Design of Positioning Hydraulic Systems

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Introduction

- Electro-hydraulic positioning systems:
  - Applied in several fields:
    - Agricultural machinery
    - Aerospace
    - Turbine speed governors
    - Injection machines
Introduction

- Design and assembly are not simple tasks:
  - The behavior needs to be analyzed according to control theory
  - Static and dynamic requirements must be fulfilled under loads not well known by the designer

- Consequences:
  - Increased development cost and time
  - Reuse of previous designs for new equipment development
The Design Method

To overcome these constraints:
- A comprehensive view of the design process is necessary

- Procedures are required for hydraulic component sizing, considering the global system behavior (closed loop)

Controller gains: 
$K_p = 2; K_i = 0,1$

Position sensor:
$K_s = 300 \text{ V/m}$

Cylinder: $d = 30 \text{ mm}$

Valve: $q_{Vn} = 32 \text{ L/min}$ ($\Delta p = 10 \text{ bar}$)
The design methodology of **mechatronic systems** can be structured into four main phases:

- Clarification of the task
- Conceptual design
- Embodiment design
- Detail design

**Pahl & Beitz Methodology**

- Technical systems
The design methodology of mechatronic systems can be structured into four main phases:

- Clarification of the task
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- Embodiment design
- Detail design
The Design Method

- Types of hydraulic circuit handled by the method:
  - (a) Symmetric proportional four-way valve and symmetric double-effect cylinder
  - (b) Asymmetric proportional four-way valve and asymmetric double-effect cylinder
  - (c) Proportional three-way valve and asymmetric single effect cylinder
Dynamic and static requirements:

- Considering a Positioning System:
  - Maximum displacement
  - Settling time
  - Forces
  - Overshoot

- Uses the mathematical expressions derived from the dynamic model of the system

- The results are the specifications for the valve and the cylinder
Step 1 - Static and Dynamic Sizing

1. \( \omega_{n}^{sys} \)
   System Natural Frequency

2. \( v_{c}^{c}; a_{c}^{c} \)
   Cylinder Speed and Acceleration

3. \( p_{c} \)
   Load Pressure

4. \( A_{c}^{c} \)
   Cylinder Area

5. \( q_{VC_{max}} \)
   Cylinder Maximum Flow Rate

6. \( \omega_{n}^{c} \)
   Cylinder Natural Frequency

7. \( \omega_{n}^{v} \)
   Valve Natural Frequency

8. \( t_{s}^{v} = 4\tau \)
   Valve Settling Time

START

END
**Step 1 - Static and Dynamic Sizing**

- **Task 1** – Calculate the system natural frequency

  \[ \zeta = 0.7 \]

  \[ \omega_n^{SYS} = \frac{5.7}{t_s^{SYS}} \]

  \[ \omega_n^{SYS} = \frac{6}{t_s^{SYS}} \]

  - Overshoot?
    - Yes
    - No

  \[ \zeta = 1 \]

  **Global system behavior**

  Time response of 2nd order under damped systems

  \[ \tau = \frac{1}{\zeta \cdot \omega_n} \]

  Time response of 2nd order critically damped systems
Step 1 - Static and Dynamic Sizing

- Task 2 – Calculate the maximum cylinder speed and acceleration

\[ \zeta_{SYS}, \omega_n^{SYS}, x^C_{max} \]

MAXIMUM SPEED

- \[ v_{max}^C = 0.46 \cdot x_{max}^C \cdot \omega_n^{SYS} \]
- \[ v_{max}^C = 0.37 \cdot x_{max}^C \cdot \omega_n^{SYS} \]

MAXIMUM ACCELERATION

- \[ a_{max}^C = x_{max}^C \cdot \omega_n^{SYS} \]
- \[ a_{max}^C = x_{max}^C \cdot \omega_n^{SYS} \]
Step 1 - Static and Dynamic Sizing

- Task 3 - Estimate the load pressure
- Task 4 - Calculate the cylinder area

\[ p_C = p_A - p_B \]

\[ p_{CP_{\text{max}}} = \frac{2}{3} \cdot p_s \]

Maximum Load Pressure: \( p_{C_{\text{max}}} \)

- At \( x_{C_{\text{max}}} \)
  \[ A^C \cdot p_C = K \cdot x_{C_{\text{max}}} + F_U \]

- At \( v_{C_{\text{max}}} \)
  \[ A^C \cdot p_C = B^i \cdot v_{C_{\text{max}}} + F_U \]

- At \( a_{C_{\text{max}}} \)
  \[ A^C \cdot p_C = M_t \cdot a_{C_{\text{max}}} + F_U \]

Cylinder Area (With \( p_{C_{\text{max}}} = p_{C_{\text{pmax}}} \))

- \( A^C \) Defined!

Valve + Cylinder \( \rightarrow \) Define Maximum Power

\( \text{Displacement (x 100\%)} \)

\[ \zeta = 0.7 \]
\[ \zeta = 1 \]

Time (s)

0.01 0.02 0.03 0.04 0.05
Step 1 - Static and Dynamic Sizing

- Task 5 – Determine the cylinder maximum flow rate:

\[ q_{VC\ max} = A^C \cdot v_{max}^C \]

- Task 6 – Determine the cylinder natural frequency

\[ \omega_{n\ min}^C = \left( \frac{4 \cdot \beta e \cdot A^{C^2}}{M_t \cdot V_t^C} \right) \]
Step 1 - Static and Dynamic Sizing

- Task 7 – Calculate the valve natural frequency
  
  \[ \omega_C \geq 5 \omega_{SYS} \]

  - If Yes:
    \[ \omega^V \geq 5 \omega_{SYS} \]
    \[ \omega^V \geq 5 \omega^C \]
  - If No:
    \[ \omega_{SYS} = \frac{5.7}{t_s} \]

- Task 8 – Calculate the valve settling time
  
  \[ t_s = \frac{5.7}{\omega_n} \]
Step 2
Conversion of Cataloged Data

1. $\Delta p_t$
   Pressure drop at valve at maximum flow rate

2. $K_v$
   Flow rate coefficient

3. $K_{v_{cat}}$
   Catalog flow rate coefficient

4. $q_{VC_{max}}$
   Maximum flow rate at the valve

END
Step 2
Conversion of Cataloged Data

- Task 1 – Calculate the pressure drop at valve at maximum flow rate

\[ \Delta p_t = p_S - p_{C \text{ max}} \]

- Task 2 – Calculate the flow rate coefficient

\[ K_v = \frac{q_{VC \text{ max}}}{\sqrt{\Delta p_t}} \]
**Step 2**
Conversion of Cataloged Data

- **Task 3** – Calculate catalog flow rate coefficient
  - For total pressure drop
    \[
    K_v^{cat} = \frac{q_{v_n}}{\sqrt{\Delta p_{tn}}}
    \]
  - For partial pressure drop
    \[
    K_v^{cat} = \frac{q_{v_n}}{\sqrt{2\Delta p_{pn}}}
    \]
  - The valve must have a Flow Rate Coefficient similar to or greater than the specified value:
    \[
    0.5K_v < K_v^{cat} < 2K_v
    \]
Step 2
Conversion of Cataloged Data

- Task 4 – Maximum flow rate at the valve

\[ q_{VC_{\text{max}}} = K_{v_{cat}} \cdot \sqrt{\Delta p_t} \]

- Power limits of the valve:

- Dynamic response:

Step 1 – Tasks 7, 8

\[ t_s^V = \frac{5.7}{\omega_n^V} \]
Step 3  
Dynamic Behavior Study

- Simulation is recommended to validate the design obtained
- A reliable model must be available
- Simulation avoids the construction of a system with unexpected behavior
Method evaluation

- LabVIEW program created to implement the method

- Matlab/Simulink model created to validate the method

- Tests carried out on Proportional Hydraulics Platform
  - Positioning hydraulic system controlled by PC with VXI data acquisition system
Method evaluation

- The Valve+Cylinder Specifications came from the Design Method

- Step Response:
  - Comparison of simulation and experimental results
  - The model gives confidence for purely theoretical analysis

<table>
<thead>
<tr>
<th>Requirements</th>
<th>Results</th>
</tr>
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<tbody>
<tr>
<td>$t_s^{SYS}$ [ms]</td>
<td>450</td>
</tr>
<tr>
<td>$x^C_{max}$ [mm]</td>
<td>80</td>
</tr>
<tr>
<td>Overshoot?</td>
<td>No</td>
</tr>
<tr>
<td>$\omega_n^{SYS}$ [rad/s]</td>
<td>12.69</td>
</tr>
<tr>
<td>$p_s$ [Pa]</td>
<td>4.50e6</td>
</tr>
<tr>
<td>$A^C$ [cm^2]</td>
<td>2.36</td>
</tr>
<tr>
<td>$F_u$ [N]</td>
<td>708</td>
</tr>
<tr>
<td>$B$ [N·s/m]</td>
<td>340</td>
</tr>
<tr>
<td>$M_i$ [kg]</td>
<td>5.50</td>
</tr>
</tbody>
</table>
Method evaluation

- System was tested under different loads
  - Faster and with overshoot when tested under a smaller load
  - Was not able to meet the settling time specification with higher loads
Method evaluation

- Performance of the system for different valves
  - Proportional gain was reduced to the minimal value
  - With $K_v = 2$ $K_v$ calculated, the system met all specifications
  - With $K_v = 3$ $K_v$ calculated:
    - The system has considerable overshoot and faster response than required
    - Reduced gains make the system slow close to the set point and increase the influence of the valve dead band and cylinder friction
Method evaluation

- **Kv > 2 Kv calculated:**
  - smaller proportional gains and no valve saturation.
  - Larger valves do not show performance improvements and are, in practice, more slow and expensive.

- **Kv < 0.5 Kv calculated:**
  - high controller proportional gains, great periods of valve saturation and tendency for oscillating movement.
Conclusions

- A design method for positioning hydraulic systems was presented.

- Case results and practical applications have demonstrated that the method is a useful tool for hydraulic positioning system design.

- The method allows a comprehensive view of the valve and cylinder requirements in the closed loop system.

- The flow rate coefficient helps the designer to select the valve from several options.
Workshop on Innovative Engineering for Fluid Power and Vehicular Systems

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