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On Displacement Controlled Drive Networks

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Abstract.

In the efforts to improve the energy efficiency of hydraulic systems, a key initiative may be to eliminate the throttle losses induced by hydraulic control valves. Existing solutions targeting this aspect include standalone electro-hydraulic variable-speed drives for control of hydraulic actuators such as cylinders and motors. A main challenge of this approach may be the electric component sizes, hence their cost, required to meet both the maximum force/torque and speed of a cylinder/motor, even though not concurrently. In case of systems with multiple hydraulic cylinders/motors to be actuated, the idea of realizing highly integrated systemwide drives in terms of so-called electro-hydraulic variable-speed drive networks have emerged in recent years. As opposed to standalone electro-hydraulic variable-speed drives, such drive networks enable power sharing not only on the electric side, but also on the hydraulic side. A consequence of this is that components may be downsized considerably in many applications, while maintaining a similar efficiency. However, the cost may still be prohibitively high in many industry segments, especially due to the number of electric machines and drives. This provides the incitement to investigate hydraulic drive networks enabling control via variable displacement machines rather than via multiple variable-speed electric machines. This approach may be feasible due to the presumed less cost of variable displacement machines compared to electric machines/drives. This paper analyses the feasibility of hydraulic drive networks using a single electric machine powering multiple variable displacement machines, organized in network configurations. Especially, emphasis is placed on the component sizing in such displacement-controlled drive networks when applied in a crane application. Furthermore, the resulting displacement-controlled drive networks are compared to existing individual displacement-controlled drive approaches, where it is found that significant reductions in variable displacement machine sizes may be enabled by drive network structures.

Keywords. Displacement-controlled Drive Networks, Displacement-controlled Drives Electro-hydraulic Drive Networks, Energy Efficiency, Crane Actuation, Component Sizing.

1. INTRODUCTION

Facing the current climate crisis, efforts to reduce carbon emissions are pursued throughout societies, including various industries. In this endeavour, machinery relying on hydraulic systems as means of power transmission is especially relevant to consider. This is partly because actuation in such machinery commonly is associated with large forces and power, and partly because traditional hydraulic systems rely on throttle valves to be controlled. That is, relying on control via loss adjustment. Consequently, traditional hydraulic machinery is inherently energy inefficient and associated with substantial energy losses and related emissions [1]. Hydraulic machinery may be stationary or mobile, and in the former case these are often already electrically powered, whereas the latter case remains powered by internal combustion engines in most cases. Trends suggests, however, that mobile hydraulic machinery are increasingly electrified by replacing internal combustion engines with battery powered electric machines. To enable an actual decarbonization of such machinery, the electric energy used must be provided by renewable energy sources, and furthermore, preferably only use as much energy as needed to minimize stress on the electric grid in a world where electrification of society is a key priority. Here, the energy inefficiency of traditional hydraulic systems poses a great challenge due to their high energy losses. As a first approach to reduce losses, variable-speed fixed or variable displacement pumps may be applied, allowing to reduce energy losses of hydraulic power supplies by adjusting shaft speeds to flow requirements, more optimal operating points etc. [2,3]. However, a substantial part of the losses in traditional hydraulic systems are the principal losses associated with throttle control, and to enable a real impact in terms of reduced energy consumption and emissions, these principal losses must be significantly reduced.

Approaches to achieve such reductions in energy consumption involves avoiding or limiting the use of control valves, and here electro-hydraulic standalone drives for actuation of e.g. hydraulic cylinders may be applied [4-8]. In systems containing two or more cylinders, however, it may be more feasible to consider more dedicated multi-axis electro-hydraulic drives such as the so-called HHEA [9,10] or so-called electro-hydraulic variable-speed drive networks [11-15]. Even though these electro-hydraulic systems suggest high energy efficiencies, the use of one or more electric machines per cylinder or motor to be actuated, may cause drive cost to be prohibitively high in an industrial setting. An approach to reduce the number of required electric machines may be so-called displacement control, where the control is shifted from speed variability to displacement variability. A recent displacement control approach suggests this to be combined with traditional valves for low power functions [16,17]. The topic of displacement-controlled drives has been considered extensively, see e.g. [18-21]. However, a one-pump per actuator requirement may also here cause cost levels to be prohibitively high [21]. To overcome this challenge the integration of on/off valves was proposed in [22], allowing to control individual actuators in a sequential way [21]. A drawback of this approach, however, is that concurrent control of all actuators is not possible.

An alternative approach mitigating the abovementioned challenges with displacement control, may be to organize multiple variable pumps on a common shaft in network configurations, similar to electro-hydraulic variable-speed drive networks. That is, having the same offset with a common shaft with multiple pumps as e.g. [18-22], but implement *appropriate* short circuit connections between cylinder chambers and *appropriate* hydraulic connections between pump and actuator ports in a network kind of way, forming *displacement-controlled drive networks* (DCDNs). This paper presents an analysis of displacement-controlled drive networks and their design in the context of a crane application. It is shown that the total required pump displacement may be significantly reduced compared to *conventional* displacement-controlled multi-actuator drive systems, while allowing concurrent control of all actuators. The nomenclature used throughout the paper is explained in the Appendix.

2. CRANE APPLICATION & DESIGN REQUIREMENTS

The drive designs considered in the following are based on the load imposed by the two-axis knuckle boom crane depicted in Figure 1.1. Any drive system considered must enable a payload mass of up to $m_L = 5000$ [kg], moving at 200 [mm/s] both vertically and horizontally throughout the operating space, i.e. $\dot{x}_L = \pm\,200$ [mm/s], $\dot{y}_L = \pm\,200$ [mm/s]. The following serves as a case study the load and speed requirements have been chosen to provide reasonable flow and pressure levels and are not related to specific application tasks.

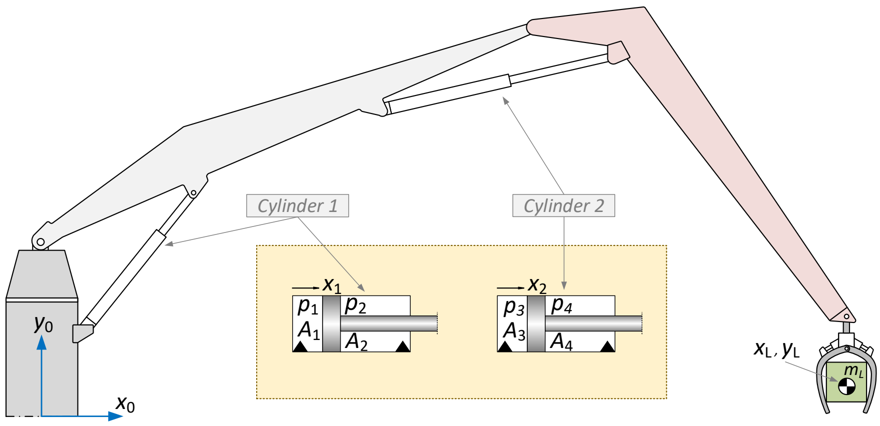

Figure 1.1. Two-axis knuckle boom crane application for investigations.

The hydraulic piston forces for the two cylinders are given by Eqs. (1.1).

$$F_{L1} = A_1 p_1 - A_2 p_2 \quad , \qquad F_{L2} = A_3 p_3 - A_4 p_4 \tag{1.1}$$

Using the crane parameters considered in [11] and a payload of 5000 [kg], the piston forces appear as depicted in Fig. 1.2. when considering the operating space spanned by the cylinder pistons.

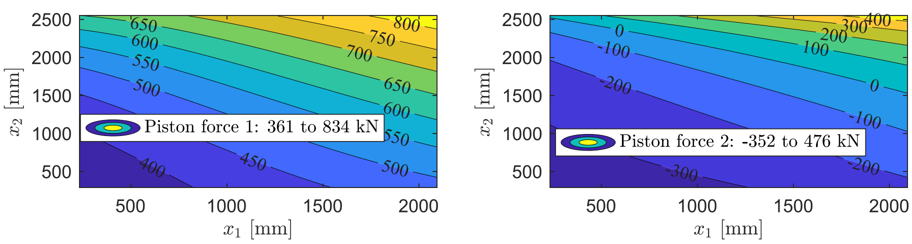

Figure 1.2. Cylinder forces ([kN]) of knuckle boom crane at payload of 5000 [kg].

The piston speeds evaluated at these extreme payload speeds, $\dot{x}_L = \pm$ 200 [mm/s], $\dot{y}_L = \pm$ 200 [mm/s], are depicted in Fig. 1.3., accounting for the gearings induced by the crane mechanism itself.

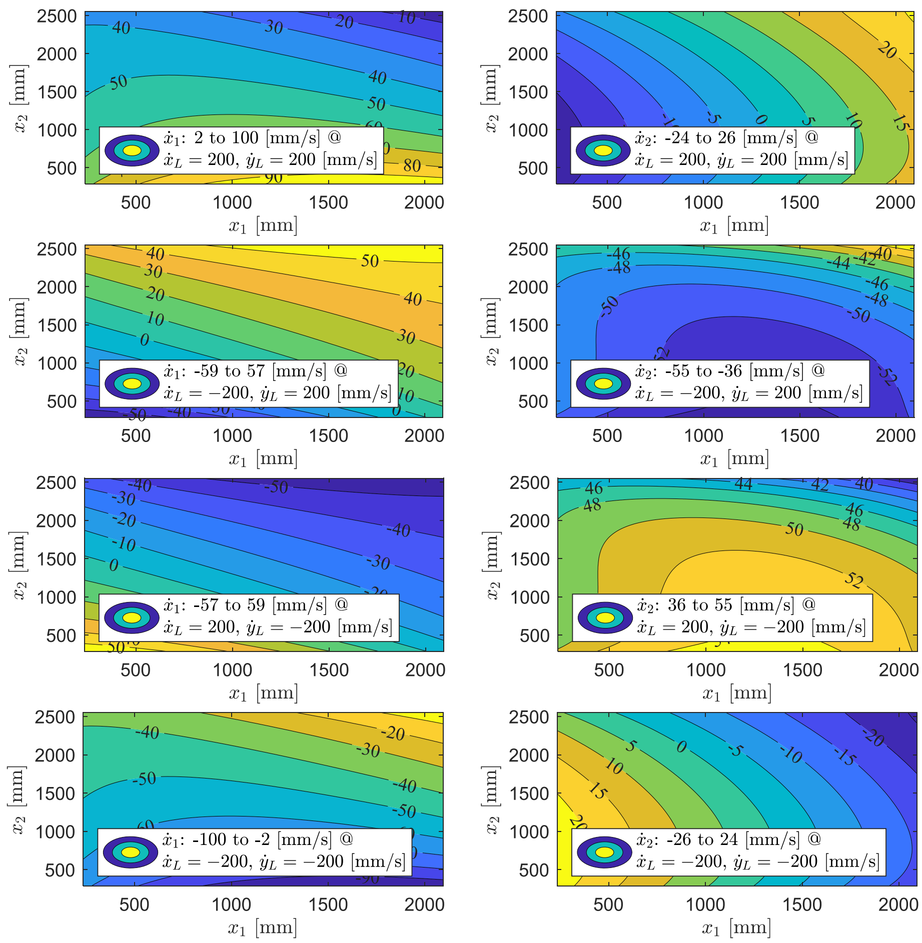

Figure 1.3. Cylinder piston speed at max./min. payload speeds and piston positions.

The piston forces and piston velocities depicted in Figures 1.2. and 1.3. serves as design requirements in the following sections.

Note: The following disregards the pilot supplies for the swivel control mechanisms and other flow losses as well as torque losses. Furthermore, all drives considered are assumed to operate at a constant shaft speed of 3000 [rpm].

3. INDIVIDUAL DISPLACEMENT CONTROLLED DRIVES

As mentioned in the introduction, individual displacement-controlled (IDC) drives have been proposed in several versions and for several applications [18-22]. Here the case of a multi-actuator IDC drive without switching valves are considered as a benchmark for displacement-controlled drive networks (DCDNs), as this allows for individual control of both cylinder pistons simultaneously, which is also the objective for the DCDNs. For the crane application the dual cylinder IDC drive depicted in Fig. 2.1 is considered.

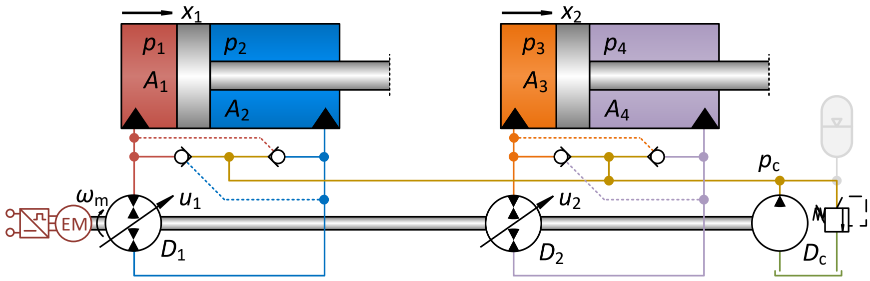

Figure 2.1. Individual displacement-controlled drive for crane application.

In literature IDC drives are generally equipped with an accumulator attached to the charge pump outlet, forming a charge line having the purpose of compensating the cylinder differential flows, flow losses and fluid exchange related to thermal management and filtering, and to supply the swivel control valves [22]. Hence, if no fluid exchange functionality is implemented as is the case in Fig. 2.1, the charge pump displacement size may be rather small. On the other hand, if no accumulator is desired, the charge pump must be sized to handle the cylinder differential flows (and in practice also flow losses and pilot flows). Considering traditional transmission pumps, these are equipped with a build-in charge pump. From these considerations, three different cases may be established.

Case 1: IDC without accumulator, but with charge pump

Case 2: IDC with accumulator and charge pump

Case 3: IDC with accumulator, but without charge pump (utopia)

Here Case 1 and Case 2 may be applied in practical use, and Case 2 is a common implementation. Case 3 is highly idealized and could work only if flow losses were absent and is thus a utopia. However, this case serves as an idealized benchmark. The theoretically required displacement sizes for all three cases are considered in the following. It should be noted that for all three cases the maximum steady state pressure here is 190 [bar].

3.1. Case 1 sizing – IDC with Charge Pump & No Accumulator

Considering Fig. 2.1 and the Case 1 conditions, the displacement machine flows in a steady state are given by Eqs. (3.1), (3.2).

$$Q_1 = D_1 u_1 \omega_\mathrm{m} = \begin{cases} A_1 \dot{x}_1, & p_1 \geq p_2 \\ A_2 \dot{x}_1, & p_1 < p_2 \end{cases} \quad , \quad Q_2 = D_2 u_2 \omega_\mathrm{m} = \begin{cases} A_3 \dot{x}_2, & p_3 \geq p_4 \\ A_4 \dot{x}_2, & p_3 < p_4 \end{cases} \tag{3.1}$$

$$Q_\mathrm{c} = D_\mathrm{c} \omega_\mathrm{m} = (A_1 - A_2)\dot{x}_1 + (A_3 - A_4)\dot{x}_2 \tag{3.2}$$

From the cylinder piston forces and speed ranges of Figures 1.2 and 1.3, the required individual displacements and the total displacement are obtained as Eqs. (3.3).

$$D_1 = 123\ [\mathrm{ccm}]\,,\ D_2 = 48\ [\mathrm{ccm}]\,,\ D_\mathrm{c} = 70\ [\mathrm{ccm}] \rightarrow D_\mathrm{tot} = 241\ [\mathrm{ccm}] \tag{3.3}$$

3.2. *Case 2 sizing – IDC with Charge Pump & Accumulator*

Due to the application of both a charge pump and an accumulator, the charge pump could be chosen to only compensate flow losses (and in reality, also the swivel control valve pilot flow) at the specified shaft speed. Here, the sizing takes offset in charge pump sizing applied in the Bosch Rexroth A4VG series [23], with relevant information outlined in Table 1.

Table 1: Charge pump vs. nominal pump sizes for A4VG [23].

A4VG	**28**	**40**	**56**	**71**	**90**	**125**
Nominal size in [ccm]	28	40	56	71	90	125
Charge pump size [ccm]	6.1	8.6	11.6	19.6	19.6	28.3
Charge pump size in % of nominal size	22	22	21	28	22	24

For the A4VG [23], the charge pumps are averagely sized at 23% of the nominal pump size. To be conservative (sizing in favour of Case 2), the charge pump size is here assumed to be 20% of the total nominal pump size required. In this case, the displacement sizes appear as Eqs. (3.4).

$$D_1 = 123\ [\mathrm{ccm}]\,,\ D_2 = 48\ [\mathrm{ccm}]\,,\ D_\mathrm{c} = \tfrac{D_1+D_2}{5} = 34\ [\mathrm{ccm}] \rightarrow D_\mathrm{tot} = 205\ [\mathrm{ccm}] \tag{3.4}$$

3.3. *Case 3 sizing – IDC without Charge Pump*

As Case 3 assumes that no charge pump is required, total displacement simply corresponds to the sum of the nominal displacement, also required in Case 1 and Case 2, and is hence given by Eqs. (3.5).

$$D_1 = 123\ [\mathrm{ccm}]\,,\ D_2 = 48\ [\mathrm{ccm}]\,,\ D_\mathrm{c} = 0\ [\mathrm{ccm}] \rightarrow D_\mathrm{tot} = 171\ [\mathrm{ccm}] \tag{3.5}$$

3.4. *IDC Shaft Torque Requirements*

In evaluating the resulting torque applied to the common drive shaft, the chamber pressures are estimated as follows with cylinder forces $F_\mathrm{L1s} = (A_1 - A_2)p_\mathrm{c}$, $F_\mathrm{L2s} = (A_3 - A_4)p_\mathrm{c}$.

$$p_1 = \begin{cases} \dfrac{A_2 p_c + F_{L1}}{A_1} & , \ F_\mathrm{L1s} < F_\mathrm{L1} \\ p_c & , \ F_\mathrm{L1s} \geq F_\mathrm{L1} \end{cases} \quad , \quad p_2 = \begin{cases} p_c & , \ F_{L1s} < F_{L1} \\ \dfrac{A_1 p_c - F_\mathrm{L1}}{A_2} & , \ F_{L1s} \geq F_{L1} \end{cases} \tag{3.6}$$

$$p_3 = \begin{cases} \dfrac{A_4 p_c + F_\mathrm{L2}}{A_3} & , \ F_{\mathrm{L2}s} < F_\mathrm{L2} \\ p_c & , \ F_{\mathrm{L2}s} \geq F_\mathrm{L2} \end{cases} \quad , \quad p_4 = \begin{cases} p_c & , \ F_{\mathrm{L2}s} < F_\mathrm{L2} \\ \dfrac{A_3 p_c - F_\mathrm{L2}}{A_4} & , \ F_{\mathrm{L2}s} \geq F_\mathrm{L2} \end{cases} \tag{3.7}$$

The common shaft torque for the IDC drives is then given by Eq. (3.8).

$$\tau_\mathrm{IDC} = D_1 u_1 (p_1 - p_2) + D_2 u_2 (p_3 - p_4) + D_\mathrm{c} p_\mathrm{c} \tag{3.8}$$

Here, the factors $D_1 u_1$ and $D_2 u_2$ are obtained using Eqs. (3.1) and Eqs. (3.9), noting that the shaft speed ω_m is constant.

$$D_1 u_1 = \frac{Q_1}{\omega_\mathrm{m}} \quad , \quad D_2 u_2 = \frac{Q_2}{\omega_\mathrm{m}} \tag{3.9}$$

Evaluating Eq. (3.8) with Eqs. (3.9) for the cylinder force maps Fig. 1.2. and the cylinder velocity maps Fig. 1.3., the maximum shaft torque for the three cases is given by Eqs. (3.10), (3.11) and (3.12).

$$\text{Case 1: } \tau_\mathrm{IDC}|_\mathrm{max} = 139 \text{ [Nm]} \tag{3.10}$$

$$\text{Case 2: } \tau_\mathrm{IDC}|_\mathrm{max} = 127 \text{ [Nm]} \tag{3.11}$$

$$\text{Case 3: } \tau_\mathrm{IDC}|_\mathrm{max} = 116 \text{ [Nm]} \tag{3.12}$$

The reason for the reduced maximum shaft torque of benchmark case 2 and 3 compared to benchmark case 1 is attributed the reduced charge pump sizes.

4. Displacement Controlled Drive Networks

Displacement controlled drive networks (DCDNs) for dual cylinder systems may basically appear as depicted in Fig. 4.1. It is desired to use as few displacement machines as possible, while maintaining controllability of both cylinder pistons and the system sum pressure as in [11]. Hence, a single short circuit connection between two chambers should be deployed, necessitating three displacement machines to achieve the desired level of controllability. This, in turn, provides 36 unique ways of realizing the drive connections, which should be considered in context of the design requirements discussed in Section 2.

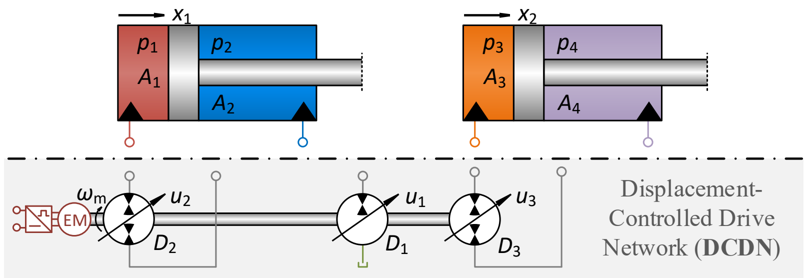

Figure 4.1. Displacement-controlled drive network (DCDN) for crane application.

4.1. DCDN Sizing Example

As an example on component sizing of one of the 36 unique configurations, consider the DCDN in Fig. 4.2. Here, the rod-side chamber of cylinder 1 is short circuited with the bore side chamber of cylinder 2, forming the combined volume *23*. Furthermore, the bore side chamber of cylinder 1 is connected to the common volume 23 via pump 2 and volume 23 is also connected to the rod side volume of cylinder 2 via pump 3 (notation 1 ↔ 23 ↔ 4). Finally, the tank volume 0 is connected to volume 23 via pump 1 (notation 0 ↔ 23).

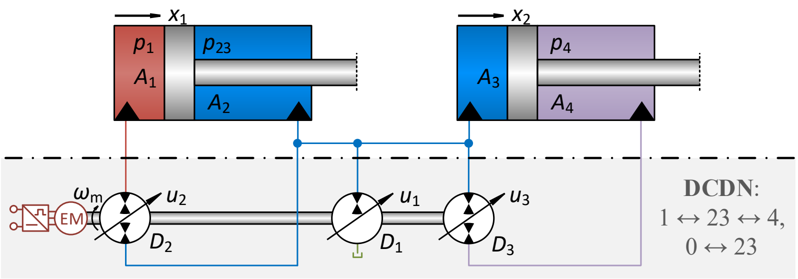

Figure 4.2. Displacement-controlled drive network 1 ↔ 23 ↔ 4, 0 ↔ 23.

The displacement machine flows in a steady state, assuming flow losses absent, are obtained considering flow continuity and are given by Eqs. (4.1)-(4.3).

$$Q_1 = D_1 u_1 \omega_m = (A_1 - A_2)\dot{x}_1 + (A_3 - A_4)\dot{x}_2 \tag{4.1}$$

$$Q_2 = D_2 u_2 \omega_m = A_1 \dot{x}_1 \tag{4.2}$$

$$Q_3 = D_3 u_3 \omega_m = A_4 \dot{x}_2 \tag{4.3}$$

When considering the cylinder piston force and speed ranges in Figures 1.2. and 1.3., the required displacement sizes in Eq. (4.4) are obtained.

$$D_1 = 70\ [\text{ccm}]\,,\ D_2 = 123\ [\text{ccm}]\,,\ D_3 = 26\ [\text{ccm}] \rightarrow D_{\text{tot}} = 219\ [\text{ccm}] \tag{4.4}$$

This is comparable to the IDC drive, Case 1, as no accumulator is used. It is notable that total required displacement for the IDC of Case 1 is 241 [ccm], whereas it is only 219 [ccm] for the DCDN example considered here.

4.2. Sizing Results for all 36 DCDNs

Applying the same sizing method for all 36 unique DCDNs, the total required displacements presented in Table 2 are obtained. It should be noted that the maximum steady state pressure depends on the short circuit applied, hence this is also specified in Table 2. The DCDNs requiring the lowest and highest displacement are depicted in Fig. 4.3 to the right and to the left, respectively.

From the sizing results in Table 2 it is evident that, even though requiring 9% less displacement compared to the IDC in Case 1, the DCDN example of section 4.1 is not the one requiring the lowest amount of displacement.

The sizing results emphasize the importance of architecture selection, and its influence on required displacement when considered for a certain application. The DCDN denoted 1 ↔ 23 ↔ 4, 0 ↔ 1 is the architecture that requires the lowest amount of displacement, 155 [ccm], whereas the DCDN denoted 13 ↔ 4 ↔ 2, 0 ↔ 2 requires the highest displacement with 337 [ccm] which is more than twice that of DCDN 1 ↔ 23 ↔ 4, 0 ↔ 1.

Table 2: Sizing results for all 36 unique DCDNs.

Configuration	Max. pressure	0 ↔ 13	0 ↔ 2	0 ↔ 4
13 ↔ 2 ↔ 4	294 [bar]	163 [ccm]	234 [ccm]	294 [ccm]
13 ↔ 4 ↔ 2	294 [bar]	197 [ccm]	**337 [ccm]**	268 [ccm]
2 ↔ 13 ↔ 4	294 [bar]	156 [ccm]	225 [ccm]	216 [ccm]
Configuration	Max. pressure	0 ↔ 14	0 ↔ 2	0 ↔ 3
14 ↔ 2 ↔ 3	269 [bar]	199 [ccm]	244 [ccm]	251 [ccm]
14 ↔ 3 ↔ 2	269 [bar]	205 [ccm]	319 [ccm]	250 [ccm]
2 ↔ 14 ↔ 3	269 [bar]	184 [ccm]	253 [ccm]	191 [ccm]
Configuration	Max. pressure	0 ↔ 23	0 ↔ 1	0 ↔ 4
23 ↔ 1 ↔ 4	190 [bar]	216 [ccm]	171 [ccm]	231 [ccm]
23 ↔ 4 ↔ 1	190 [bar]	313 [ccm]	204 [ccm]	268 [ccm]
1 ↔ 23 ↔ 4	190 [bar]	219 [ccm]	**155 [ccm]**	279 [ccm]
Configuration	Max. pressure	0 ↔ 24	0 ↔ 1	0 ↔ 3
24 ↔ 1 ↔ 3	190 [bar]	262 [ccm]	191 [ccm]	198 [ccm]
24 ↔ 3 ↔ 1	190 [bar]	331 [ccm]	196 [ccm]	260 [ccm]
1 ↔ 24 ↔ 3	190 [bar]	247 [ccm]	183 [ccm]	254 [ccm]

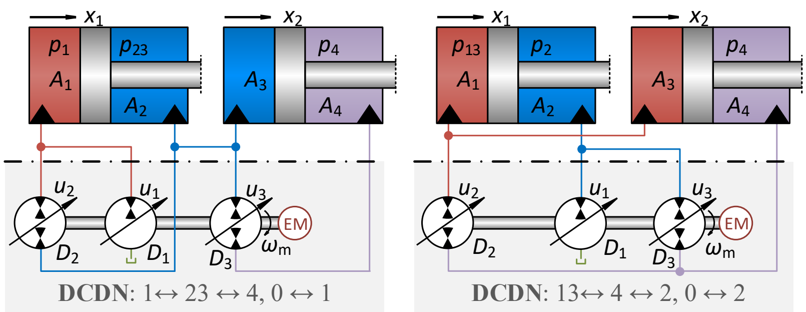

Figure 4.3. DCDNs with min. (left) and max. (right) required total displacement.

4.1. DCDN Shaft Torque Requirements

Numbering the 36 unique DCDN architectures from $i = 1 \dots 36$, the instantaneous power of the i^{th} DCDN architecture is given by Eq. (4.5) where $\tau_{\mathrm{DCDC},i}$ is the total torque acting on the common shaft.

$$\tau_{\mathrm{DCDC},i}\omega_{\mathrm{m}} = \left(D_{1,i}u_{1,i}\Delta p_{1,i} + D_2 u_2 \Delta p_{2,i} + D_{3,i}u_{3,i}\Delta p_{3,i}\right)\omega_{\mathrm{m}} \tag{4.5}$$

Indeed, when the shaft speed ω_{m} is constant, the swivel angles $u_{1,i}$, $u_{2,i}$, $u_{3,i}$ must inevitably be adjusted such that the total shaft torque of any of the 36 DCDNs is identical for any identical operating point and given by Eq. (4.6), whereas the individual displacement machine torques may differ among the different DCDNs.

$$\tau_{\mathrm{DCDC},i} = D_{1,i}u_{1,i}\Delta p_{1,i} + D_2 u_2 \Delta p_{2,i} + D_{3,i}u_{3,i}\Delta p_{3,i} \tag{4.6}$$

The instantaneous displacements may be obtained as Eq. (4.7).

$$D_{1,i}u_{1,i} = \frac{Q_{1,i}}{\omega_{\mathrm{m}}} \quad , \quad D_{2,i}u_{2,i} = \frac{Q_{2,i}}{\omega_{\mathrm{m}}} \quad , \quad D_{3,i}u_{3,i} = \frac{Q_{2,i}}{\omega_{\mathrm{m}}} \tag{4.7}$$

Evaluating Eq. (4.6) with Eqs. (4.7) for the cylinder force and velocity maps in Fig. 1.2. and Fig. 1.3, the maximum shaft torque for all 36 DCDNs is given by Eq. (4.8).

$$\tau_{\mathrm{DCDN}}|_{\max} = 139 \text{ [Nm]} \tag{4.8}$$

This is coincident with the IDC Case 1 and is a consequence of not including any hydraulic energy recuperation abilities, the specified shaft speed and not considering torque losses.

5. Comparison of Required Displacements

To provide an overview of what can be gained in terms of component downsizing using a DCDN compared to IDC drives, all sizing results are depicted in Figure 5.1 for comparison.

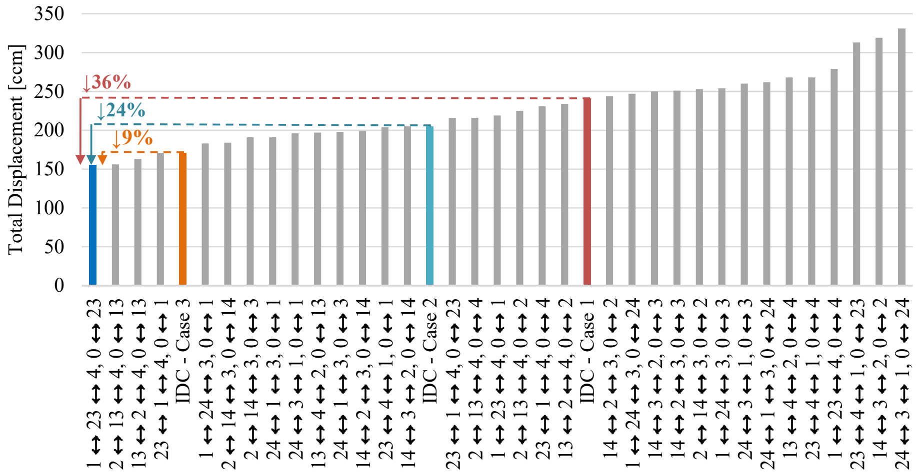

Figure 5.1. Ideal DCDN and IDC drive sizing results for the crane application.

Considering IDC Case 1, which is the case most comparable with the DCDNs as no accumulators are used, the DCDN subject to the lowest amount of displacement allows a total displacement reduction of 36%. Compared to the more conventional IDC Case 2, the same DCDN allow for a displacement reduction of 24%, and 9% when compared to the utopian IDC Case 3. The highest DCDN displacement requirement, on the other hand, by far exceed those of all IDC cases. These results emphasize the downsizing perspectives of DCDNs compared IDCs implying that cost may be significantly reduced.

6. SUMMARY & OUTLOOK

This paper considers the possible feasibility of displacement-controlled drive networks when compared to individual displacement-controlled drives applied to a dual axis crane application. The presented study takes offset in idealized cases, not considering losses such as torque and flow losses. Special attention is given to displacement requirements, and the possibility to reduce installed displacement using displacement-controlled drive networks. As benchmarks to the displacement-controlled drive networks, three cases for individual displacement-controlled drives are considered, namely being a case with a charge pump but no accumulator, a case with both a charge pump and an accumulator and a utopian case with no charge pump but with an accumulator. For the crane application considered, the use of a displacement-controlled drive network allows for a reduction in required displacement of 36% compared to the individual displacement-controlled drive without accumulator and a reduction of 24% compared to the individual displacement-controlled drive with accumulator. These results suggest that the cost of using displacement-control may be significantly reduced compared to earlier reports in literature, potentially rendering such drives commercially feasible.

In the endeavour to further develop displacement-controlled drive network technology, multiple topics should be addressed. The work presented in this paper takes offset in the static loads of the chosen crane application, however, future design developments should also consider the impact of transient loads. Especially, loads resulting from acceleration and rapid motion reversal may significantly influence the common shaft torque as well as the flow balance of a displacement-controlled drive network, and really any displacement-controlled drive system, and should be taken into consideration.

From a cost point of view, it is preferred to use as small component sizes as possible. However, transient loads may cause flow and torque/current saturation preventing a system to perform as intended, and hence peak demand conditions should be considered with great care. Such challenges may be mitigated by the system level controls via the pressure and flow coupling introduced in displacement-controlled drive networks, allowing to distribute hydraulic energy in alternative ways compared to individual controlled drive networks. Furthermore, the bandwidths of displacement control mechanisms and electric machine/inverter drive assemblies should be considered to ensure that these allow the system level controls to maintain proper behaviour of a system, both under rapid changes in motion commands and as well as under impact loads.

To this end, the use of charge pumps in displacement-controlled drive networks may also have a perspective and should be considered in future research. This approach is not considered in the work presented in this paper, where the use of three variable displacement machines with two cylinders allows the control of the lower pressure level in the system. A further advantage of the presented approach is, that excess energy of a fixed charge pump is not dissipated in the tank, hence the energy losses are anticipated to be less compared to individual displacement-controlled drives. This, however, depends on the energy consumption of the additional swash plate control mechanism introduced with the network solutions. Other losses introduced in displacement-controlled drive networks as well as individual displacement-controlled drives, are associated with idle conditions when the cylinders are inactive. These idle losses may be mitigated by adjusting the shaft speed command to the instantaneous flow command.

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APPENDIX

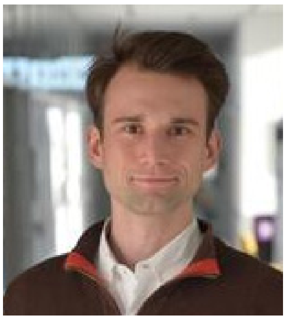
The nomenclature used in this paper is outlined in Table 3 below.

Table 3: Nomenclature used.

Symbol	Description
A_1, A_2, A_3, A_4	Cylinder piston areas
D_1, D_2, D_3	Max. displacement of displacement machine
D_c	Displacement of charge pump
$F_{\mathrm{L}1}, F_{\mathrm{L}2}$	Piston pressure forces
m_L	Crane payload
p_1, p_2, p_3, p_4	Cylinder piston pressures
p_c	Charge pressure
$\Delta p_1, \Delta p_2, \Delta p_3$	Displacement machine pressure differences
Q_1, Q_2, Q_3	Displacement machine flows
Q_c	Charge pump flow
u_1, u_2, u_3	Normalized swivel angles
$\dot{x}_L, \dot{y}_L$	Payload speed components
$\dot{x}_1, \dot{x}_2$	Cylinder piston speeds
ω_m	Shaft speed
$\tau_\mathrm{IDC}, \tau_\mathrm{DCDC}$	Common shaft torques

Biographies

Lasse Schmidt received the M.Sc. degree in engineering (mechatronics) from Aalborg University, Denmark, in 2008. From 2008 he was with the application engineering group of Bosch Rexroth A/S, Denmark, and from 2010 an industrial Ph.D. fellow also associated with Aalborg University. He received the Ph.D. degree in robust control of hydraulic cylinder drives in 2014. Subsequently, he has been a postdoctoral researcher at AAU Energy while concurrently being with Bosch Rexroth AG. Hereafter he became an Assistant Professor with AAU Energy. He is currently an Associate Professor with AAU Energy and heading research activities related to electro-hydraulic drive network technology, a field in which he is the founder of the fundamental design and control principles. He is the main author or co-author of nearly 70 scientific peer-reviewed publications, most of them on topics related hydraulic drives and systems control. Lasses current research interests are in design and control of electro-hydraulic drive networks and their integration into both mobile working machines and industrial systems.



Mikkel van Binsbergen-Galán received the B.Sc. and M.Sc. degrees, both from Aalborg University, Denmark, in 2020 and 2022 respectively, where he is currently working towards the Ph.D. degree at AAU Energy. His research interests are related to design and control of electro-hydraulic drives, actuators and systems, with special focus on electro-hydraulic variable-speed drive technology.