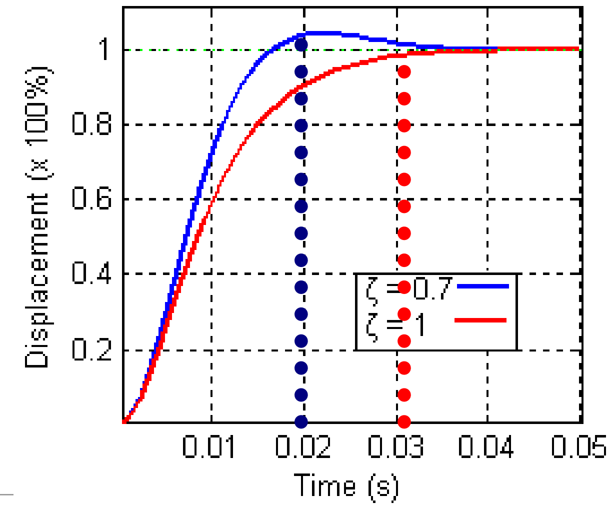
No: $\omega_{n\_V} \geq 5\,\omega_{n\_A}$

$$t_{s\_V} = \frac{4}{\zeta\omega_{n\_V}} = \frac{5.7}{\omega_{n\_V}}$$

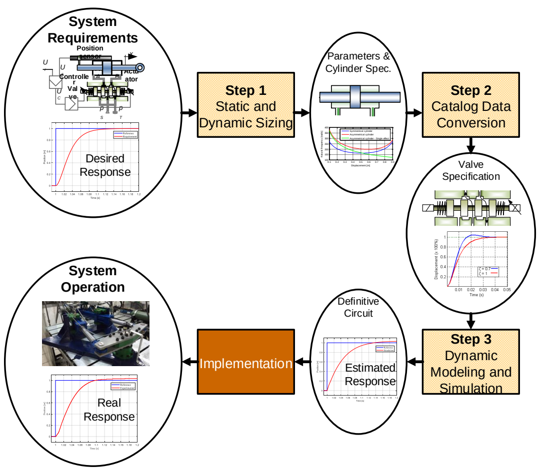
$\zeta = 0.7$

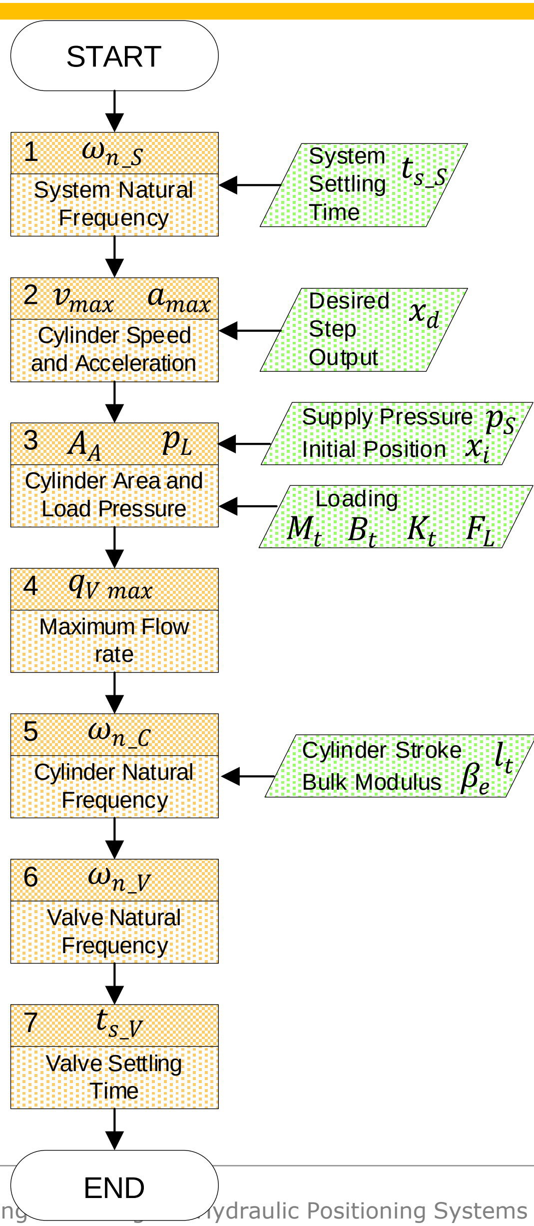
END

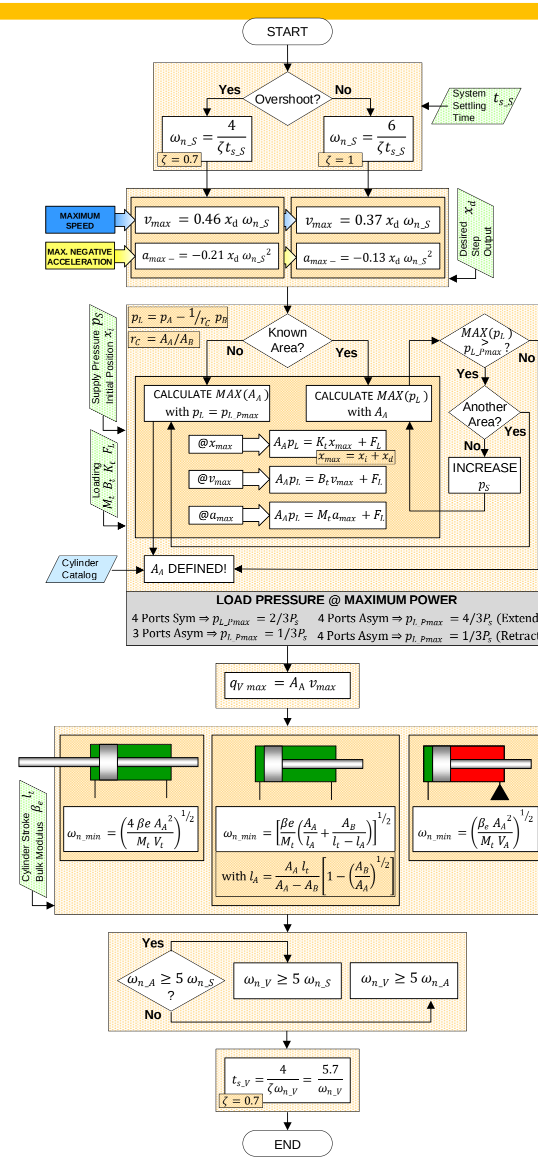




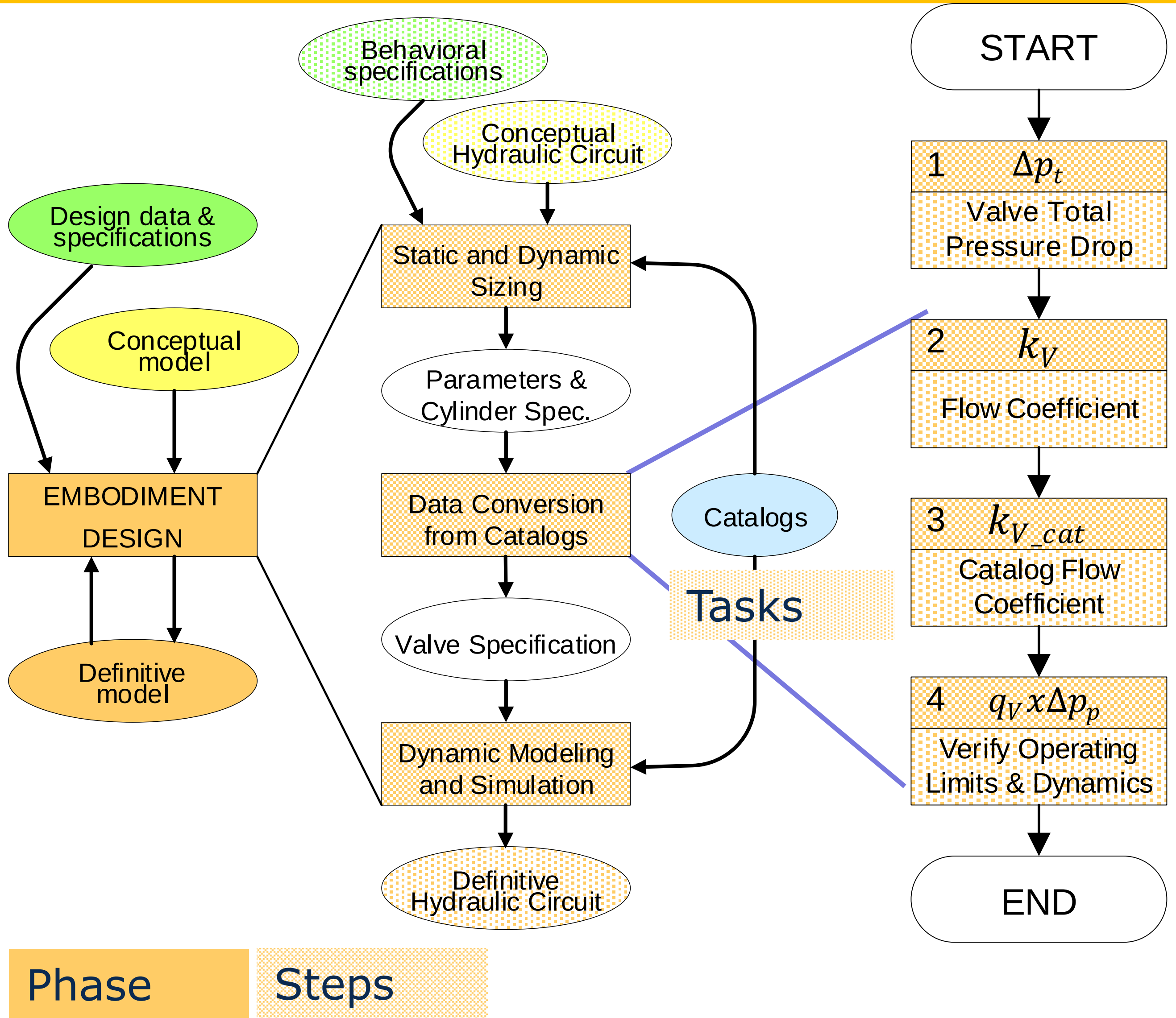
Static and Dynamic Sizing







Catalog Data Conversion



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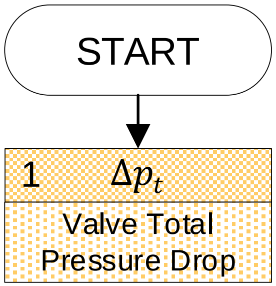
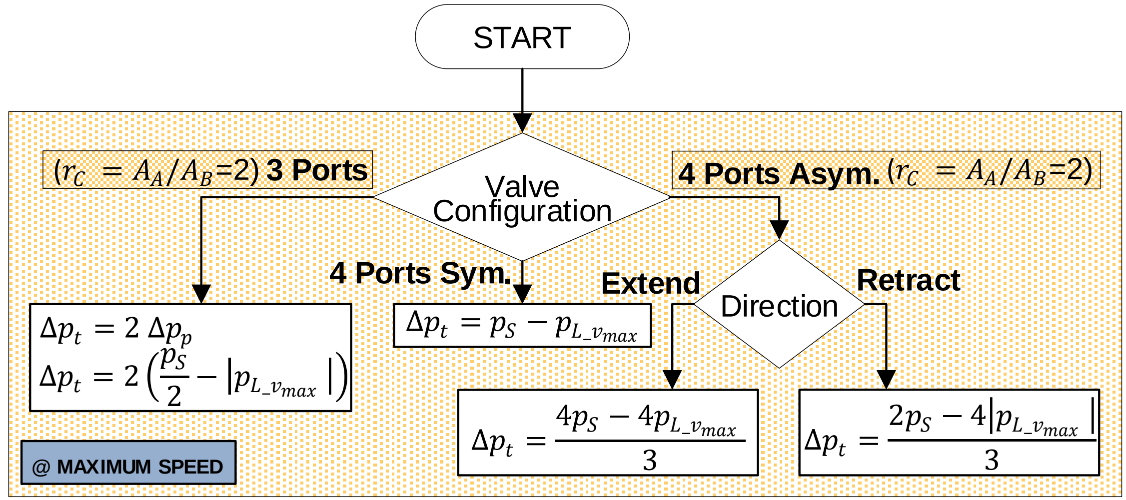
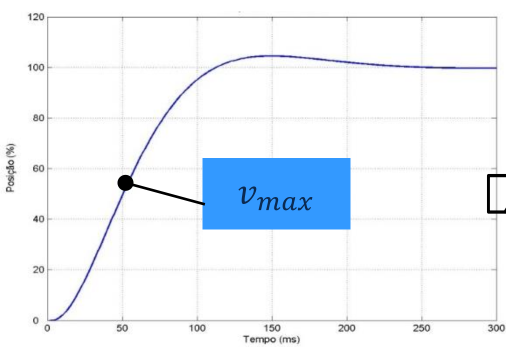
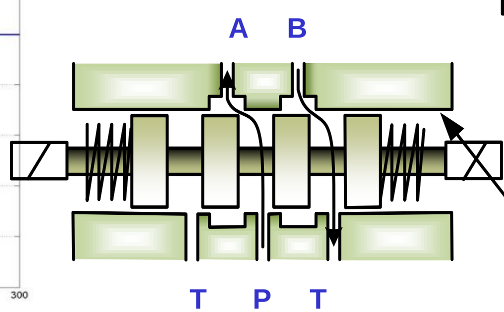




$$@v_{max} \Rightarrow A_A p_L = B_t v_{max} + F_L$$





TOTAL pressure drop: Δp_t

P => A + B => T or P => B + A=>T

PARTIAL pressure drop: Δp_p

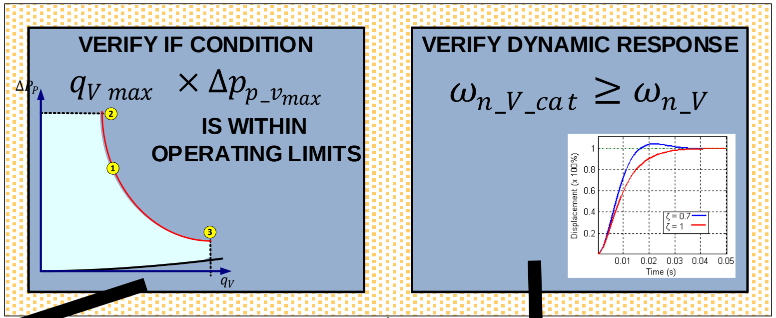
P => A or B => T   P => B or A => T

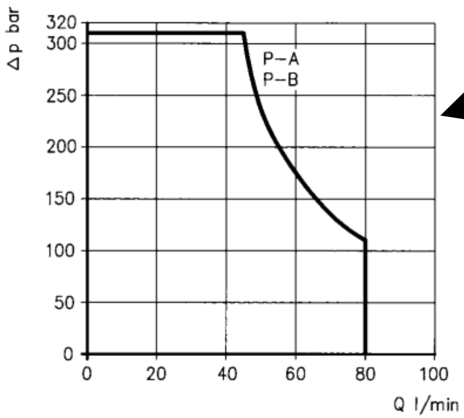
Catalog Data Conversion

4 $q_V x \Delta p_p$

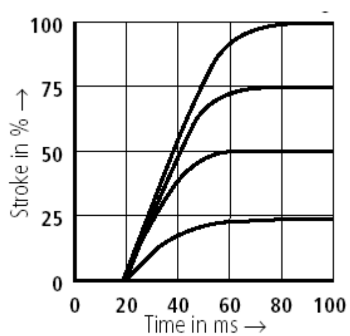
Verify Operating Limits & Dynamics

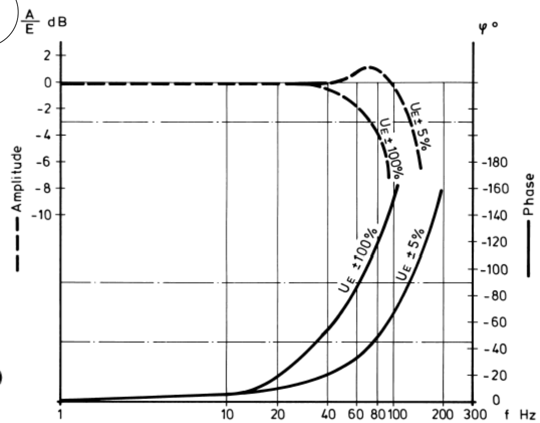
END



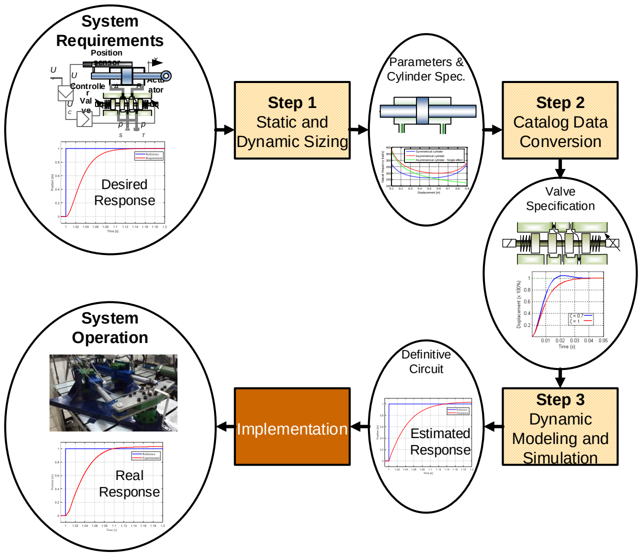


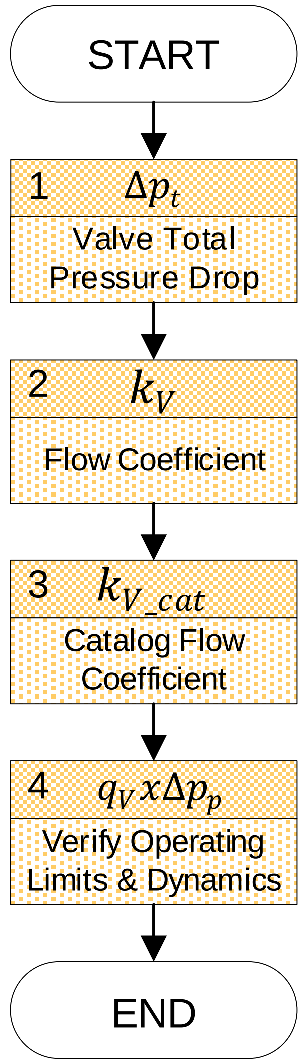
END

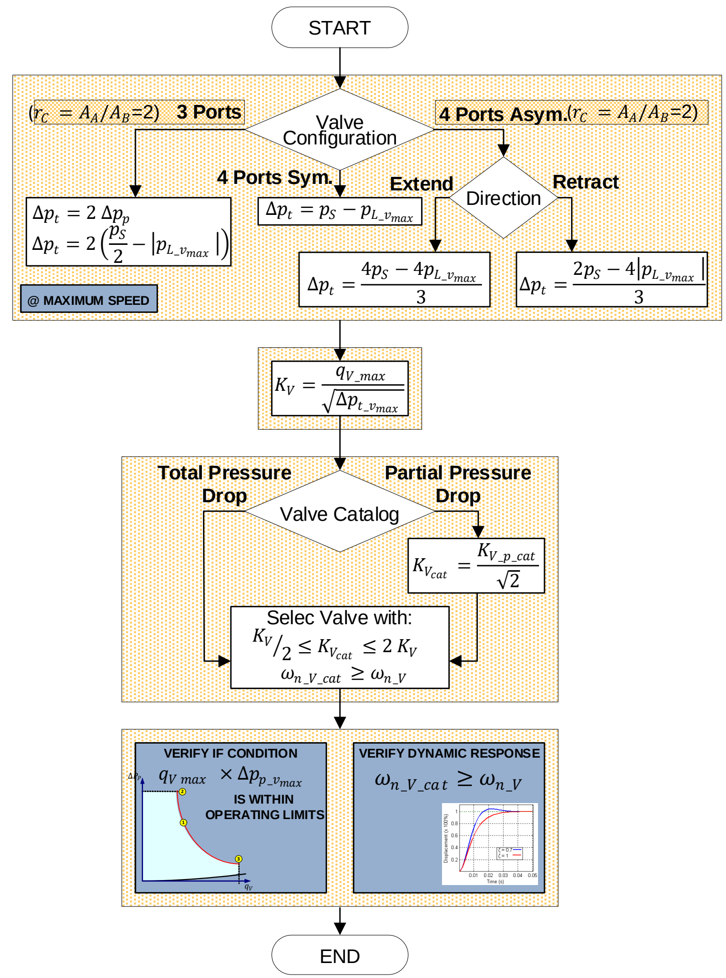




Catalog Data Conversion







Catalog Data Conversion

2 k_V Flow Coefficient

3 k_{V_cat} Catalog Flow Coefficient

$$K_V = \frac{q_{V\_max}}{\sqrt{\Delta p_{t\_vmax}}}$$

Valve Catalog

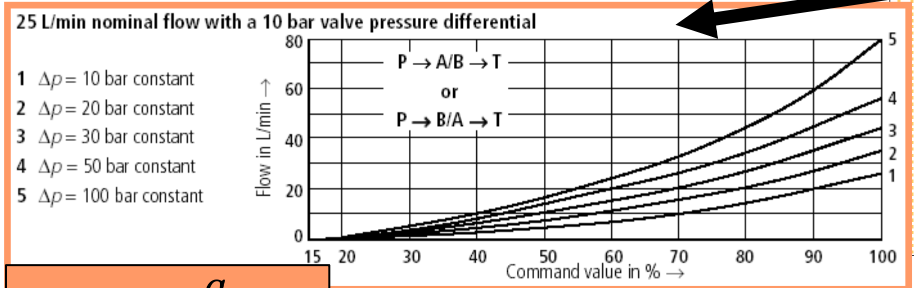
Total Pressure Drop | **Partial Pressure Drop**

$$K_{V_{cat}} = \frac{K_{V\_p\_cat}}{\sqrt{2}}$$

Selec Valve with:

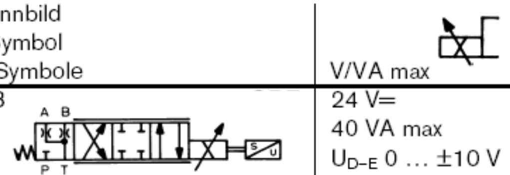
$$K_V/_2 \le K_{V_{cat}} \le 2\,K_V$$

$$\omega_{n\_V\_cat} \ge \omega_{n\_V}$$



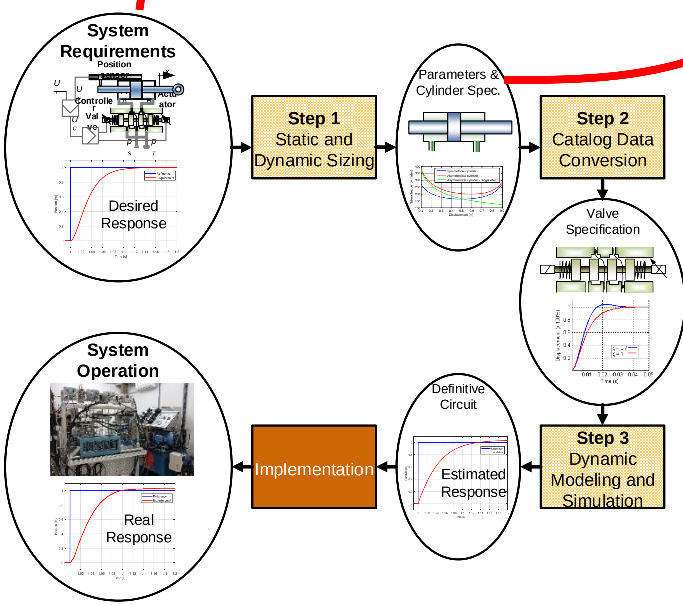
$$K_{V\_cat} = \frac{q_{V\_n}}{\sqrt{\Delta p_{t\_n}}}$$

$$K_{V\_p\_cat} = \frac{q_{V\_n}}{\sqrt{\Delta p_{p\_n}}}$$

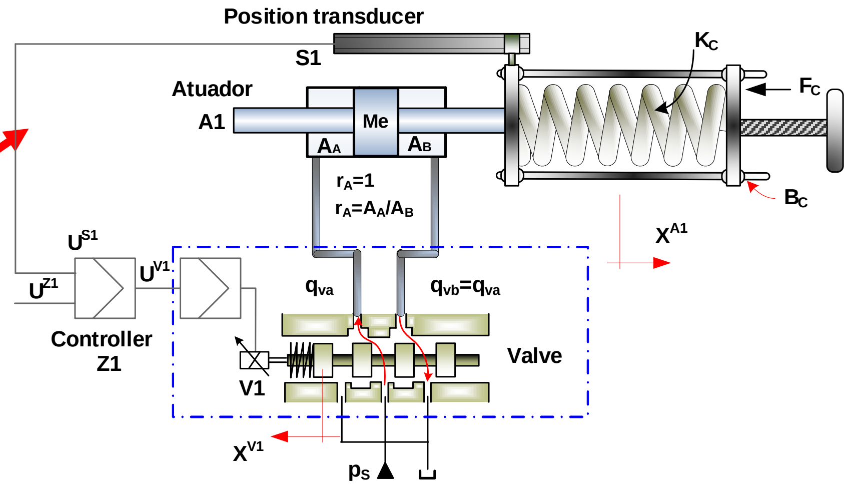
Sinnbild Symbol Symbole	V/VA max	Δp [bar]	Q nom. [l/min]
18 	24 V= 40 VA max U D-E 0 … ±10 V	35	4
			12
			24
			40

Using the Design Method

Requirements	Value
Settling time	0.13 s
Displacement	0.05 m
Overshoot?	No
Supply pressure	7×10^6 Pa
Mass	76.5 kg
Stiffness	2618.4 N/m
External Load	440 N
Friction Coefficient	730 Ns/m
Buck Modulus	1×10^9 Pa

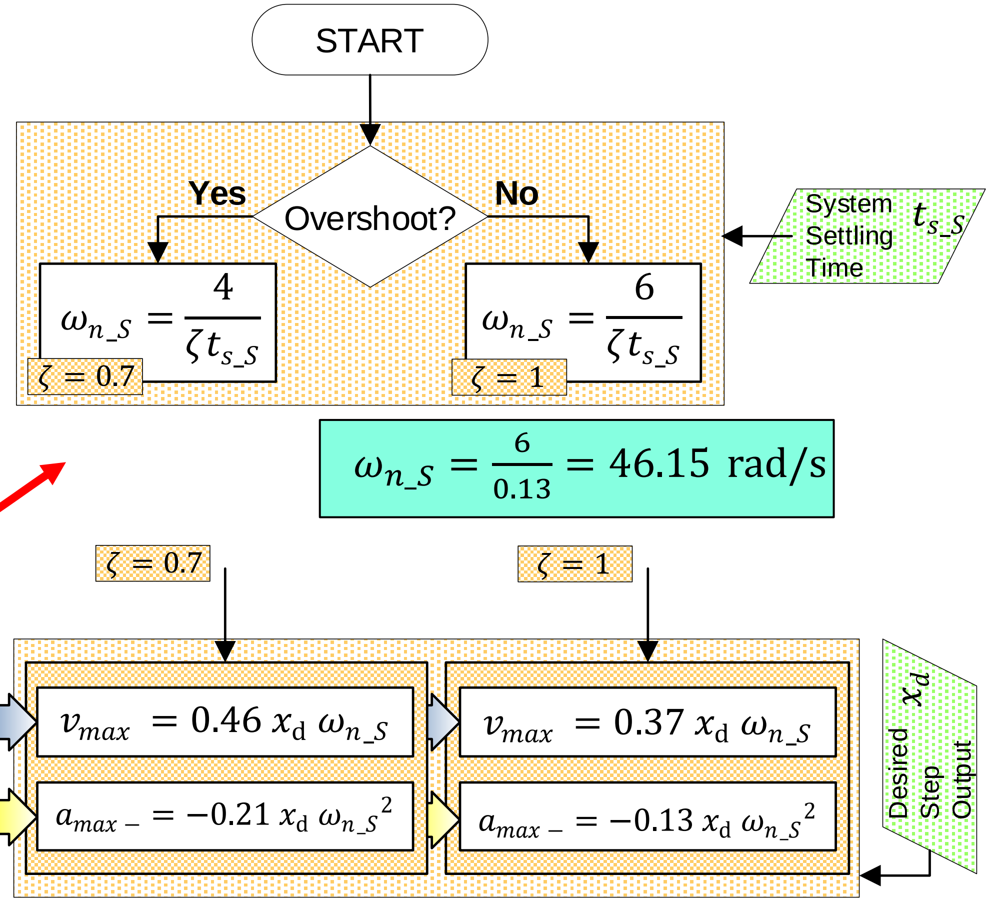


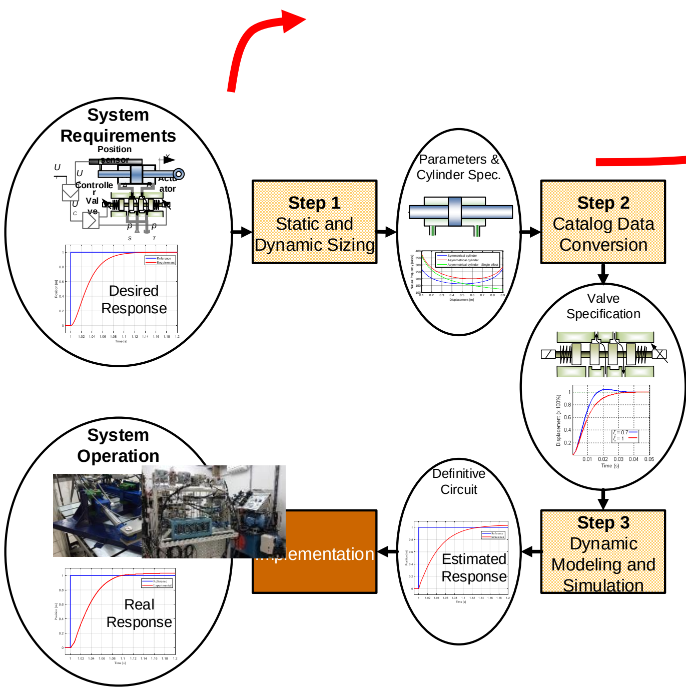
- System configurations:
  - Symmetrical cylinder + symmetrical valve



Using the Design Method

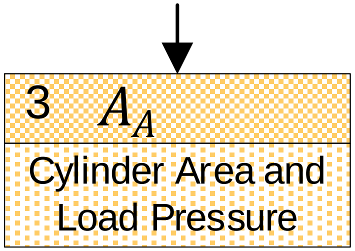
Requirements	Value
Settling time	0.13 s
Displacement	0.05 m
Overshoot?	No
Supply pressure	7×10^6 Pa
Mass	76.5 kg
Stiffness	2618.4 N/m
External Load	440 N
Friction Coefficient	730 Ns/m
Buck Modulus	1×10^9 Pa

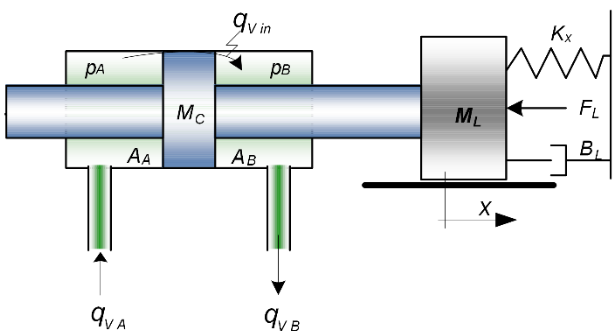


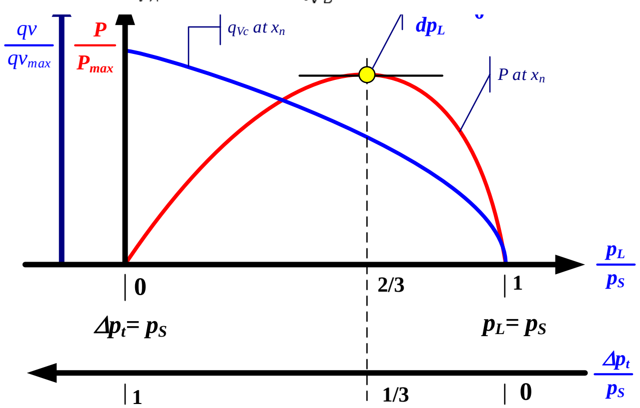


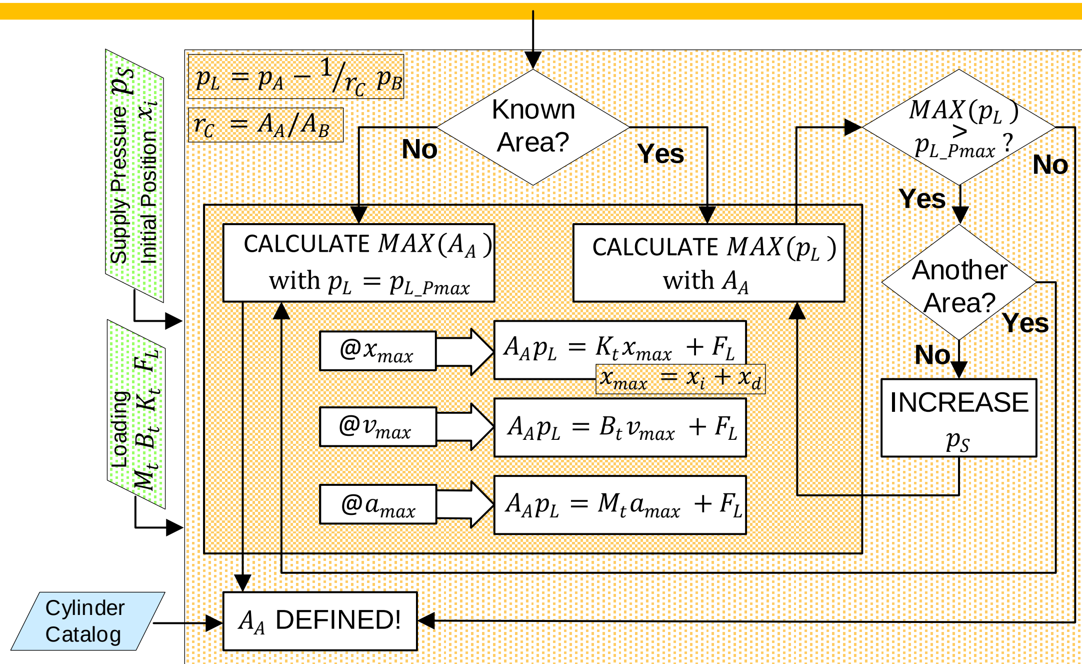
$$v_{max} = 0.37 \times 0.05 \times 309 = 0.853 \text{ m/s}$$

$$a_{max} = -0.13 \times 0.05 \times 309^2 = -13.8 \text{ m/s}^2$$









LOAD PRESSURE @ MAXIMUM POWER

4 Ports Sym ⇒ $p_{L\_Pmax} = 2/3 P_s$ | 4 Ports Asym ⇒ $p_{L\_Pmax} = 4/3 P_s$ (Extend)

3 Ports Asym ⇒ $p_{L\_Pmax} = 1/3 P_s$ | 4 Ports Asym ⇒ $p_{L\_Pmax} = 1/3 P_s$ (Retract)

$$p_{L\_Pmax} = \frac{2}{3} p_S = \frac{2}{3} 70 \times 10^5 = 46.7 \times 10^5 \text{ Pa}$$

$$@x_{max} \Rightarrow A_A = \frac{2618.4 \times 0.05 + 440}{46.7 \times 10^5} = 1.22 \times 10^{-4} \text{ m}^2$$

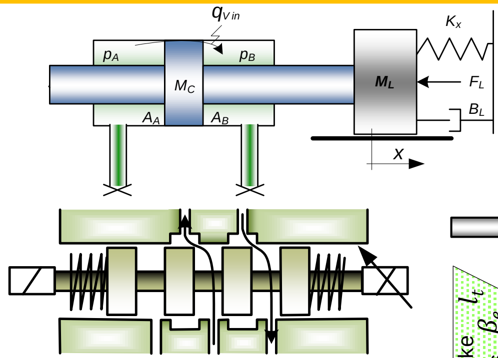
$$@v_{max} \Rightarrow A_A = \frac{730 \times 0.849 + 440}{46.7 \times 10^5} = 2.28 \times 10^5 \text{ m}^2$$

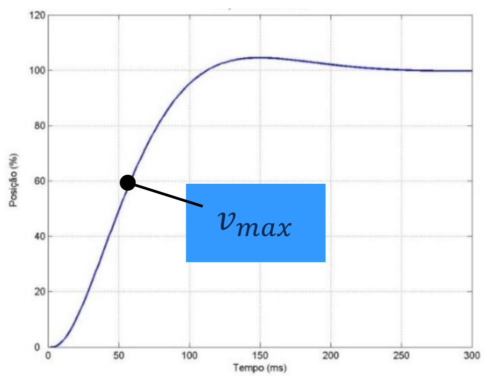
$$@a_{max} \Rightarrow A_A = \frac{76.5 \times (-13.8) + 440}{46.7 \times 10^5} = 1.32 \times 10^5 \text{ m}^2$$

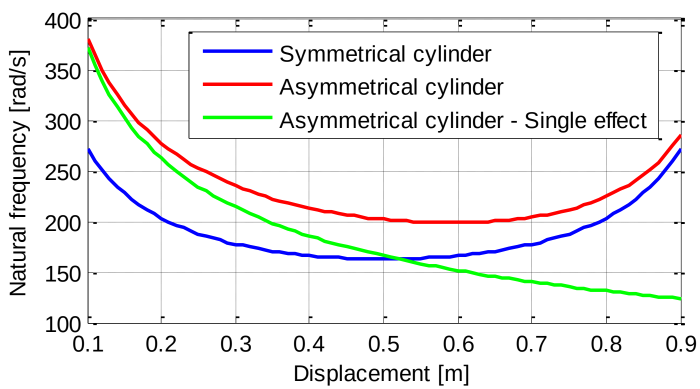
Static and Dynamic Sizing

4 $q_{V\ max}$ Maximum Flow rate

5 ω_{n_C} Cylinder Natural Frequency



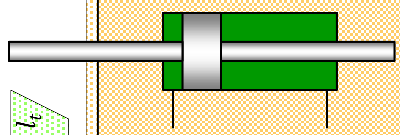




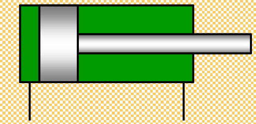
$$q_{V\ max} = A_A\, v_{max}$$

Cylinder Stroke l_t

Bulk Modulus β_e

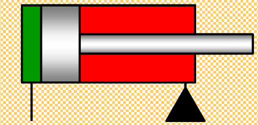


$$\omega_{n\_min} = \left(\frac{4\,\beta e\,{A_A}^2}{M_t\,V_t}\right)^{1/2}$$



$$\omega_{n\_min} = \left[\frac{\beta e}{M_t}\left(\frac{A_A}{l_A} + \frac{A_B}{l_t - l_A}\right)\right]^{1/2}$$

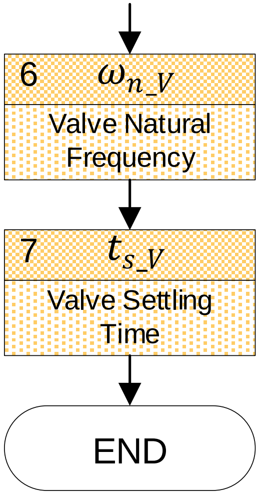
$$\text{with } l_A = \frac{A_A\, l_t}{A_A - A_B}\left[1 - \left(\frac{A_B}{A_A}\right)^{1/2}\right]$$

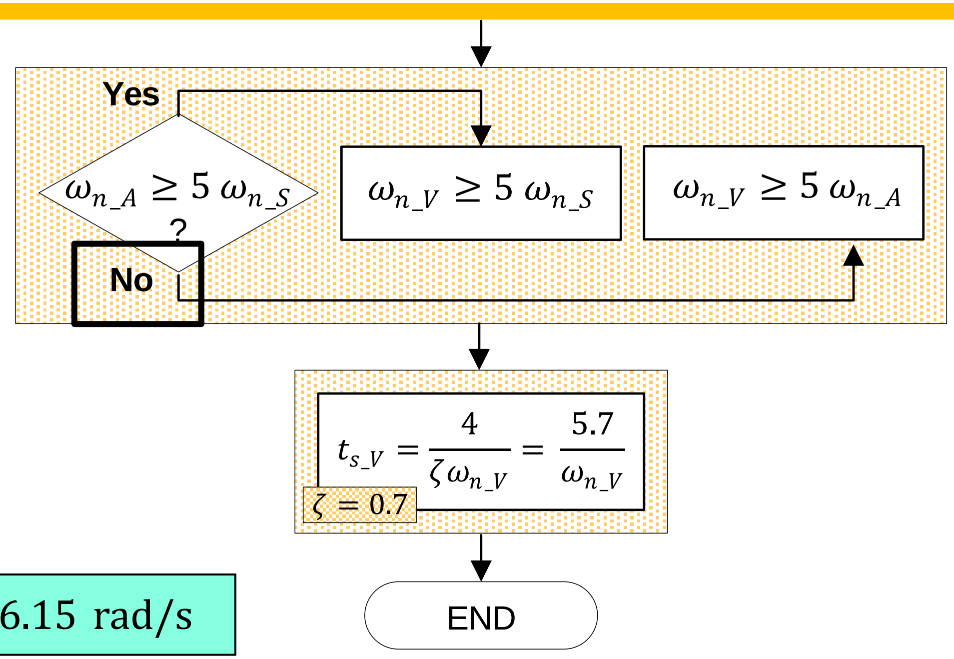


$$\omega_{n\_min} = \left(\frac{\beta_e\,{A_A}^2}{M_t\,V_A}\right)^{1/2}$$

$$q_{V\ max} = 2.28 \times 10^{-4} \times 0.853 = 2 \times 10^{-4}\ \text{m}^3/\text{s} = 12\ \text{L/min}$$

$$\omega_{n\_min} = \left(\frac{4\times1\times10^{9}\times(2.28\times10^{-4})^{2}}{76.5\times1.14\times10^{-4}}\right)^{1/2} = 218.37\ \text{rad/s}$$

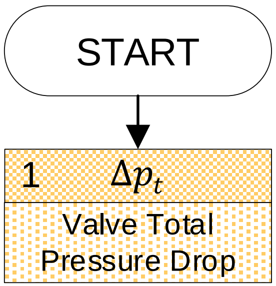


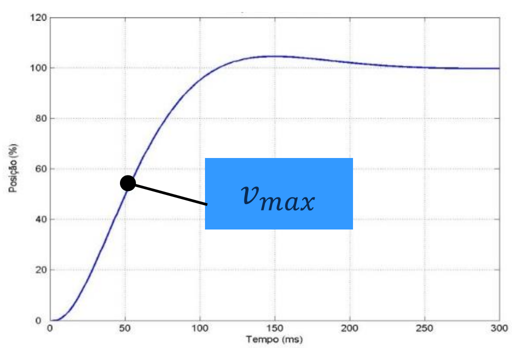


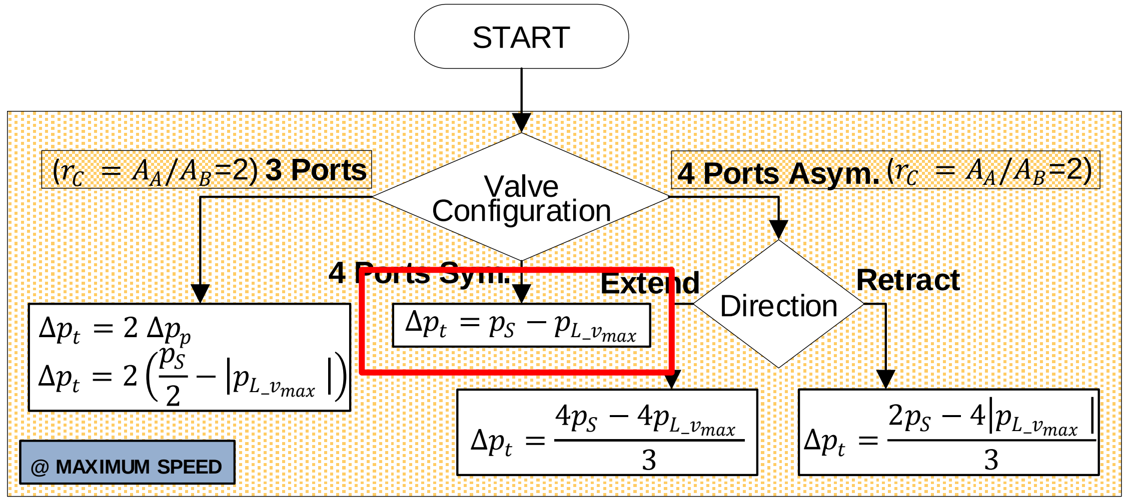
$$\omega_{n\_A} = 218.37\text{rad/s} \geq \omega_{n\_S} = 46.15\ \text{rad/s}$$

$$t_{s\_V} = \frac{5.7}{5\times\omega_{n\_A}} = \frac{5.7}{5\times218.37} = 5\ \text{ms}$$

Catalog Data Conversion







$$@v_{max} \Rightarrow A_A p_L = B_t v_{max} + F_L$$

$$\Delta p_t = 70 \times 10^5 - 46.63 \times 10^5 = 23.37 \times 10^5 \text{ Pa}$$

Catalog Data Conversion

$$K_V = \frac{2\times10^{-4}}{\sqrt{23.37\times10^5}} = 1.31\times10^{-7}\ \mathrm{m^3/(sPa^{1/2})}$$

$$K_V = \frac{12}{\sqrt{23.37}} = 2.48\ \ \mathrm{L/(min.\,bar^{1/2})}$$

2 k_V

Flow Coefficient

3 k_{V_cat}

Catalog Flow Coefficient

$$K_V = \frac{q_{V\_max}}{\sqrt{\Delta p_{t\_vmax}}}$$

Valve Catalog

Total Pressure Drop

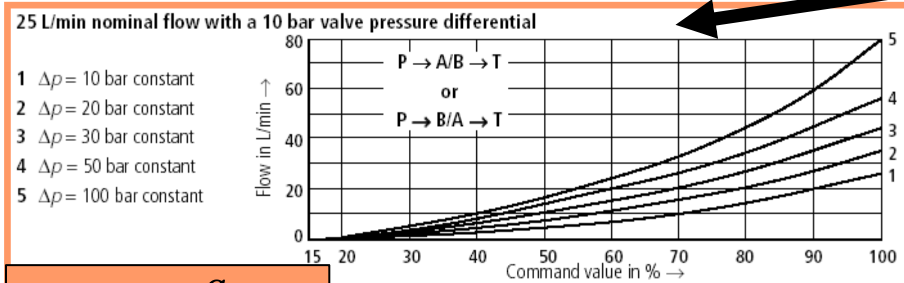
Partial Pressure Drop

$$K_{V_{cat}} = \frac{K_{V\_p\_cat}}{\sqrt{2}}$$

Selec Valve with:

$$K_V/_2 \le K_{V_{cat}} \le 2\,K_V$$

$$\omega_{n\_V\_cat} \ge \omega_{n\_V}$$



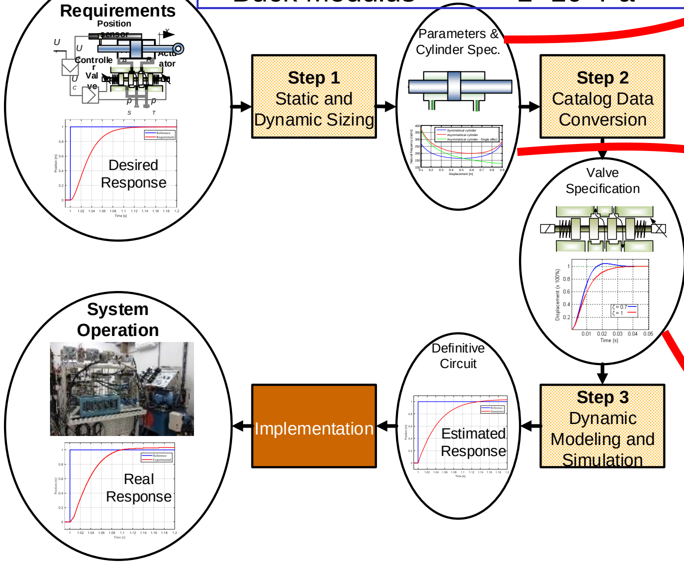
$$K_{V\_cat} = \frac{q_{V\_n}}{\sqrt{\Delta p_{t\_n}}}$$

$$K_{V\_p\_cat} = \frac{q_{V\_n}}{\sqrt{\Delta p_{p\_n}}}$$

Sinnbild Symbol Symbole	V/VA max	Δp [bar]	$Q_{nom.}$ [l/min]
18 	24 V= 40 VA max U_{D-E} 0 … ±10 V	35	4 12 24 40

Using the Design Method

Requirementv	Value
Settling time	0.13 s
Displacement	0.05 m
Overshoot?	No
Supply pressure	7×10^{6} Pa
Mass	76.5 kg
Stiffness	2618.4 N/m
External Load	440 N
Friction Coefficient	730 Ns/m
Buck Modulus	1×10^{9} Pa



Parameter	Value
System nat. freq.	46.15 rad/s
Max. Cyl. speed	0.849 m/s
Max. Negative Cyl. acceleration	-13.8 m/s^2
Max. Power Load Pressure	4.67×10^{6} Pa
Cyl. area	2.28×10^{-4} m^2
Max. flow rate	2×10^{-4} m^3/s (12 L/min)
Cylinder nat. frequency	218.4 rad/s
Valve nat. frequency	230.75 rad/s

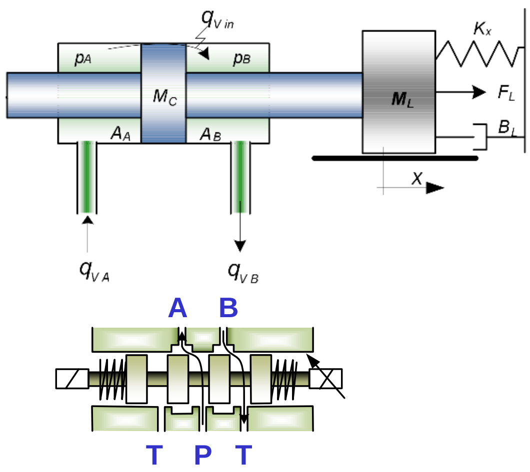
Parameter	Value
Cylinder rod diameter	18 mm
Cylinder bore diameter	25 mm
Cylinder annulus area	2.364 cm^2
Valve flow coefficient	1.31×10^{-7} $m^3/s.Pa^{1/2}$ (2.48 $L/min.bar^{1/2}$)

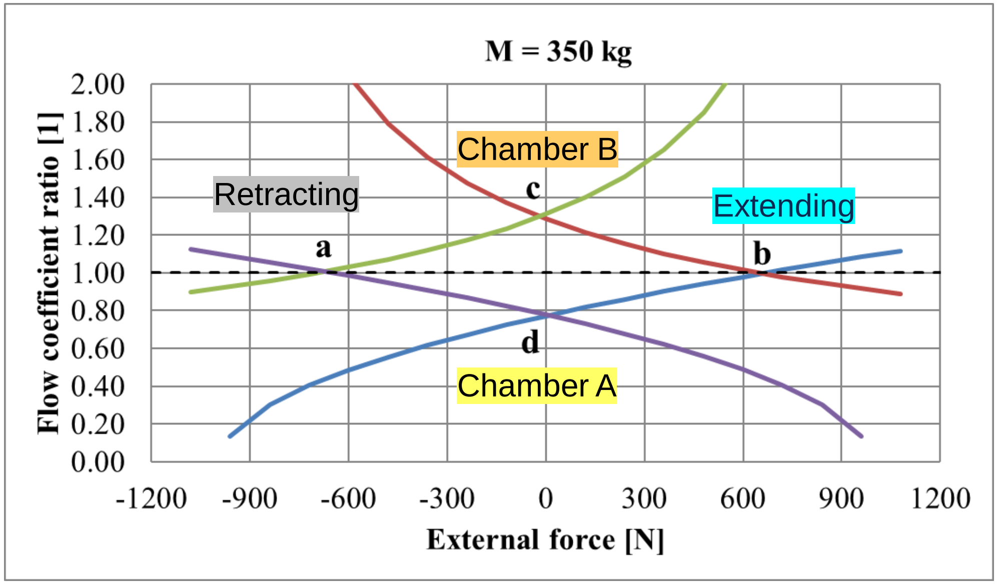
Selected components:
Bosch Rexroth CGT3MS2 25/18/200
Bosch Rexroth valve 0 811 404 038 (Kv = 2.68)

Cylinder and Valve Matching

- Symmetrical cylinder

  Maximum deceleration

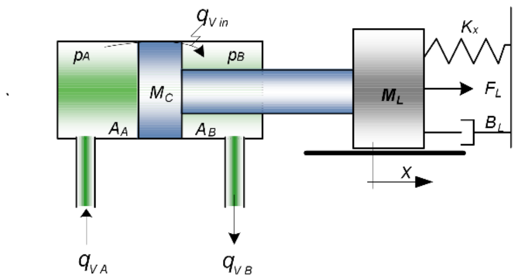


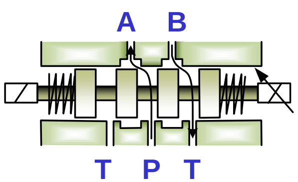


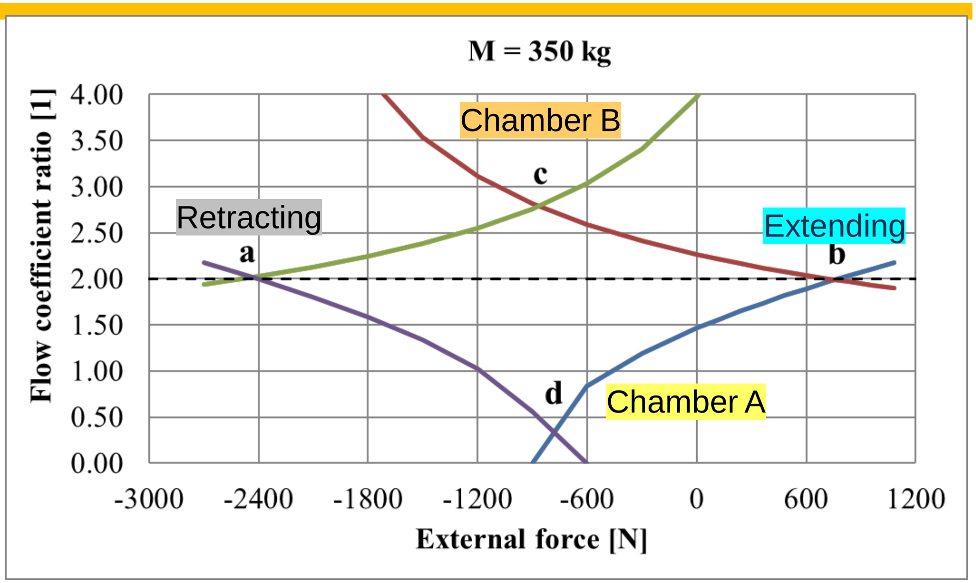
- Operating points:
  - a = Retracting: Chamber A: higher pressure; Chamber B: lower pressure.
  - b = Extending: Chamber A: lower pressure; Chamber B: higher pressure.
    - Points A and B: maximum negative and positive external forces
  - c = Chamber B: Retracting: lower pressure; Extending: higher pressure.
  - d = Chamber A: Retracting: higher pressure; Extending: lower pressure

- Asymmetrical cylinder

  Maximum deceleration



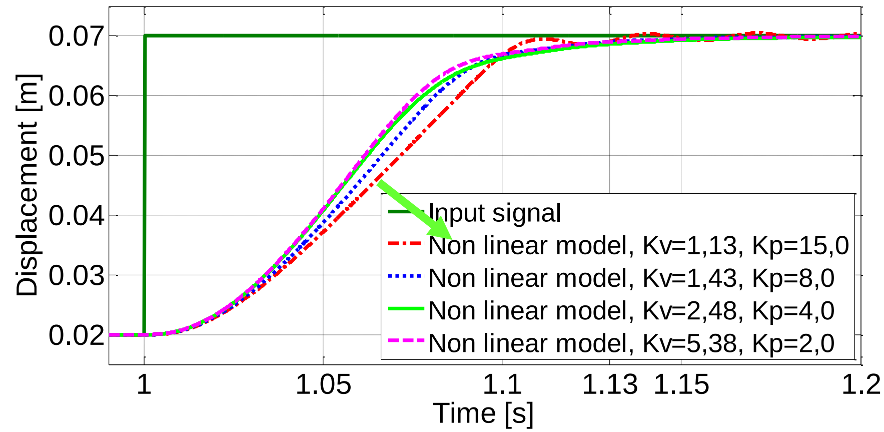


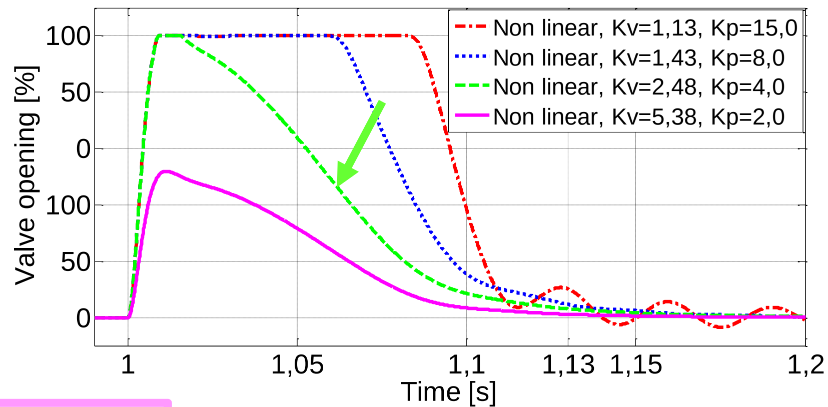


- Operating points:
  - a = Retracting: Chamber A: higher pressure; Chamber B: lower pressure.
  - b = Extending: Chamber A: lower pressure; Chamber B: higher pressure.
    - Points A and B: maximum negative and positive external forces
  - c = Chamber B: Retracting: lower pressure; Extending: higher pressure.
  - d = Chamber A: Retracting: higher pressure; Extending: lower pressure

Influence of the Valve Flow Coefficient

Flow Coefficient (Kv)		Relative to selected value
L/min.bar$^{1/2}$	$\times 10^{-7}$ m^3/s.Pa$^{1/2}$	%
1.13	0.60	~ 40
1.43	0.76	~ 50
2.48	1.31	Specified value
5.38	2.84	~ 200





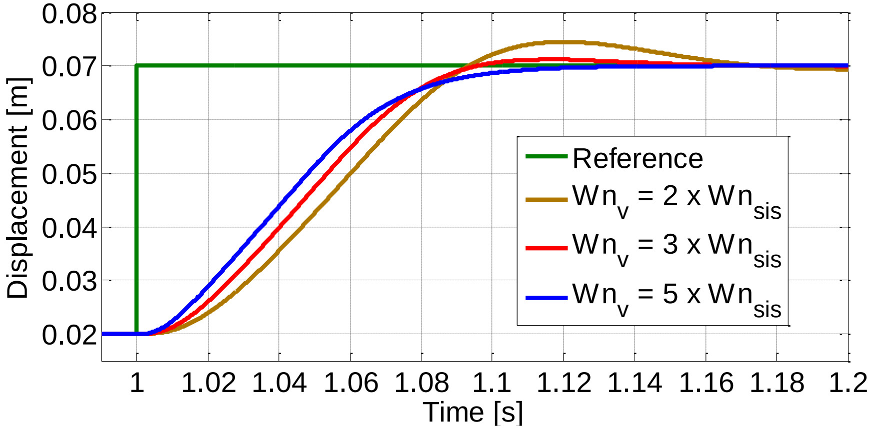
- Results (M = 76.5 kg):
  - Kv < 1.24 L/min.bar1/2 (50%)
    - Higher controller gains are required
    - Larger periods of valve saturation
    - Tendency of instability
  - Kv > 4.96 L/min.bar1/2 (200%)
    - Smaller proportional gains
    - No valve saturation
    - Larger valves do not exhibit performance improvement
    - Slower and expensive valve

- Valve Natural Frequency x Positioning System Natural Frequency:
  - Experimental system:
    - Valve natural frequency = 440 rad/s (70 Hz)
    - System natural frequency = 46.15 rad/s (7.3 Hz)
    - Without load mass and coupled spring:

$\omega_{n\_V} \approx 9.5\ \omega_{n\_S}$

Ensure a high cylinder+load natural frequency $\omega_{n\_C} > 5\ \omega_{n\_S}$

- With $\omega_{n\_V} < 3\ \omega_{n\_S}$
  - The dynamic performance cannot be achieved
- With $\omega_{n\_V} \geq 5\ \omega_{n\_S}$
  - The dynamic performance is effectively achieved
    - Desired response without overshoot



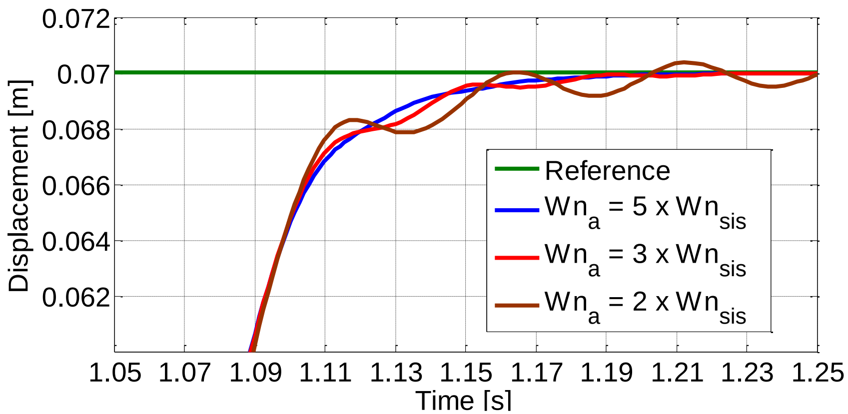
Correlation between Natural Frequencies

- Cylinder Natural Frequency x Positioning System Natural Frequency:
  - Cylinder natural frequency = 220 rad/s (35 Hz)
  - System natural frequency = 46.15 rad/s (7.3 Hz)

  $\omega_{n\_C} \approx 5\ \omega_{n\_S}$

  - Changing fluid volumes: $\omega_{n\_C} = 2\ \omega_{n\_S}$ $\omega_{n\_C} = 3\ \omega_{n\_S}$

- With $\omega_{n\_A} < 3\ \omega_{n\_S}$
  - The dynamic performance cannot be achieved
- With $\omega_{n\_A} \geq 5\ \omega_{n\_S}$
  - The dynamic performance is effectively achieved
    - Desired response without overshoot



Design Method for Hydraulic Positioning Systems

This method originated from the master's thesis by Fernando Luiz Furst (2001) and was updated based on the later contributions of other graduate students: José Roberto Branco Ramos Filho, Alisson Dalsasso Corrêa de Souza (2005), Rodrigo Spack (2008)), Irving Muraro (2009) and Mario Destro (2015).

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