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Using input shaping and pressure feedback to suppress oscillations in slewing motion of lightweight flexible hydraulic crane

Magnus B. Kjelland* and Michael R. Hansen

Faculty of Engineering and Science, Department of Engineering Sciences, University of Agder, Jon Lilletunsvei 9, 4879 Grimstad, Norway

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This paper presents a method to actively reduce vibrations in the flexible mechanical structure of a hydraulically actuated vehicle loader crane. Based on information on the natural frequency and damping of a simplified model of the crane an input shaping scheme is set up to control the proportional valve leading to substantially reductions in oscillations. The method is compared and combined with a pressure feedback control of the proportional valve that actively suppresses transient variations in pressure. A full scale vehicle loader crane is used in the experimental verification of the method, and the motion of the tool point of the crane is measured by a high precision laser tracker. The results shown in this paper demonstrate that input shaping and pressure feedback are useful tools to minimize vibrations in hydraulically actuated flexible structures.

Keywords: pressure feedback; input shaping; flexible manipulator; vibration reduction

1. Introduction

Hydraulic actuation is extensively used in controlling mobile cranes. The vehicle loader crane is a well-developed, weight optimized product, used in many areas due to its low weight, large reach and high payload capacity. Using hydraulics has some advantages regarding compactness and reliability, but can also cause undesired dynamic effects, leading to vibrations. Such oscillatory behavior combined with a long flexible mechanical structure, like the vehicle loader crane, is detrimental to efficient and safe operation.

Work regarding control of vehicle loader crane often includes tool point control, Ebbesen *et al.* (2006) and Pedersen *et al.* (2010). The performance of tool point control is influenced by structural vibrations caused in the crane. Some efforts to reduce this effect can be seen in Nordhammer *et al.* (2012a) and Nordhammer *et al.* (2012b).

A successful approach to reducing unwanted dynamics in machinery with low damping and low eigenfrequencies is called input shaping. Input shaping is an open loop control technique used to filter reference commands in such a way that they cancel out their own motion induced oscillations, Sorensen *et al.* (2010).

Input shaping is particularly effective for lightly damped oscillations in low frequency systems, such as overhead cranes with the payload suspended by wires, see Singhose and Seering (2011). There are no example of applying the method to hydraulically actuated boom cranes, although, recently Moon (2012) used input shaping on a pneumatically actuated and downscaled excavator arm.

Cranes that are hydraulically actuated often includes components such as counterbalance valves and pressure compensated flow valves. It is reported on several occasions that they tend to introduce oscillations and even instability in hydraulic systems, Miyakawa (1978), Persson *et al.* (1989), Zähe (1995) and Handros *et al.* (1993). These effects cannot be suppressed by input shaping.

A technique to avoid these vibrations is to employ some kind of pressure feedback which basically works by filtering the pressure in the controlled actuator, and then subtracting this from the command signal, see Hansen and Andersen (2010). The goal is to remove or suppress high frequency variations in the pressure without counteracting the steady state variations required to move a payload in a prescribed manner. An example of this approach was described recently by Cristofori *et al.* (2012) where both high filtered pressure and the pressure gradient where used successfully to suppress oscillations in a boom crane.

This article presents how vibrations due to flexibility and hydraulic effects can be reduced using input shaping and pressure feedback in the control loop on the slewing motion of a hydraulically actuated full scale vehicle loader crane. The main purpose of the work is to introduce input shaping and pressure feedback to a cranes slewing system and to set up guidelines for their application in that context.

*Corresponding author. Email: magnus.b.kjelland@uia.no

2. Considered system

The crane used in this paper is a standard vehicle loader crane with a maximum reach of 12.5 m. It has a payload capacity of 1250 kg at full reach.

The crane has four degrees of freedom, i.e., slewing, main boom, knuckle boom, and telescope, noted in Figure 1 as q_0, q_1, q_2 and q_3, respectively.

A real-time I/O system is used to control the hydraulic valves on the crane. The control system can record sensor information from all the position and pressure sensors mounted on the crane. In this paper, only the slewing function of the crane is actively controlled. The other three degrees of freedom will be in a fixed position.

The hydraulic circuit of the slewing motion is shown in Figure 2. The mechanical system consists of a rack and pinion with the latter rigidly connected to the tower of the crane.

The main hydraulic components include the double acting piston, two counterbalance valves and the pressure compensated directional control valve, which is connected to the pressure and tank line, noted as P and T. The counter balance valves are identical and have the same pilot area ratio.

The crane is considered as a long flexible link with high inertia. Given that it is actuated by a piston suspended by two oil volumes that act like springs in parallel, there are bound to be vibrations when slewing the crane. There are also two counterbalance valves that tend to introduce vibrations is these types of systems. To demonstrate this effect the crane was given an input and the tool point velocity was recorded during the motion.

Figure 3 shows three different reference signals for the slewing motion. The command signal for the motion is divided into three sections: ramp up, constant speed, and ramp down.

A laser tracker measures the global X, Y and Z coordinate of the tool point of the crane and these signals are differentiated to yield the tool-point velocities. In Figure 4 the absolute velocity is shown.

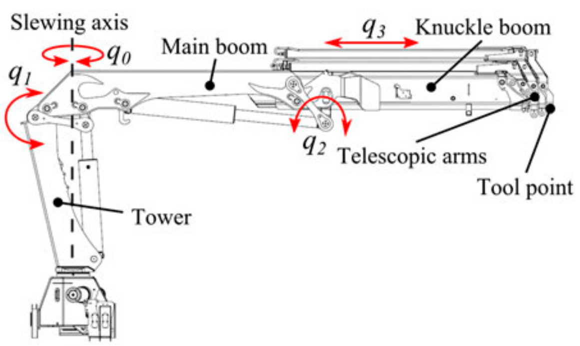

Figure 1. Hydraulic loader crane showing the degrees of freedom.

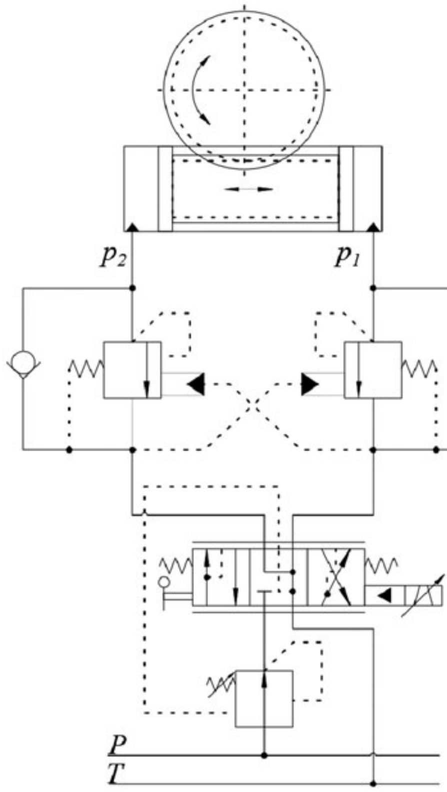

Figure 2. System model circuit.

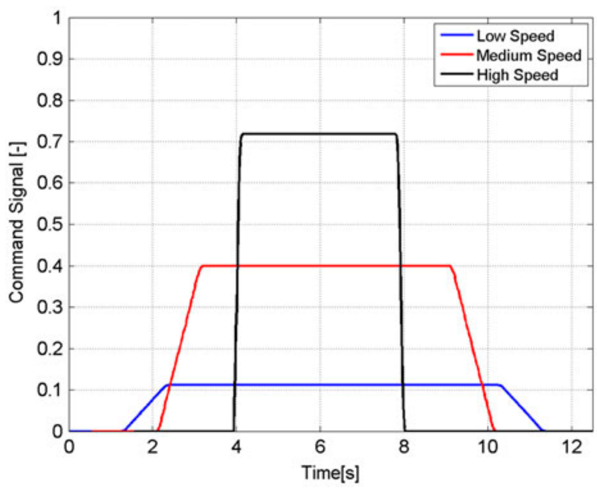

Figure 3. Ramped command signal.

2.1. System model

When setting up the input shaping, the nonlinear system of the hydraulically actuated crane can be simplified to a second order transfer function, see (1), with the system natural frequency and damping denoted as ω_s and ζ_s, respectively. Because the valve is pressure compensated, the speed v_{tp} of the tool point is proportional to the input signal u, which is the dimensional valve opening, ranging from $-1 \ldots 0 \ldots 1$, see (2).

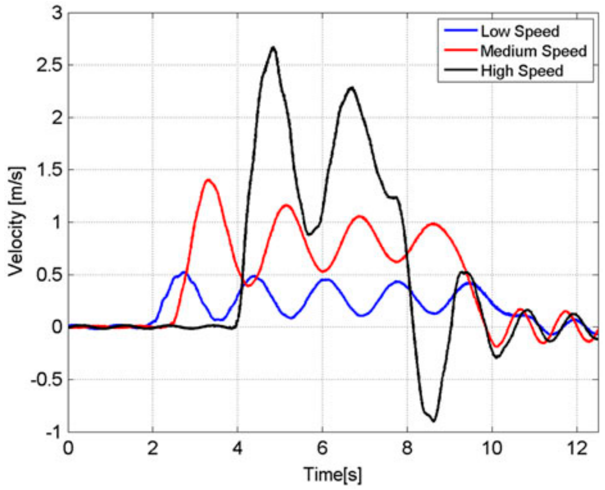

Figure 4. Tool point velocity without any control.

$$G_s(s) = K_s \frac{\omega_s^2}{s^2 + \zeta_s \omega_s s + \omega_s^2} \tag{1}$$

$$v_{tp}(s) = u(s) \cdot G_s(s) \tag{2}$$

The systems natural frequency and damping was found using a step input on the experimental crane. A transfer function was estimated to fit the response, shown in Figure 5.

The experimental verified natural frequency and damping ratio is shown in Table 1.

3. Pressure feedback

Work done by Hansen and Andersen (2010) shows implementation of a lead compensation to the control effort u can suppress transient vibrations in these types of circuits:

$$u^* = u - K_f \frac{s \cdot p}{\frac{s}{\omega_f} + 1} \tag{3}$$

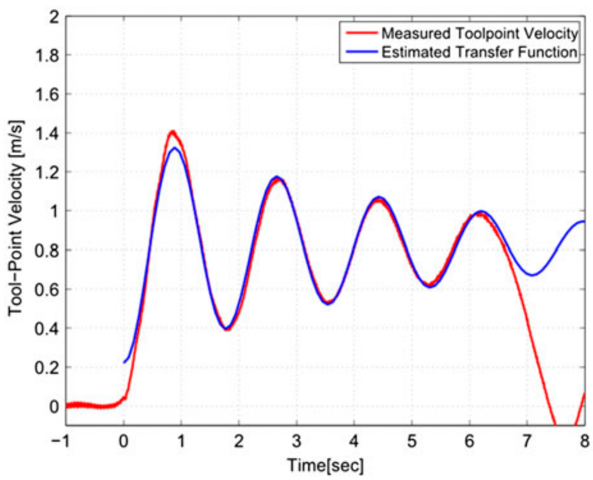

Figure 5. Estimated transfer function of tool point velocity.

Table 1. Damped natural frequency and damping ratio of crane.

Parameter		
ω_s	0.57	Hz
ζ_s	0.11	–

In (3) p is the metering-in pressure, either p_1 or p_2 in Figure 2. The filter has two tuneable parameters. The filter gain K_f and the filter frequency ω_f. Implementing the filter in the control of the directional control valve requires feedback from the system pressure, which is easily available in the physical crane. The parameters of the was determined based on the procedure outlined by Hansen and Andersen (2010).

4. Input shaping

As described by Singhose and Seering (2011), when a step command input is given to a flexible system where two or more masses are connected in series with springs, equivalent to a flexible link, the input will act on the first mass or in the base of the flexible link. This will cause a deflection on the first spring, and will cause a series-reaction which will cause the whole system to start moving.

Because the system is connected with springs i.e. flexible, the whole system will start to oscillate. One way to reduce the oscillations is to ramp up the command signal more slowly, but this is may be difficult with an operator in the loop and it will be detrimental to productivity. Alternatively, information on the natural frequency and damping of the system can be used to shape the input so that fast response and reduced vibrations are obtained.

An example of a shaped signal is shown in Figure 6. In this figure, two positive impulses cause motion in a system. Careful timing and sizing of the second impulse yields a constant resulting velocity.

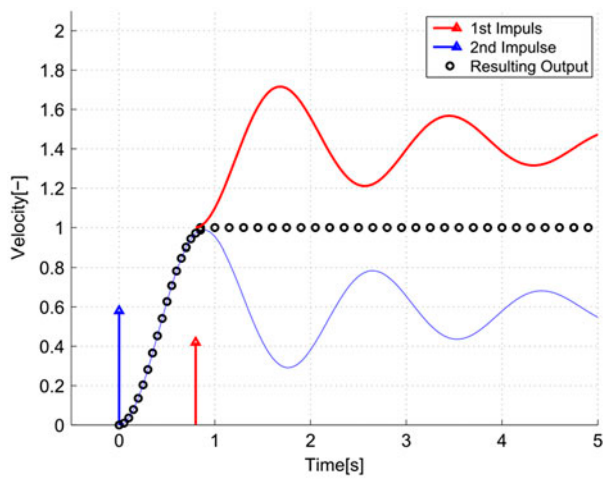

Figure 6. Timing and amplitude of impulses.

The timing and amplitude as shown in the figure can be found from the mathematical formulation of residual vibration, V, from a sequence of impulses (4).

$$V(\omega_n,\zeta) = \sqrt{C(\omega_n,\zeta)^2 + S(\omega_n,\zeta)^2} \tag{4}$$

$$C(\omega_n,\zeta) = \sum_{i=1}^{n} A_i e^{\zeta\omega_n t_i}\cos(\omega_d t_i) \tag{5}$$

$$S(\omega_n,\zeta) = \sum_{i=1}^{n} A_i e^{\zeta\omega_n t_i}\sin(\omega_d t_i) \tag{6}$$

In (4)-(6), A_i is the amplitude and t_i is the time location of the impulses, n is the number of impulses in the impulse sequence, t_n is the time location of the final impulse, and ω_d is the damped natural frequency. The damped natural frequency is defined in the usual way as:

$$\omega_{\mathrm{d}} = \omega_{\mathrm{n}}\sqrt{1-\zeta^2} \tag{7}$$

In many situations there is a demand for a resulting steady state output. This is obtained by simply replacing any impulse with a step that has the same time location and the same amplitude. In that case, to avoid attenuation, we apply the following constraint on the normalized amplitudes:

$$\sum_{i=1}^{n} A_i = 1 \tag{8}$$

The amplitude A_i could have both positive and negative values, but in this paper only positive amplitudes are considered. This is described by:

$$A_i > 0, i = 1\ldots n \tag{9}$$

The objective of the input shaping is to have zero residual vibration and to do this in the shortest time possible. For simplicity the number of steps is limited to $n = 2$. To get zero residual vibrations, (4) must be equal to zero. This can only be done if both (5) and (6) is equal to zero independently. The constraints formulated in (8) and (9) must also be fulfilled. Substituting $t_1 = 0$ into (5) and (6) results in the following two equations:

$$0 = A_1 + A_2 e^{\zeta\omega_n t_2}\cos(\omega_d t_2) \tag{10}$$

$$0 = A_2 e^{\zeta\omega_n t_2}\sin(\omega_d t_2) \tag{11}$$

Solving (11) for the smallest value of t_2 gives:

$$t_2 = \frac{\pi}{\omega_d} \tag{12}$$

The amplitudes A_1 and A_2 are based on the decay in the signal due to damping. Using (8) and (12) in (10) yields the solution for the damped system:

$$A_1 = \frac{1}{1 + e^{\frac{-\zeta\pi}{\sqrt{1-\zeta^2}}}} \tag{13}$$

Using the experimentally verified waves from Table 1 and inserting them into (12) and (13) gives the solution for the two amplitudes and timing of the second step:

$$\begin{bmatrix} A_i \\ t_i \end{bmatrix} = \begin{bmatrix} 0.5865 & 0.4135 \\ 0 & 0.8845s \end{bmatrix} \tag{14}$$

These values are used when shaping the input signal shown in Figure 6. The system function is based on the parameters in Table 1.

In Figure 7 the resulting input signal is shown as the sum of the two steps. Further, this signal should then be multiplied by the desired steady state output value.

5. Implementation

The control strategies for both the input shaping and the pressure feedback are implemented on the real time control system.

The implementation of the pressure feedback filter is placed in a cycle loop, running at 200 Hz. The filter is as described in (3) with the output subtracted from the reference signal, u, as shown in Figure 8 where the logic block simply corresponds to choosing the active inlet pressure depending on the direction of motion.

To implement the input shaping, a function that delays part of the signal is introduced as shown in the block diagram in Figure 9.

In the current paper only the slew motion is controlled with the remaining degrees of freedom, q_{1-3}, held fixed and without applying any payload at the crane tip. Therefore, the eigenfrequency of the slew motion is constant, however, this is not the case when considering the crane in its entire operational range. Especially, the telescopic motion, q_3, will have a significant influence and because both control strategies depend directly on the value of the main eigenfrequency, they should be adaptive in order to work satisfactorily in any situation. It is outside the scope of this paper with a detailed investigation, however, an adaptive scheme based on an estimated

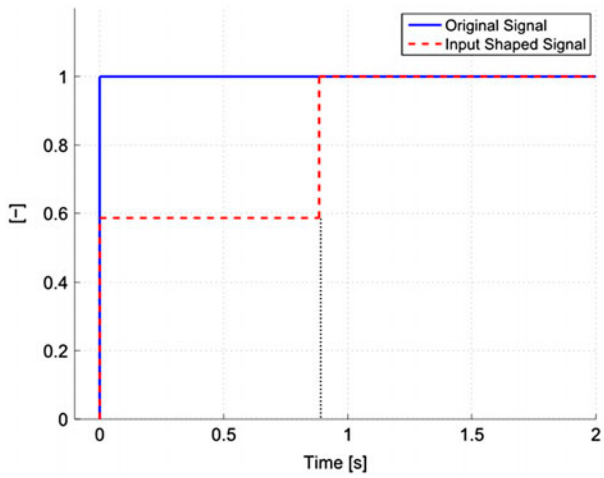

Figure 7. Shaping of normalized input signal.

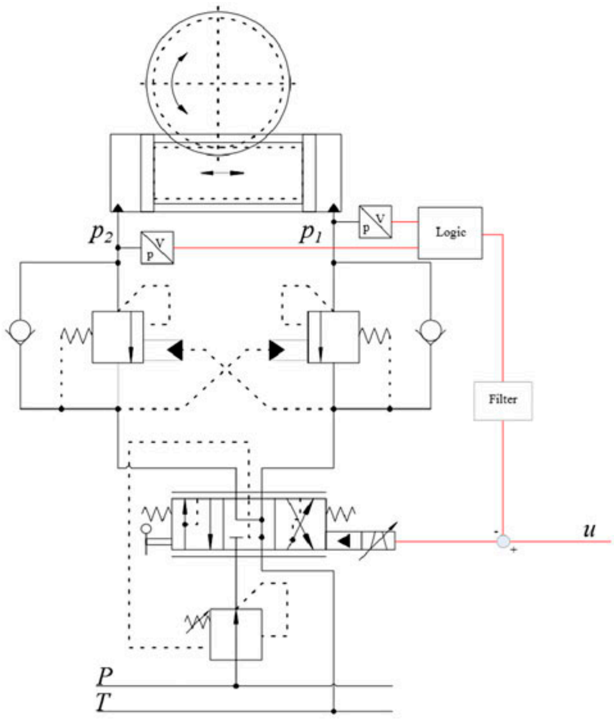

Figure 8. Diagram showing the main hydraulic and electrical circuits for the implementation of pressure feedback.

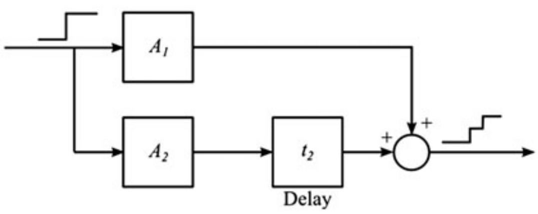

Figure 9. Block diagram for input shaping.

eigenfrequency obtained from a two-dimensional look-up table is proposed. The telescopic motion, q_3, and either a load cell at the crane tip or the piston pressure in the main boom cylinder could serve as entries to the look-up table. The boom motions $q_{1–2}$, could also be included, in an expanded look-up table.

Depending on the bandwidth of the PDCV the adaptive scheme must also include a cut-off strategy, i.e., the pressure feedback and the input shaping must be turned off, either gradually or abruptly, when the estimated eigenfrequency goes above a certain value because the PDCV is not fast enough. A rule of thumb often used in servo-hydraulics is to have a servo valve with a bandwidth of three times the eigenfrequency of the controlled system. In this case the bandwidth of the Danfoss PDCV is approximately 4 Hz, see Bak and Hansen (2012), and therfore it is well suited for the 0.57 Hz of the system, see Table 1. If the telescope is retracted or rotated into a more vertical position this frequency would increase whereas any added payload at the crane tip would reduce it. Hence, the eigenfrequency should be estimated continuously both for adjusting the control parameters but also to turn off or reduce the control effort.

6. Experimental verification/results

The full scale vehicle loader crane used for the experimental verification can be seen in Figure 10. It is a fully functional commercial loader crane with all the original parts and it is instrumented with position and pressure sensors.

The same strategy of using velocity ramps of the command signal, as in Figure 3 is done in the experimental trials. A low, medium and high velocity reference is given to the crane, with and without the active vibration damping.

Each of the different control strategies are tested independently for each velocity (Low–Medium–High). This is done in order to see the impact of the different control strategies with the different velocities. Four modes are tested. The first is with no control, the second uses only the input shaping (IS), the third uses only pressure feedback control (PFB) and the last is a combination of IS and PFB. In all the tests the same parameters for both pressure feedback and input shaping are used.

6.1. Low velocity

The crane is set to move with a tool point velocity of 0.25 m/s as seen in Figure 11. There is some improvement in the damping of the system when applying IS, however, only PFB (or IS + PFB) seem capable of suppressing the vibrations during motion.

This is so, because at low speed the combination of pressure compensated flow input and overcenter valve yields instability. This cannot be handled by means of IS but PFB is dedicated to the removal of this type of

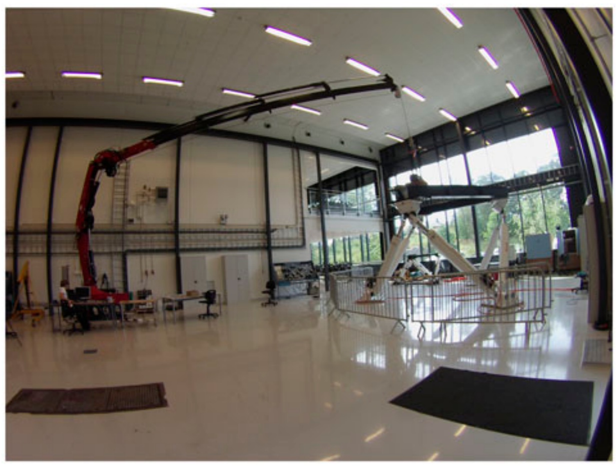

Figure 10. Photo of the vehicle loader crane located in the motion laboratory.

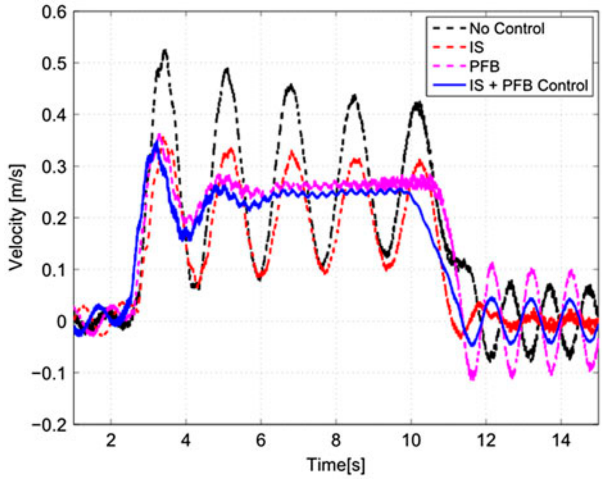

Figure 11. Low tool point velocity with different control strategies.

vibrations during motion and performs well until the nominal signal goes to zero and the motion should stop.

After that, the PFB performs poorly because it is not well suited for position control when the directional control valve has a certain dead band as in this case.

Clearly, PFB is a good solution for speed control of systems prone to hydraulically induced vibrations. In the experiments PFB remains active even after the reference speed is exactly zero. In practice, it should either be turned off or adjusted to handle this special situation, where the spool preferably should remain stationary in neutral. IS has only limited positive effect on such a system.

The pressure difference $\Delta p = p_1 - p_2$, in the hydraulic slew cylinder is shown in Figure 12. It can be observed that the oscillations in the pressure are reduced with the combined pressure feedback and input shaping control.

6.2. *Medium velocity*

The medium slewing velocity of the tip is 0.8 m/s. When comparing Figures 11 and 13 it is clear that the system is better damped at higher speeds and that the hydraulically induced vibrations are less dominating.

On the other hand, the steeper acceleration introduces structural vibrations that IS is well suited to counteract. Because of this, the vibration damping of IS and PFB are of similar size and here a combination of IS and PFB is clearly a good choice.

The pressure difference in the hydraulic slew cylinder is shown in Figure 14.

6.3. *High velocity*

Due to physical limitations in the experimental setup, the motion with high slewing velocity can only be active in 4 s. It does, however, reach the referenced velocity of 1.7 m/s.

The structural vibrations caused by acceleration are even more pronounced and the natural damping is similar to that witnessed during medium velocity. Figure 15 shows that IS is now superior to PFB because the vibrations predominantly come from the increased acceleration. PFB is only marginally better than the non controlled system during motion and continues to have difficulties when the system is supposed to be at rest.

Despite this, the combined solution of IS and PFB offers a clearly better performance than IS alone.

Without any control the tool point velocity seen in Figure 15 has an error of almost 2 m/s peak to peak. The vibration is almost cancelled out with the combined input shaping and pressure feedback control, even when the crane stops.

Pressure difference for high slewing velocity can be seen in Figure 16.

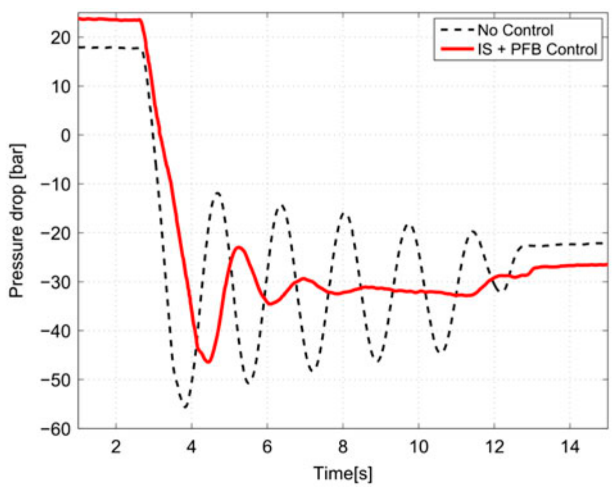

Figure 12. Pressure difference across the slew cylinder at low velocity.

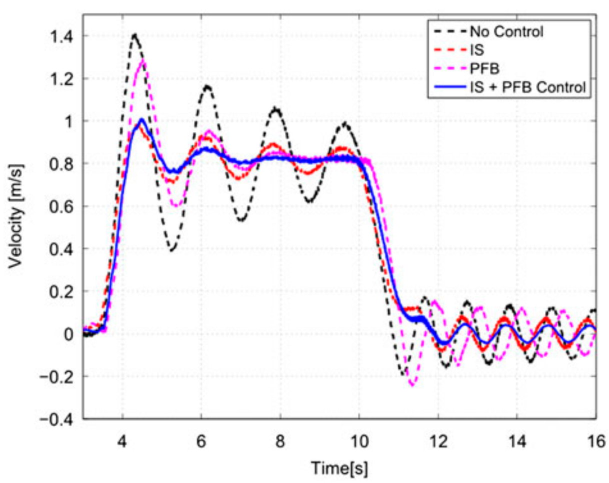

Figure 13. Medium tool point velocity with different control strategies.

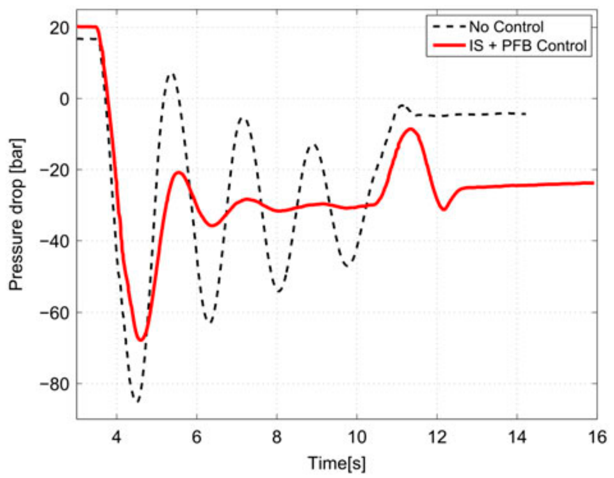

Figure 14. Pressure difference across the slew cylinder at medium velocity.

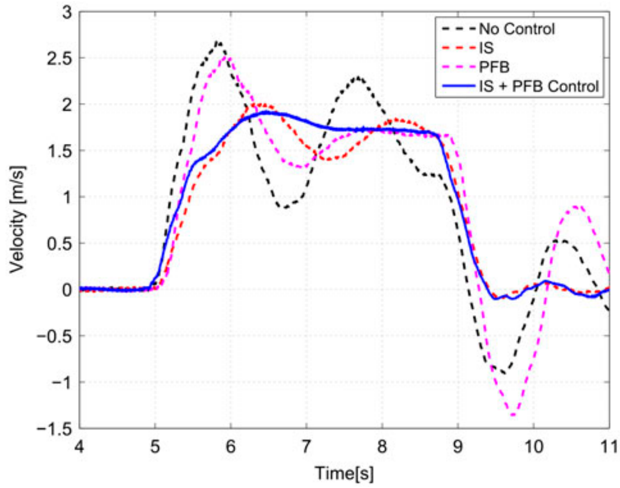

Figure 15. High tool point velocity with different control strategies.

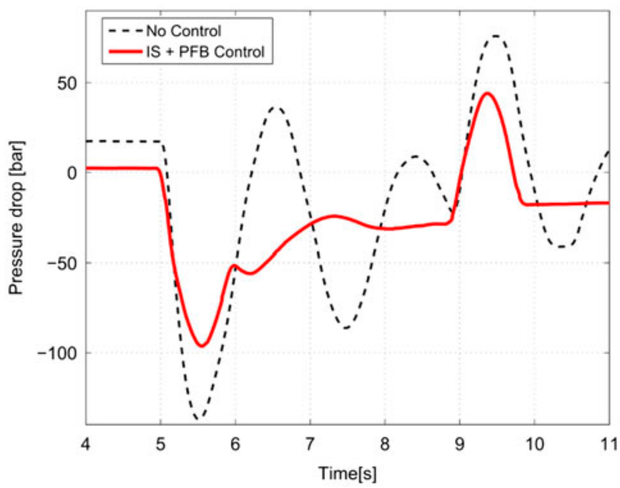

Figure 16. Pressure difference across the slew cylinder at high velocity.

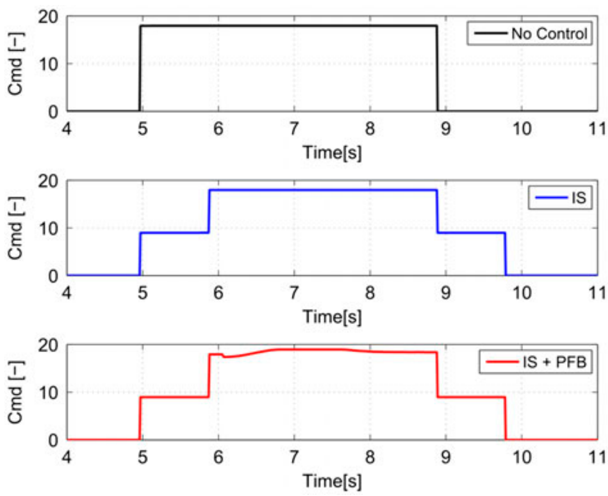

Figure 17. Command signal at high velocity.

To visualize the difference between the three control methods the command generation for high slewing velocity can be seen in Figure 17.

The first graph shows the original signal. The second shows the signal after the input shaping. The third is the input shaped signal combined with the pressure feedback.

7. Conclusions

The development of an active vibration damping control for the slewing motion of a hydraulic vehicle loader crane has been presented in this paper. The control strategy combines open loop input shaping with closed loop pressure feedback in the actuation of the electro-hydraulic proportional directional control valve.

The control scheme has been evaluated for three different velocity references on a full scale vehicle loader crane. Four different scenarios are compared: no control, only pressure feedback (PFB), only input shaping (IS), and combined pressure feedback and input shaping. It can be seen in the results that the pressure feedback has the most effect on the low velocity and less effect on the higher.

For the input shaping it is opposite, low effect at low velocity and most effect on high. For the pressure feedback this is explained by the effects of the counterbalance valves since they tend to be more stable at larger openings due to the increased flow-pressure gain. Similarly, the improved impact by input shaping at high velocities is explained by the increased importance of suppressing the excitation of the natural eigenfrequency of the system during ramp up and ramp down of the speed.

However, in all the three cases the combination of PFB and IS is the strategy that has the best dynamic performance showing a significant reduction in crane vibrations.

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Disclosure statement

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