Investigation of a New Orbital Steering Concept With Focus on the Control Loop Performance

Olesen, Emil Nørregård; Andersen, Torben Ole

FPMC2022-90897

INVESTIGATION OF A NEW ORBITAL STEERING CONCEPT WITH FOCUS ON THE CONTROL LOOP PERFORMANCE

Emil N. Olesen[1],[†],[*], Torben O. Andersen[2],[†]

[1]Danfoss Power Solutions, Nordborg, Denmark
[2]AAU Energy, Aalborg, Denmark

ABSTRACT

This paper investigates a new asymmetric steering concept within orbital steering units called OSPS, which asymmetry enables new steering features for the operator of the vehicle compared to traditional OSP variants. A traditional orbital steering unit functions as a proportional valve system with a certain overlap, where the operator input to the steering wheel will initiate the rotation of the wheels with a small dead-band of 2-3 degrees. The new asymmetric unit functions instead as a proportional valve system with a certain under-lap, where the operator will initiate the rotation of the wheels directly related to the steering wheel rotation instead. The objective with this paper is to evaluate the closed loop performance for the new asymmetric concept, because the right and left control loop is changed compared to a tradional OSP unit.

Keywords: Orbital Steering Unit, Nonlinear Control System, Disk Margin, Relative Stability margins

1. INTRODUCTION

Orbital Steering Pumps (OSP) [1] has for more than 50 years been utilized as power steering systems for off-road vehicles in a large variety of applications, from small loaders to large scale tractors and combines. These units functions as a closed loop control system, with the input being the rotation of the steering wheel and corresponding output being the rotation of the wheel's or linear movement of a hydraulic cylinder.

The steering units control system has a mechanical feedback and consist of mechanical parts which actuates a hydraulic valve system and gearset through a spring package, the main parts of a OSP is shown in figure 1.

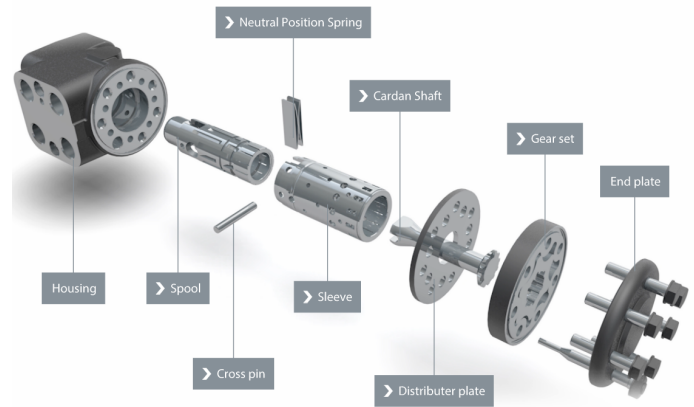The working principle is described in the following, with reference to parts indicated in figure 1.

FIGURE 1: Exploded view of a orbital steering unit (OSP).

In the figure an exploded view is shown, the spool is in a application attached to a steering column through a spline connection. This spool and column is able to rotate in the housing together with the sleeve and gearset. The spool and column achieves an input from the operator, and will thereby follow the operators desired trajectory for steering. However, the sleeve and gearset will not follow the spool immediately. Because of the spring package, it needs a small relative deflection between the spool and sleeve before the sleeve and gearset starts to rotate. This relative angle works with the same analogy as a error in a closed loop control system, where the mechanical feedback is the sleeve and gearset angle. This means that the spool angle will be ahead of the sleeve and gearset angle, which in the system can be related to the small delay between operator input and wheel rotation.

In a traditional unit the gearset [1] is driven symmetrically for steering left and right, where the oil will flow from the spool and sleeve, overlapping valve system, through the gearset, rotary meter that meters the oil to the cylinder. The returning oil from

the opposite side of the cylinder will be led to tank through the spool and sleeve.
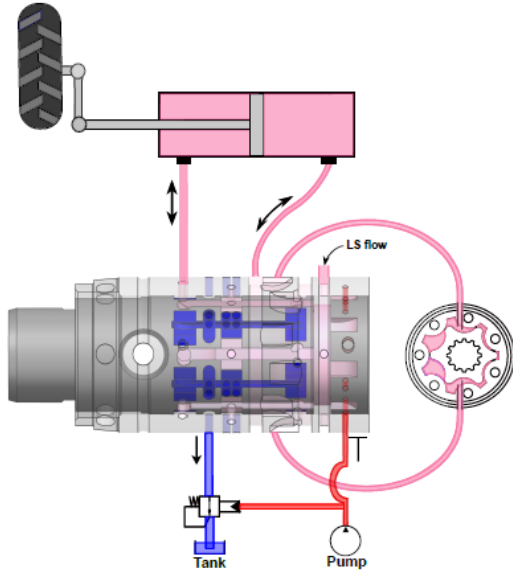

FIGURE 2: System overview of asymmetric concept.

In the new concept the flow path is changed, such that when steering right the flow is as usual going through the spool and sleeve and then metered through the gearset to the cylinder. However the left side is changed, such that the oil is directly flowing to the cylinder, and it is instead the returning flow which is metered to tank. This change in architecture enables the possibility to design the spool and sleeve as a under-lapping valve system instead. This change eliminates the dead-band, because the cylinder at all time is controlled trough a pressure balancing valve system where a small deviation of the relative angle will cause a pressure change for either left or right cylinder port. It is necessary for the asymmetric design to have a small flow at all time, to ensure it can create the pressure balancing of the cylinder ports, where the flow will be balanced between port connection and tank connection. In this paper a prototype for a Dynamic Load Sensing unit is considered, where the LS line delivers a small amount of flow. A overview of the internal flow path is shown in figure 2 and a corresponding system diagram is shown at figure 3.

The spool and sleeve valve consist of the bleed rows A1, A2, A1314, A3 and A10. These bleeds are utilized to direct and control the amount of oil flowing through the unit, and creating the desired performance of the closed loop system.

The traditional OSP [1] has more bleed rows, typically A1, A2, A3, A4 and A5, which has the same influence on the system response for left and right turning. In this new concept there is a change in how the control performance is affected for left and right, due to the asymmetry. The objective of this paper will be to analyze and study the asymmetric influence. To study this a model of the system is necessary together with a verification with experimental measurements. Based on the obtained model it is then possible to derive a linear transfer function, and evaluate the closed loop performance for given operating points. The focus in this paper will be the relative stability of the system for left
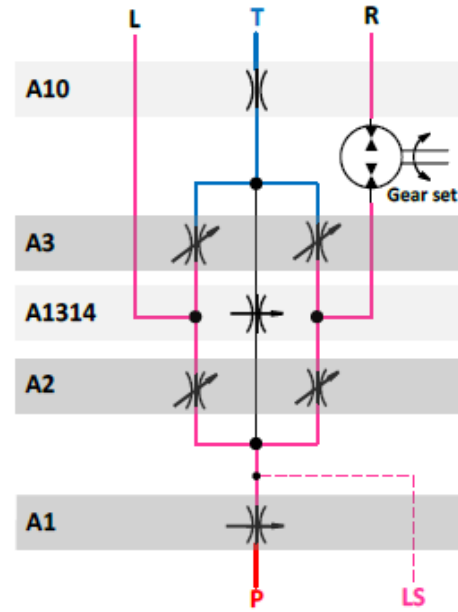

FIGURE 3: Hydraulic diagram of asymmetric concept.

and right, where the disk margin approach is used to evaluate different parameters influence. The paper will cover this in the following sections:

- Establishment of Time Domain Model

- Verification of Model

- Linear Analysis

- Stability Analysis with Disk margin

The studied OSPS prototype unit has a gearset size of 125 cubic centimeters, and is operated with a priority valve setup, which can deliver up to 80 L/min. The unit has been tested with laboratory facilities and at a tractor in South Denmark, at the Nordborg plant, for Danfoss Power Solutions.

2. TIME DOMAIN MODEL

This section will describe a standard steering setup for a Dynamic Load Sensing Steering system. It will consist and cover the hydraulic components: a priority valve, the OSPS unit and a cylinder load. In the following the analytical relations are derived for the system.

2.1 System Model

The theory is explained with basis in the hydraulic diagram shown in figure 4. For a more deeply description, see the reference [2].

First component, the priority valve OLS 80 [1], is a flow compensator for the steering unit, which regulates the flow from the pump towards steering circuit and working hydraulics. The priority valve regulates the amount of flow depending on the
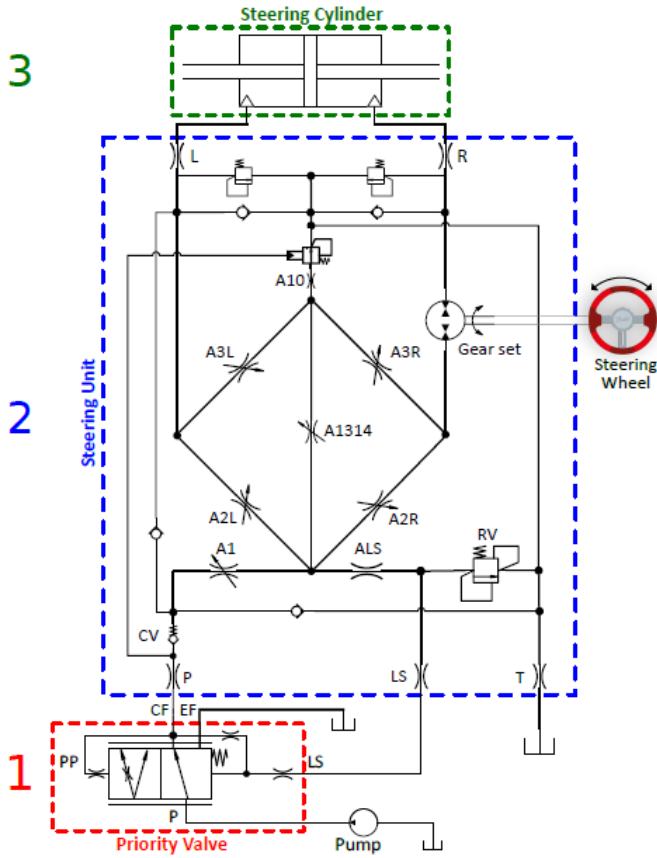
<div align="center">2</div>

FIGURE 4: Hydraulic diagram of the steering system.

FIGURE 5: Hydraulic diagram of the priority valve.
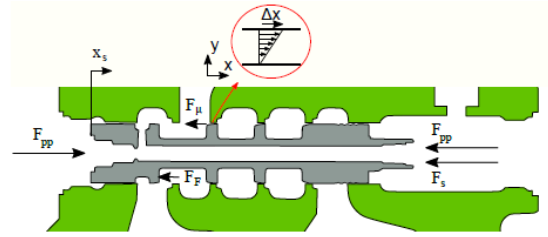
FIGURE 6: Free body diagram of mechanical spool priority valve.

LS signal from the steering unit, which ensures that the steering circuit is prioritized over the working hydraulics.

The priority valve consist of a combined mechanical and hydraulic system, where the mechanical movement of the internal spool controls the opening towards steering (CF) port and working hydraulics (EF) port. The hydraulic system, shown in figure 5, can be described with the orifice equation (1), and the continuity equation (2).

$$Q_i = C_d A \sqrt{\frac{2}{\rho} \Delta P} \qquad (1)$$

$$Q_{in} - Q_{out} = \frac{dV}{dt} + \frac{V}{\rho} \frac{dp}{dt} \qquad (2)$$

The valve dynamics of the mechanical spool internally, can be described with Newtons second law (3), where the force equilibrium is shown in figure 6. The F_{pp} is the pressure point p_{pp} working on the spool area against the pressure force F_{ls}, the friction from the oil film F_μ, flow forces F_F and the margin spring F_s.

$$\ddot{x}_s M_s = F_{pp} - F_{ls} - F_F - F_s - F_\mu \qquad (3)$$

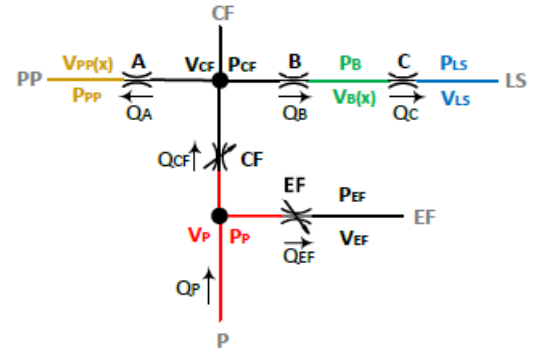Second component, the OSPS unit, is working as described in the introduction 1. Again here the system consist of a hydraulic part and a mechanical part. The hydraulic system is shown in figure 7, and can be described with the same equations as for the priority valve, the continuity equation (2) and orifice equation (1).

The valve dynamics of the OSPS unit depends on the mechanical system, which is illustrated with a free body diagram in figure 8. A description of the torques introduced while steering are described in the following, and expressed with equation 4 and 5.

FIGURE 7: Hydraulic model illustration of OSPS.

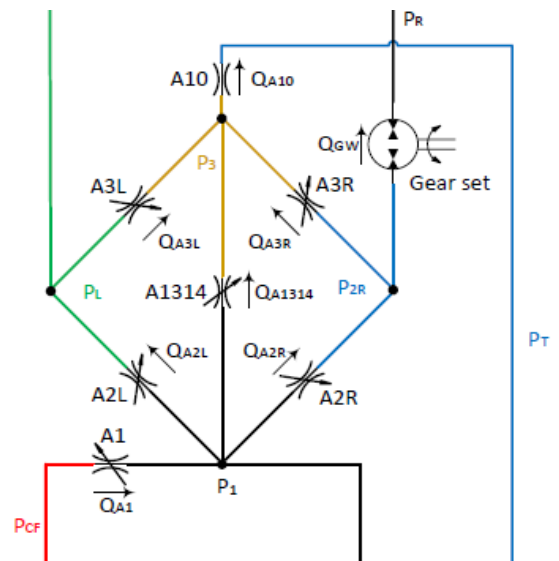$$J\ddot{\theta} = \tau(t) - k(\phi)(\theta - \theta_1) \tag{4}$$

$$J_1 \ddot{\theta}_1 = \tau_M(\phi) - \tau_\mu(\dot{\theta}_1) - k(\phi)(\theta_1 - \theta) \tag{5}$$
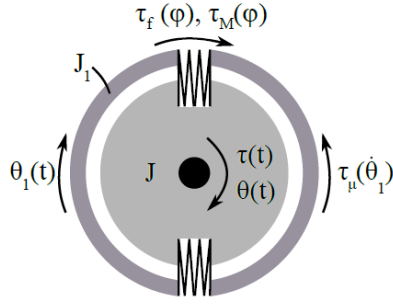

FIGURE 8: Free body diagram of the mechanical system of OSPS.

When the steering wheel is turned a steering torque $\tau(t)$ is applied at the steering wheel, which results in an acceleration of the spool and steering column inertia (J) and an elastic deformation of the rotational spring (ϕ). Simultaneous with this a spring torque (τ_f) will be applied to the sleeve and gearset together with a hydraulic motor torque (τ_M) from the gear set which will depend on the slip angle. The resulting torque from these two contributions will be transferred into a acceleration of the inertia of the sleeve and gearset (J_1) which is affected of a opposite acting friction torque (τ_μ).

The third component, the cylinder load, is attached to the steering geometry of a given vehicle and will in this way rotate the wheels linearly dependent on the metered flow from the gearset. The cylinder will in each application have an attached mechanical system with a dynamic response depending on this, it is therefore a critical part for the system model to be able to understand the dynamic influence.

The tractor utilized in this paper is simplified to a mass-spring-damper system, which is shown in figure 9 together with the hydraulic cylinder.
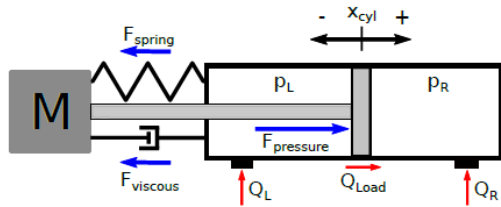

FIGURE 9: Figure for simplified tractor load.

The hydraulic volumes of the cylinder for each side R and L, is described with the continuity equations (6) and (7). Where the change in volume and flow is described with equation (8) and (9).

$$\dot{p}_L = \frac{\beta}{V_{L0} + \Delta V}(Q_L - Q_{load}) \tag{6}$$

$$\dot{p}_R = \frac{\beta}{V_{R0} - \Delta V}(Q_R + Q_{load}) \tag{7}$$

$$\Delta V = A_{cyl} x_{cyl} \tag{8}$$

$$Q_{load} = A_{cyl} \dot{x_{cyl}} \tag{9}$$

The mechanical system is again described with Newtons second law, with the expression (10).

$$\ddot{x_{cyl}} M_{cyl} = A_{cyl}(p_L - p_R) - F_{\mu cyl} - k_{cyl} x_{cyl} \tag{10}$$

Where the friction of the mechanical tractor system is modelled as being coulumb and viscous, expressed with equation (11).

$$F_{\mu cyl} = C_{cyl} \frac{\dot{x_{cyl}}}{|\dot{x_{cyl}}|} + B \dot{x_{cyl}} \tag{11}$$

3. VERIFICATION OF MODEL

In this section a verification of the derived model of the system will be covered, where both laboratory test and vehicle test on the tractor will be evaluated and compared. For both cases the measured input for the steering unit and pump supply will be fed to the model, to verify that the transient response of pressure and velocity is caught with the model. The dynamics of the system is of great concern here, because these represent the behavior of the closed loop performance.

The section will be split into two, a subsection covering laboratory test and one covering tractor test.

3.1 Laboratory test

Two different input cases is considered to tune model parameters and verify transients for the steering unit and priority valve. In both cases the load seen by the steering unit is a pressure relief valve system, which delivers a almost constant servo load pressure for the unit independent of the steering velocity.

First input case is a static test performed to verify area characteristics and hydraulic parameters as discharge-, friction coefficients and so on. Test is performed with a ramp input from 0 to 110 RPM for both left and right side. Graphs showing the correlation between steering velocity and four of the pressure points is shown in figure 10, where pressure behavior is matching from 0 to 60 RPM, and then a deviation is introduced.

Second and more interesting test performed to verify transient response, is a step input test where the steering wheel is stepped in 10 seconds to 30 RPM and then left at 0 RPM again. This test indicates if the pressure transients is caught, and verifies time constants of the model. Four pressure points is again shown with a graph in figure 11, where it is concluded that the model dynamics can represent the steering units.

From the comparison there is a clear tendency between model dynamics and static relations depending on the steering velocity for a setup with priority valve and steering unit alone with a almost constant load case. It is therefore chosen to use the achieved parameters in the evaluation of the tractor setup to reduce the amount of unknown values from the tractor setup.
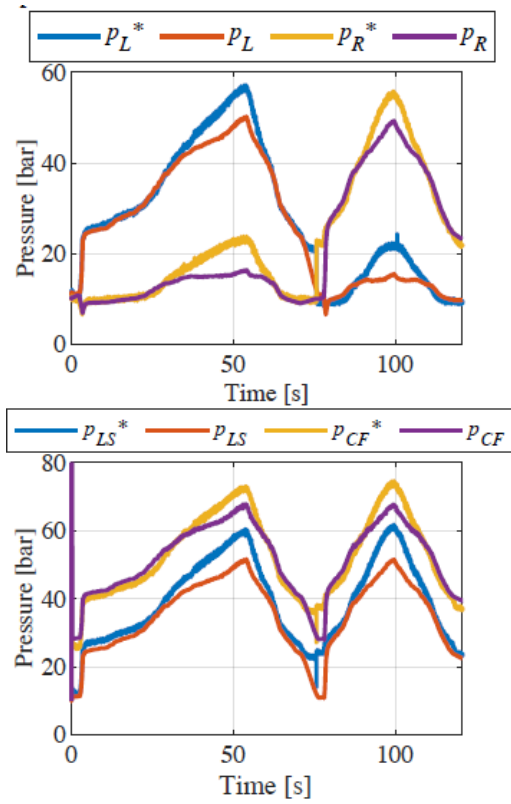
FIGURE 10: Comparison of first input for pressure points depending on steering velocity, starred is measured values.

FIGURE 11: Comparison of second input for pressure points depending on step input 0 to 30 RPM, starred is measured values.

3.2 Tractor Test

One input test from the vehicle test will be covered in this subsection, which is a snake movement test of the tractor. The tractor is here maneuvered by the operator such that the steering wheel follows a snake movement trajectory. This test combines left and right steering at a almost constant vehicle velocity of approximately 10 km/h.

It is here interesting to see how the dynamics of the tractor influences the steering unit, and secondly how well the model can replicate the different state responses with the relatively simple model for the tractor setup. Figure 12 and 13, shows a comparison between pressure points, flows and cylinder position depending on time during the snake trajectory.

By examining the pressure graphs, it can be verified that the oscillations in the pressure points is closely related with a deviation in the steady state value, but this deviation is concluded to be acceptable compared to the level of complexity for the model. It is therefore quite impressive that the pressure dynamics can be replicated that well, with a fairly simple tractor model.

With reference in the laboratory and vehicle test and comparisons shown it is verified that the model can replicate the dynamics of the system and hereby be utilized to describe the closed loop performance for the system. The next section will cover a linear analysis of the derived model, to study the closed loop performance.
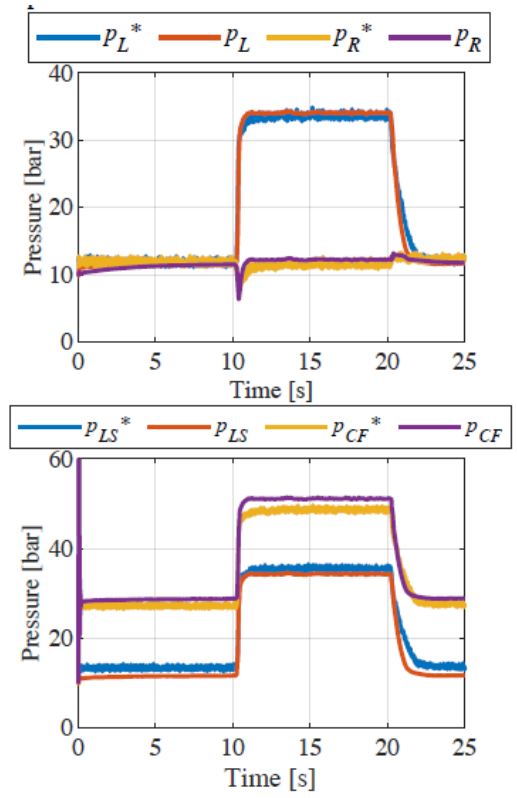
4. LINEAR ANALYSIS

In this section, a linearized model will be presented. The linear analysis is here chosen to be able to evaluate the relative stability of the system and analyse the control performance.

The nonlinear system model is linearized through Taylor approximation, where the nonlinear partial differential equations are approximated with a first order Taylor expansion see equation 12. Then it is subtracted such that it describes a constant change in the variables, see equation 13. This approach is needed because of the criteria for control theory, demanding it to be linear time invariant system.

$$\widetilde{f}(x) = f(x_0) + \frac{\partial f(x)}{\partial x_1}|_{x_0} (x_1 - x_{10}) \Rightarrow \quad (12)$$

$$\widetilde{f}(x) = f(x_0) + k_1 (x_1 - x_{10}) \Rightarrow$$

$$\widetilde{f}(x) - f(x_0) = k_1 (x_1 - x_{10}) \Rightarrow$$

$$\Delta f(\Delta x) = k_1 \Delta x_1 \quad (13)$$

The linear transfer function obtained for the system is formulated such that the input is the steering velocity of the spool and the output is the velocity of the sleeve and gearset. This relates the mechanical closed loop system of the unit itself, and a Bode Plot is shown in figure 14 and 15, which compares right and left SISO systems for open and closed loop response.

The presented Bode Plots indicate no noticeable difference in the gain magnitude of the closed-loop response when turning
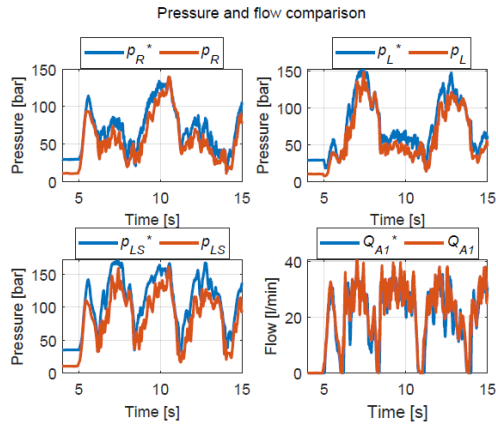
5

FIGURE 12: Comparison of pressure points depending on snake steering trajectory for tractor test, starred is measured values.
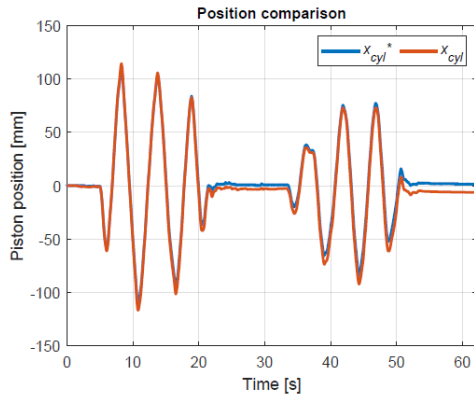

FIGURE 13: Comparison of cylinder position depending on snake steering trajectory for tractor test, starred is measured values.

FIGURE 14: Bode Plots of the two SISO open-loop turning systems.
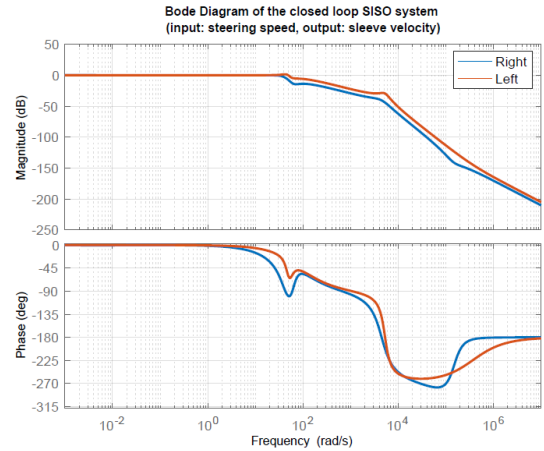
FIGURE 15: Bode Plots of the two SISO closed-loop turning systems.

left or right. But when comparing the relative stability during a right and a left turn, the Bode Plots indicate a difference. The relative stability is therefore further analysed with the disk margin approach.

5. STABILITY ANALYSIS WITH DISK MARGIN

The relative stability of a system measures how robust a system is to change in the phase or gain of the system in terms of phase- and gain margins. These margins describe how much an open-loop system may vary in terms of phase and gain before its closed-loop system becomes unstable. It is therefore very interesting to investigate the sensitivity of the system parameters towards relative stability. It is however not as straight forward, because there exist many parameters, 9. order system, in the unit which can have a coupled effect that can mislead the results obtained.

Disk margin is a relatively new method of evaluating the relative stability of an LTI system. The reasoning behind this theory is that the classical stability margins only examine pure gain or pure phase margin but not both simultaneously. Hence, systems with both high gain and phase margin may become unstable if the system is exposed to a combination of phase and gain pertur-
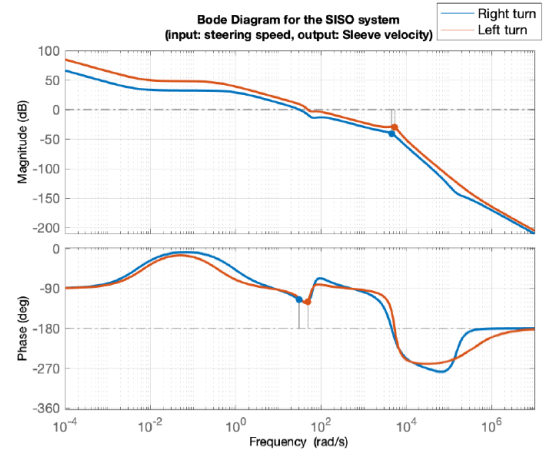
bations [3]. The new method is described with figure 16, where the closed loop system is affected by a pertubation f that changes the gain and phase margin of the system.

Figure 17 shows how the variation of f affects the system where the classical stability margins are shown alongside stable and unstable combinations of phase and gain. Black circles indicate the stable variations of f determining the classical phase and gain margins alongside the periphery of the unit circle and the real axis. Furthermore, combinations of f that do not yield pure change in phase or gain are plotted. The green x indicates that these systems are stable, while red indicates unstable systems.

The classical stability margins is a bit misleading with a large order system as the case here. This is shown in figure 18, where the suggested stable regions from disk margin is compared with the classical ones. It is here clear to see why the approach is called disk margin, because it illustrates a stable disk in the optimal point, where the highest gain and phase margins are achieved for the closed loop system.

The question is then how this approach can be utilized in the design phase to gain knowledge for the designer about the closed loop performance. A study of the different areas and discharge
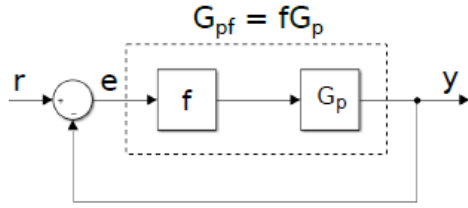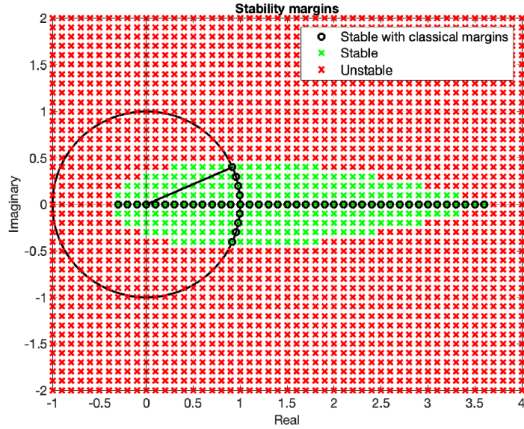
6

FIGURE 16: Feedback system with perturbation f.

FIGURE 17: Visualisation of stability margins.

FIGURE 18: Stable regions of the ninth order left turn system.
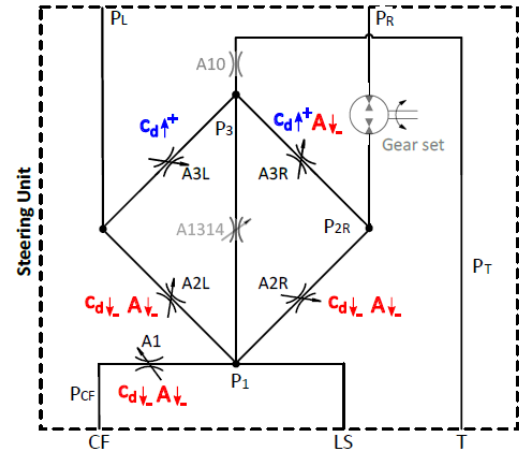
FIGURE 19: Bleed configuration increasing relative stability for a left turn.

coefficients for individual bleeds of the spool and sleeve identified which parameters that affected the margins. It was here found that a change in an area or discharge coefficient through a optimization of the disk margin could improve the asymmetric Bode Plots for the closed loop response. Figure 19, illustrates how a design change in decreasing and increasing individual bleeds of the unit could impact this. It has to be mentioned here that this change is a optimized behavior for only one operating point, a further study of the whole range needs to be done to ensure a overall improvement of the response. But it clearly visualises that a relatively high system order can be impacted by this approach and utilized as a design tool.

6. CONCLUSION

The objective of the paper was to analyze the influence on the closed loop performance from a new asymmetric concept which asymmetry enables new steering features for the operator of the vehicle compared to traditional OSP variants.

From this analysis it can be concluded that an nonlinear model is derived and verified, which enables to replicate the dynamic response of the closed loop system for the orbital steering unit OSPS.

The nonlinear model is linearized, to achieve a linear transfer function for the system with input being the spool velocity and output being the sleeve velocity.

The relative stability of the linear transfer function is optimized with regard to phase and gain margin, where the disk
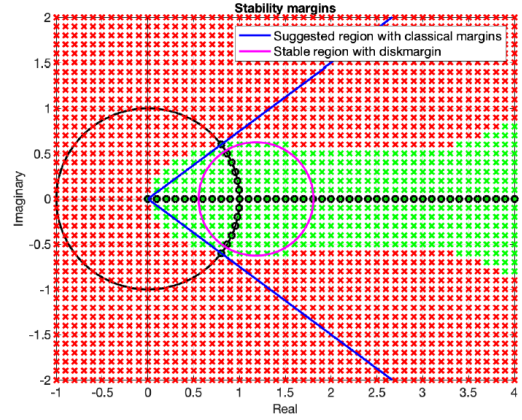
margin approach is used to evaluate the different parameters influence on the margins. It can here be concluded that changes on the discharge and area curves for the spool sleeve valve system can equalize the performance of the two sides, which is essential for the steering feeling.

REFERENCES

[1] Solutions, Danfoss Power. "OSPB/C/F/D/L LS, OLS Priority Valves, OSQ Flow Amplifiers." Technical datasheet -. Danfoss, Nordborg, Denmark. 2019. URL http://hdl.handle.net/-.

[2] Markus Nikolaj Friis, Mikkel Almind Jensen. "Development, validation and optimization of a new steering concept." Master's Thesis, Aalborg University, Aalborg, Nordjylland. 2021.

[3] Peter Seiler, Andrew Packard and Gahinet, Pascal. "An Introduction to Disk Margins." *IEEE Control Systems* .