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REVIEW PAPER

AN OVERVIEW ABOUT ACTIVE OSCILLATION DAMPING OF MOBILE MACHINE STRUCTURE

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Abstract

One main current demand for mobile machinery development is the improvement of operator comfort and productivity in order to be competitive on the global market in the future. Actual trends towards cost effective actuator systems in offroad vehicles and thereby the use of electrohydraulic actuators reflects also the task of active oscillation damping of the machine structure. In the past several concepts, which differ in hydraulic system, control and sensor strategy, have been developed. However, the practical use is still minor as in most cases passive oscillation systems, which base upon high pressure hydro-pneumatic accumulators, are widely used although the component costs and frequent check intervals are problematic for this technology. This paper presents an overview of research work done in the area of active oscillation damping technologies for offroad vehicles.

Keywords: active oscillation damping, vibration suppression, vibration control, active oscillation control, mobile machines, working hydraulics

1 Introduction

Different system solutions for active oscillation damping have been developed for different applications in the recent past. Some examples are: automotive area (suspension), aircrafts (structure damping), ships (stabilizers), trains (suspension), tanks (mostly cannon stabilization), or even hard discs, bridges or others. Due to the possibilities of improved control hardware and sensor technology in the past, the automatic task of active oscillation damping is able to enter more and more applications, especially where electrohydraulic actuators are used, in order to realize different aims like improved comfort, safety, productivity, lifetime, etc. A complete new field where active oscillation damping technologies has started to be of interest are offroad vehicles.

Considering the actual developments in the field of construction, agricultural, mining and earth-moving machines, a strong trend towards cost effective actuator systems and by this the use of electrohydraulic actuators can be obtained. With this kind of actuator the possibility of more automation of working cycles and support of the user in order to gain a higher productivity and comfort is realizable. However, the increasing competitive situation on the global market of machine manufacturers forces towards cost reduction, system simplification and higher reliability. The integration of new sensors and/or the serial CAN-Bus (CAN = Controller Area Network) generates new possibilities of partial and full automatic functions. Especially, the use of sensors originally developed for automotive industry offers due to the low price new possibilities for the development of active oscillation damping systems for the offroad vehicle market. One interesting full automatic task is the active oscillation damping of mobile machine structures using the machine working functions.

Most of today´s mobile machines with propel drive and working hydraulics have the problem of low frequency oscillations (first natural order) during travelling. Hereby, the machine frame and working structure are oscillating (pitching) intensively so that the operator has to reduce the driving speed, especially, in case a load is transported. Clearly, this leads to a decreased productivity of the machine and to a continuous disturbance of the operator. Today, only passive oscillation systems are subject to mobile machine series production. These systems base upon high pressure hydro-pneumatic accumulators which are connected by simple switching valves to the linear hydraulic actuators (Ulrich, 1980; Brönner and Kolb, 1983; Drake and Jaecks, 1991; Hosseini, 1992; Leidinger, 1992; Kum-

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mer, 1996; Ufheil and Rector, 1996; etc.). Main drawbacks of this kind of systems is the high component effort for the high pressure accumulators, in many cases the required regular safety checks of the accumulators, and the problem that the individual system can only be optimized according to a certain frequency. The oscillation frequency depends mainly on the machine load and velocity and ground condition, respectively. Advantage of passive oscillation systems is the fact that no additional energy consumption is required in contrast to active systems.

Generally, for mobile machines two active damping strategies based on hydraulic actuators are possible.

I. Damping of the machine frame by separate installed hydraulic actuators between machine frame and wheel axes (damping by suspension (e.g. Altmann et al, 1992; Crolla, 1996), used also in automotive applications), or damping by moving a damper mass - these system have a *single actuator function:* damping. Also semi-active damping systems can be mentioned here (e.g. Lizell, 1988 and 1994; Guglielmino et al, 2000; Heiskanen et al, 2003; Long et al, 1998) whereby in contrast to active systems a semi-active system can only dissipate energy and not introduce energy into the machine system.

II. Damping of the machine frame or/and the machine working structure by use of the machine working hydraulics (described in section 4) which consists mostly of linear actuators, i.e., in this case the machine working structure and load are used as damper mass and the linear actuators represent the adaptive spring/damper elements (damping by using working hydraulics) - these systems have a *double actuator function*: working and damping. And therefore, here the additional problem has to be noticed that the actuators are layout for working and not for damping in contrast to strategy I, finally generating special control challenges.

Concluding, the first option (I), requiring separate hydraulic actuators for active damping, is clearly too extensive for offroad vehicles due to additional installation and control effort. Therefore the research work in the area of mobile machines is concentrated on strategy II, i.e. the use of the existing actuators (e.g. working hydraulics) in order to realize an active oscillation damping. However, in many works the actuators were extended by separate valves or pumps, etc. in order to realize sufficient damping quality. And once again, note that this strategy raises the problem that layout of the used actuators for damping is not done for this task but for the normal working task.

The goal of this review paper is to demonstrate the state of the art in research on this very actual topic.

2 Machine Oscillations in Offroad Vehicles

In this section the machine oscillations of three machine classes which are relevant to this topic and have been mainly used for active damping in the past are described: mobile excavator, wheel loader and tractor. These machines are all travelling during work and thereby disturbances are introduced into the system. Figure 1 shows a mobile excavator as an example for a typical self driving mobile machine.

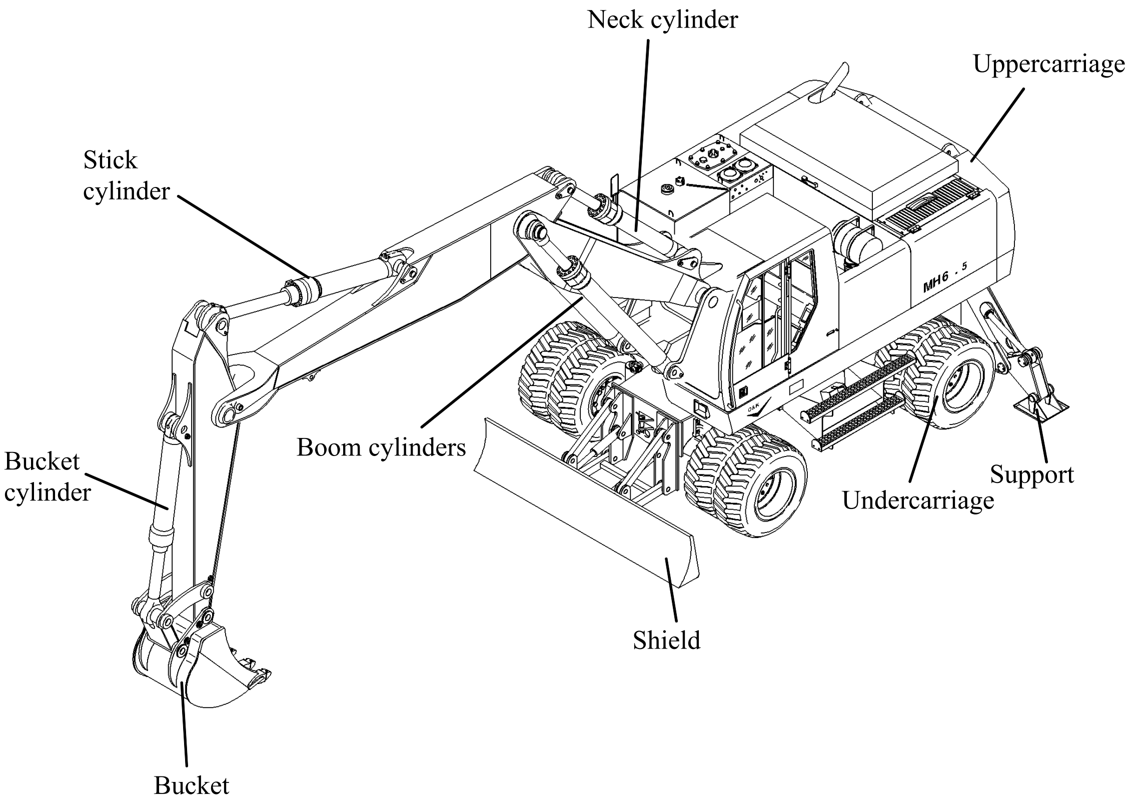

Fig. 1: *Mobile excavator as typical mobile machine with propel drive (O&K)*

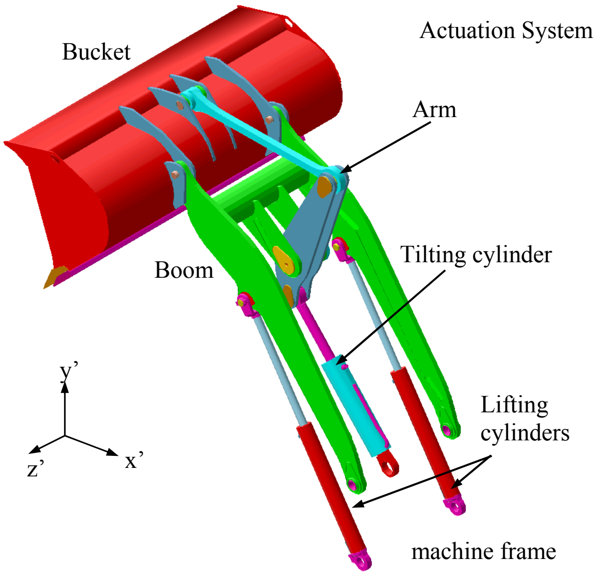

Fig. 2: *Wheel loader working hydraulics*

The hydraulic system of a mobile excavator consists besides propel drive and brake system mainly of a relative complex linear and rotary actuator system serving for the movements of: boom, stick and bucket, but often a neck cylinder is added to the boom structure for higher operator flexibility. And the swing is located in the uppercarriage to rotate relatively to the undercarriage.

The wheel loader represents another typical self driving machine. Figure 2 depicts a typical wheel loader working hydraulics which contains two linear actuators: lifting and tilting. Again the load is manipulated within the bucket. As said before, the machine and boom oscillations during driving lead in most cases to a reduced vehicle velocity and productivity here.

Figure 3 includes another typical application example for active oscillation damping. Here, a tractor hitch control is shown, where the load is realized by the hitch. The hitch actuation system normally contains one degree of freedom: lifting (mostly single operated), but often actuators like a top link cylinder are integrated.

All three machines have in common that normal air tires are used for wheels which influence the driving characteristics. Furthermore, automatic tasks and support of the user are desired by the customers today. Moreover, the use of more and more sensors and/or the CAN-Bus (Etschberger, 2000) technology has been progressed in last time (Weber, 2000; Bittner et al, 2001).

As said before, during driving of offroad vehicles on different rough grounds disturbance forces are introduced by the machine wheels. Furthermore, acceleration forces are caused by the propel drive. These forces are applied on the multi-body machine system leading to machine frame and working structure oscillations (and represent the disturbance for active damping). This is caused on the one hand due to the characteristics of the wheels and the suspension, and on the other hand due to the flexibility res. dynamic characteristics of the boom structure and load in combination with the actuator stiffness. Concluding, the main influences on the machine oscillations are the machine load and the driving conditions. And main influences on the boom oscillations are the boom movements commanded by the operator and again the machine load.

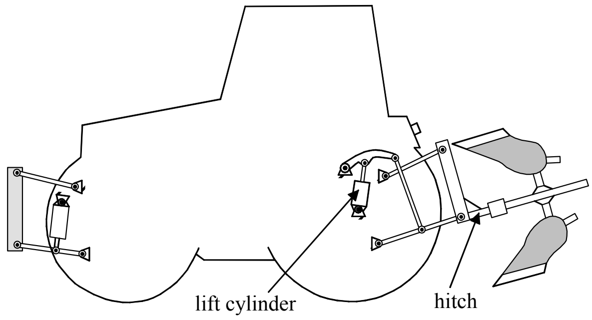

Fig. 3: *Tractor hitch control*

Generally, the oscillations, meaning the first natural frequency, which usually represents the important part, lies in a frequency range of about 1-3 Hz. Several measurements made at different locations of machines led to this result (Patel et al, 1999; Leidinger, 1992; Rahmfeld and Ivantysynova, 2003; Frediani et al, 2004). Normally, higher orders cannot be considered, where the actuators normally used for machine working function shall be simultaneously used for active damping, because of limited actuator bandwidth. The amplitudes in these frequency regions are so high that in some cases the actuator or bucket amplitude is about 0.5 m or even more.

For mobile machines with chain drives the active oscillation damping is of minor importance. In this special case the driving velocities are so small that the oscillation disturbance is relatively low. However, some works concentrated on the minimization of the natural boom oscillations which is also a kind of active oscillation control. These works will be described below among others.

Normally, two kinds of oscillations appear in mobile machines. Firstly, the dominant part is caused by the rotation (pitch) oscillations around the machine centre of mass. The linear lifting oscillations are the higher frequent but lower amplitude part.

Figure 4 shows principally a typical mobile machine boom structure which can be also found in the working hydraulics of the mobile excavator and wheel loader as well as in the hitch system of the tractor, where a boom structure is moved by a single rod cylinder. The equation of motion for the boom (according to boom rotating point 0) is (Rahmfeld, 2002):

$$J\ddot{\phi} = -M_R(\dot{\phi}) - M_L(\phi) + F_L \frac{dx}{d\phi} \qquad (1)$$

whereby changes in inertia (and therefore the expression $\dot{J}\dot{\phi}$) are being neglected. The kinematic relationship $dx/d\phi$ between cylinder and boom velocity depends upon the boom position ϕ res. cylinder position x. It yields:

$$\frac{dx}{d\phi} = \frac{\dot{x}}{\dot{\phi}} = \frac{l_c\, h \sin(\phi+\theta)}{\sqrt{l_c^2 + h^2 - 2 l_c\, h \cos(\phi+\theta)}} \qquad (1a)$$

This equation includes the geometric constants of the boom structure l_c, h and the angle θ.

Here, the load torque is:

$$M_L(\phi) = (2m_L + m_B)\,g\,l\sin\phi \tag{1b}$$

with the end-effector load mass m_L, the boom mass m_B, whereby the boom has a length of $2l$ in this case.

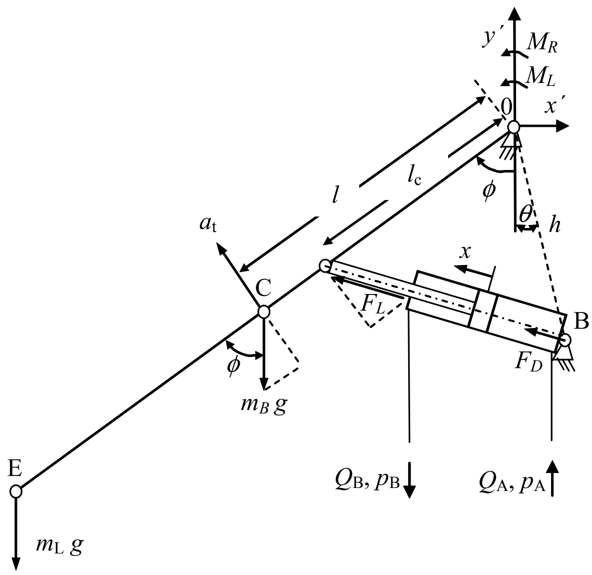

Fig. 4: *Typical boom structure of mobile machines*

Regarding the cylinder pressures, which cause mainly the cylinder force, the boom equation of motion becomes (neglecting $\dot{m}\dot{x}$):

$$J\ddot{\phi} = -M_R(\dot{\phi}) - M_L(\phi) + \\ + \left(A_K(p_A - \alpha p_B) - F_R(\dot{x}) - m\ddot{x}\right)\frac{dx}{d\phi} \tag{2}$$

Clearly, the disturbances for the boom are introduced via point B (summarized to a disturbance force F_D), influencing the force of the cylinder F_L by the cylinder pressures. As explained before, the disturbances are generated by the wheel forces coming from unsteadiness of the underground.

The cylinder rod and moved oil mass m can be neglected compared to the dynamic part (boom moment of inertia) on the left equation side. Furthermore, the velocity dependent boom friction torque M_R and cylinder friction force F_R will be neglected, too, as these parts are of minor importance here. The natural load torque M_L will be also neglected assuming no high change by variation in boom position ϕ, meaning nearly only static influence. Then the equation of motion becomes:

$$J\ddot{\phi} = F_L\frac{dx}{d\phi} = A_K(p_A - \alpha p_B)\frac{dx}{d\phi} \tag{3}$$

For the acceleration at the point C of the boom, normally to the boom axis (0-E, tangential acceleration), yields:

$$a_t = l\ddot{\phi} \tag{4}$$

Now one can write:

$$\ddot{\phi} = \frac{A_K(p_A - \alpha p_B)\frac{dx}{d\phi}}{J} = \frac{F_L\frac{dx}{d\phi}}{J} = \frac{a_t}{l} \tag{5}$$

And this equation gives the relationship between the cylinder pressure difference, the cylinder force and the boom acceleration under the above made assumptions. (Note that for determining the boom acceleration in some cases it might be necessary to measure also the machine/cabin acceleration in order to eliminate the machine oscillation during heavy pitching - caused by moving point 0.) As the boom is in a specific angular position range during active damping (during driving) the kinematic relationship $dx/d\phi$ as a function of the boom angular position is nearly constant. The same yields for the cylinder rod side pressure which can be also assumed as constant. So, the cylinder force is qualitative similar to the cylinder bottom side pressure and the boom acceleration.

This derivation shows that the use of

- force,
- pressure or
- boom acceleration

signal can be seen as *nearly* equal for active oscillation damping. However, one constraint has to be made. For active damping the dynamic signal is required. This leads in case of a pressure or force signal to the problem of eliminating the static signal which is often a problem to determine. This problem is avoided by the acceleration signal as the static value is known.

Figure 5 explains the general multi-body model of a mobile machine with working hydraulics: here as example for the tractor. The machine with mass m_2 is supported by the wheels as spring-damper elements. Therefore, the forces F_1 and F_2 are active on the machine axes. The working lift cylinder is symbolized by an adjustable spring-damper element which influences the movement of the working hydraulics with the load mass at the working tool m_1. The mass of the working hydraulics system can be utilized as damper mass known from machine dynamics. The lift cylinder realizes the required spring and damper constant. Note that the machine in Fig. 5 is depicted in horizontal position but mostly during driving the machine is rotating around the centre of mass (pitching).

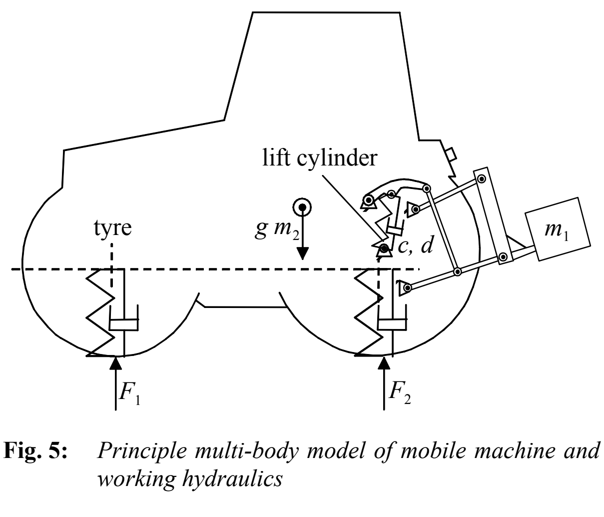

Fig. 5: *Principle multi-body model of mobile machine and working hydraulics*

2.1 Passive Oscillation Damping Systems

Figure 6 shows the often in industrial series production used passive oscillation system (mentioned above) (Ulrich, 1980; Brönner and Kolb, 1983; Drake and Jaecks, 1991; Hosseini, 1992; Leidinger, 1992; Kummer, 1996; Ufheil and Rector, 1996; etc.), here as an example for a wheel loader lifting drive. By a switching valve the high pressure accumulator(s) are connected to the piston side of the single rod cylinders where the rod side is connected to tank.

However, it has to be mentioned that the high pressure accumulator(s) cause a high component effort and underlie often regular check intervals. This method does not allow an adaptation of the system (damping and eigenfrequency by hydraulic capacity), i.e., the dynamic characteristics can be only influenced optimally for one operating point. I.e., the dynamic characteristics cannot be changed during operation and be adapted to the oscillation profile.

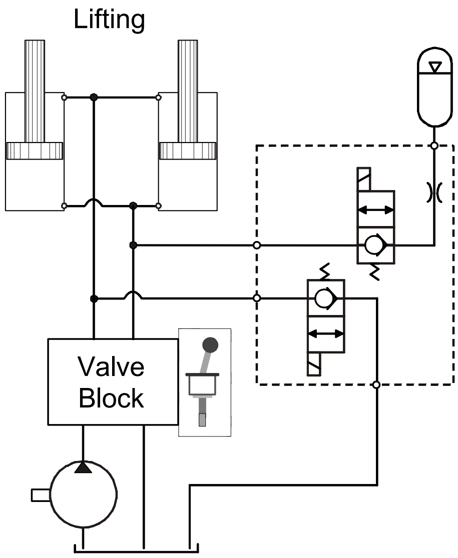

Fig. 6: *Passive oscillation damping for a wheel loader - hydro-pneumatic accumulators at the bottom side of the single rod lifting cylinders*

Another point to mention is that the passive oscillation system can be only used on the construction site or offroad due to public traffic laws existing in most countries. This yields also for active oscillation concepts as many common laws require cut-off of the working cylinders in case of driving on public road.

Main advantage of passive oscillation systems is the fact that no additional energy consumption is required in contrast to active systems.

2.2 Active and Semi-active Damping Strategies

For realizing an active damping in order to improve the result of the passive dampers and to reduce the component effort, different strategies are useful and have been investigated in the past. Following strategies can be applied:

- damping of boom structure,
- damping of machine chassis/structure or
- combined damping.

Whereby this can be basically achieved by active and semi-active damping. In semi-active systems kinetic energy can be only dissipated but not introduced into the system (see chapter 3).

In fact, the pure damping of the boom structure can be analyzed as a partial state feedback when designing the damping controller. The proportional feedback of force, pressure or acceleration represents a partial state feedback which allows a modification of the system poles. By that a higher damping can be realized. In Fig. 7 and 8 the principle difference of boom and machine damping is outlined in control block diagrams. In both cases a feedback is necessary, but in case of the boom damping, clearly a direct feedback of the control plant is given back - this is the main difference. That is the reason why in case of the machine damping the boom mass is utilized as a damper mass and the linear actuator as an adjustable spring damper element. In case of boom damping, the boom is directly damped.

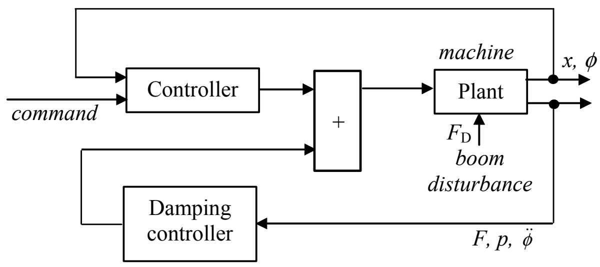

Fig. 7: *Principle damping strategies: boom structure damping*

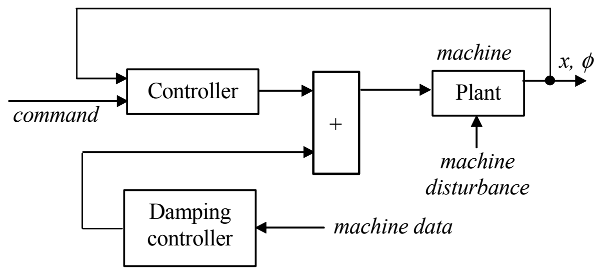

Fig. 8: *Principle damping strategies: machine structure damping or combined dampings*

However, for damping the machine structure or combined damping further sensors are required than the sensors mentioned above. Both, damping the boom and the cabin (ride) leads to an improved working environment and a higher productivity. Especially, in case of transporting sensitive goods the boom structure should be damped, in case of just driving, the machine structure should be damped.

First developments according to active oscillation damping were made for agricultural machines like tractors. The reason for this is the fact that here the machine owner is also mostly the user and is therefore willing to invest for special features like an active damping what simplified the development in an earlier stage, in contrast to e.g. construction machines.

3 Active Damping Technologies in Other Areas

In total, many investigations according to active stabilization and damping (according to principle strategy I, see above) have been done for instance in the automotive field. Hereby, exclusively valve controlled actuators have been used (e.g. Murrenhoff und Wallentowitz, 1998; Öttgen und Bertram, 2001; Bräss und Seiffert, 2000). For instance, Rieger und Schiehlen (in Burrows and Keogh (1994)) introduce an active oscillation damping concept for a vehicle wheel suspension (McPherson suspension of middle-class car), see Fig. 9. In this concept a hydraulic cylinder is implemented in the suspension and controlled by a servo valve based on a force control. Especially, for low frequencies an improvement of damping characteristics in comparison with a passive system could be reached. However, the use of valve controlled actuators for this task is always combined with throttling losses and a load dependence, especially in case many actuators are supplied by a central supply simultaneously.

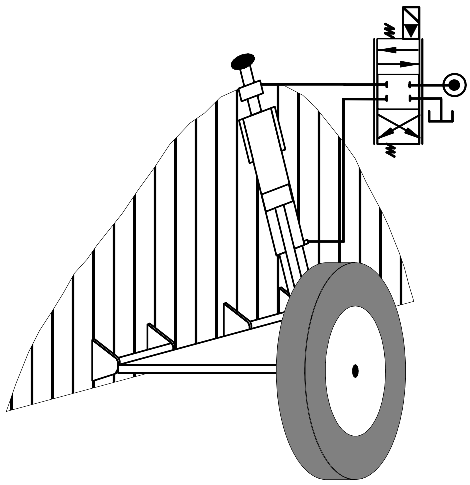

Fig. 9: *Active damping by suspension of a car (Rieger und Schiehlen in Burrows and Keogh, 1994)*

One example of the automotive area according to semi-active damping can be found in Fig. 10 (Guglielmino et al, 2000). In an automotive vehicle suspension the friction damper element comprises a friction pad fixed to the wheel which acts on a plate attached to the chassis of the car. The normal friction force is controlled by an electrohydraulic actuator whereby position and velocity feedback is utilized. Different hydraulic designs have been analyzed; the final design consists of a proportional underlapped control valve with a spool as final control element. From the efficiency point of view semi-active damping is very problematic as on the one hand energy is dissipated for damping, and on the other hand, in case valve controlled actuators are used, additional throttling losses are generated.

An example for semi-active damping in train technology is described by Sasaki (2003) for the current Shinkansen train configuration or by Streiter et al (2001) who also consider active damping strategies.

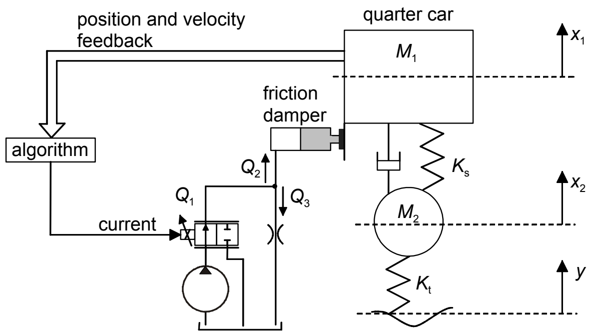

Fig. 10: *Semi-active friction damping of a car (Guglielmino et al, 2000)*

In the area of aircraft technology the damping of the aerolastic effects within the aircraft structures is often an important topic. Hereby, most strategies utilize the implemented flight control actuators to realize this (e.g. Merkel et al, 2002). In this case the damping is realized by pole (dynamics) assignment of the closed control loop, meaning that there is not a specific damping control routine utilizing special sensors for this task.

There is, of course, a variety of further proposed active damping concepts but due to aim of this paper all concepts described further are only related to offroad vehicles.

4 Active Oscillation Damping Methods developed for Offroad Vehicles

In this section recent research results achieved at universities, research centres and industry are being summarized. The developed active oscillation damping methods can be classified into the following:

- first and special damping approaches,
- damping by force control,
- by pressure control,
- by velocity control,
- by combined controls,
- by acceleration control,
- by displacement control.

Finally, a comparison is given.

4.1 First and Special Damping Aproaches

First developments according to active oscillation damping were done in agricultural machines like tractors. In Kauss et al (1984 and 1985) a combined damping system is introduced for a tractor hitch system, which was one of the first active damping approaches in tractors. Firstly, kind of a passive damping approach is done as the rigid coupling of the attachment of the tractor is replaced by a damping element res. vibration absorber designed for one specific oscillation frequency. Secondly, an auxiliary damping control is realized whereby a 3-way control valve for the hitch lifting cylinders is used. This damping control bases upon position measurement in the hitch system as well as tractor pitch angle and one acceleration signal which can be replaced by pressure measurement. It is on overlying control loop next to the hitch position control.

Sunamura et al (1990) invented a special low pressure circuit for an excavator lifting cylinder drive in order to damp the pressure oscillations when changing from lifting to lowering. I.e., the aim was similar as in the works by Yamagata (1993). Different hydraulic methods of connecting the lifting cylinder piston side temporarily to the cylinder rod side, the tank or an accumulator are explained by Sunamura et al (1990) by the help of solenoid valve. This concept can be extended by different sensor information like boom position, etc.

Other active oscillation damping concepts using different approaches than the utilization of the machine working hydraulics should be only mentioned briefly. An interesting concept bases on the use of the machine transmission for damping whereby the machine is a skid steer loader (Hansen et al, 2002) which is similar to the hitch control in (Andersen and Hansen, 2003). The machine consists of a standard hydrostatic transmission with one variable displacement pump and fixed motor. The active damping control strategy contains a velocity control of the machine pitch angle with desired value zero, and the pump is the final control element. Thereby, the active damping controller follows as a simple DT1 high pass filter whereby the break point frequency ω_0 in the numerator is set below the lowest natural resonance frequency in the system: ω_0 = 1 Hz, control gain is K = 0.5:

$$G_c(s) = \frac{u(s)}{\dot{\phi}(s)} = \frac{K\,\omega_0\,s}{s+\omega_0} \tag{6}$$

The first results showed a strong reduction in the pitch angle oscillations during driving. Required is a high quality machine pitch angle measurement. However, the concept was published with simulation only, the experimental proof is missing.

4.2 Damping by Force Control

As mentioned above, first developments according to active oscillation damping were done in agricultural machines like tractors, and this yields also for first electrohydraulic concepts in closed control loop. The reason for this is the fact that the machine owner is also mostly the user in case of agricultural machines and is therefore willing to invest for special features like an active oscillation damping what simplified the development in an earlier stage, in contrast to e.g. construction machines. As in some hitch control systems a force (draft) and position sensor are already included, the active oscillation system is based on these signals: F and s. Hesse (1991) developed a cascade control for active oscillation control where the innermost loop is the force (oscillation damping) control and the outer loop is the position control of the hitch, see Fig. 11.

With the help of a low pass filter the static force $\overline{F}_0$ was estimated and subtracted from the total force signal F_0 to achieve the dynamic force part F_{dyn} for the oscillation damping control. Figure 12 depicts the corresponding control scheme. The results were quite promising as the oscillations of the front axis could be reduced up to 40 %, so that the steering of the machine became much more secure. However, the use of an expensive force sensor in contrast to many hitch control systems of today, and the additional energy consumption for the active damping control are two clear disadvantages of this solution.

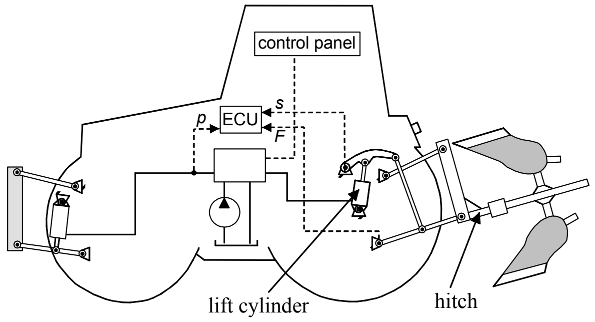

Fig. 11: *Tractor hitch active oscillation control - principle (Hesse, 1991)*

These works resulted also in different patents which base upon the concept from above (Göhlich and Hesse, 1986; Aichele et al, 1989; Höfer et al, 1991), whereby additional sensors were used in the earlier works.

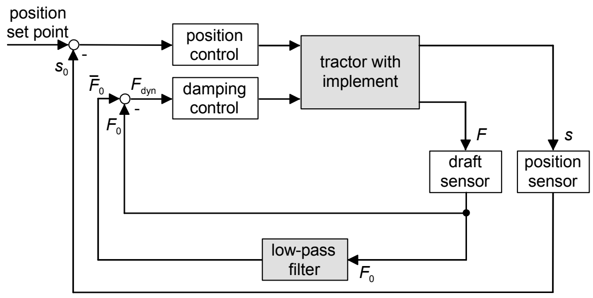

Fig. 12: *Tractor hitch active oscillation control - control structure (Hesse, 1991)*

A similar tractor hitch damping represents the concept by Maichle (1989, 1990) which bases upon a cascaded force and position control, whereby the position control is the underlying loop in contrast to the concept above. The force is measured in the attachment joint. The control signal is commanded to the standard electrohydraulic lift valve. Interesting is that for dynamic force estimation a filter with variable time constant is used and that the position controller has an adaptive gain.

In an excavator application Kuromoto (1993) introduced a force control via the standard electrohydraulic lifting spool valve (inside the valve block) of the boom oscillations during working. By the measurements of the two single rod cylinder pressures (or force directly) the force res. acceleration is calculated. The damping control signal generated by a band pass filter and a nonlinear gain is compared with the operator control (joystick) and the higher value is commanded to the electrohydraulic pilot control of the lifting valve ensuring full working functionality. The hydraulic valve supply is an open centre system with a fixed displacement pump what leads to a high energy consumption especially for damping action where not often the full pump flow rate is required. These works were continued for a crane application (Seichii, 1994).

Patel et al (1999) developed an active damping for a tractor hitch system. Hereby, a force and a position sensor in the hitch kinematics are used similar to the concepts above. Again a single operating hitch cylinder is subject to this invention. Whereas a valve block with two electrohydraulic (pilot controlled) valve units is used in order to connect the cylinder bottom side to the pump or tank line. A special *Chebychev* filter is utilized to determine the dynamic force so that the reaction time for the damping control is minimized. The filter time constant is thereby depended on the hitch position leading to an advanced low pass filter.

4.3 Damping by Pressure Control

In Hesse (1995) as well as Lödige and Kaplick (1995) the ideas of Hesse (1991) are transferred onto wheel loader systems. In this case a pressure control was chosen in order to avoid a force sensor. The pressure in the bottom side chambers of the lifting cylinder was controlled. Once again, signal preparation by filtering was necessary. The machine load and boom were acting as damper mass with an adjustable spring-damper element in form of the lifting cylinders.

Two concepts were presented in these works.

- The use of a separate pump and control valve logic for the oscillation damping task in order to reduce the energy consumption and improve the dynamics, where the main valves are controlled manually.
- The use of the standard electrohydraulic lifting control valve (with standard pump) for the oscillation damping task.

These two concepts are illustrated with hydraulic scheme in Fig. 13 and 14, respectively, for a wheel loader system as described before.

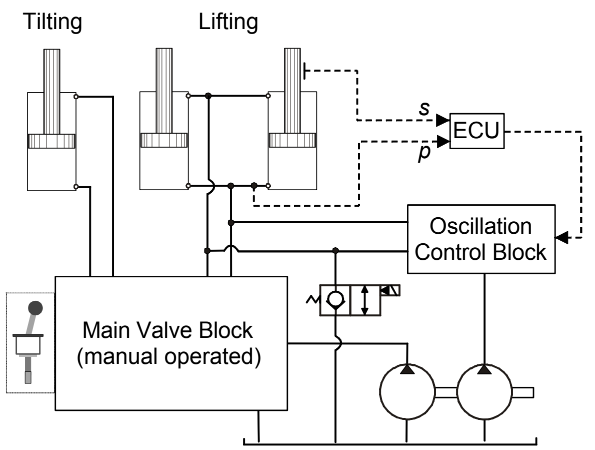

Fig. 13: *Wheel loader active oscillation control with separate pump and valve (Hesse, 1995; Lödige and Kaplick, 1995)*

For active damping of a grader machine working tool implement a damping concept was introduced (Hausman et al, 1996) which bases upon an electrohydraulic control system. Both pressures in both the working tool cylinders are measured and simultaneously a threshold pressure level is estimated by calculation. In case the current pressure level deviates strongly from the threshold value the pressure is being controlled to return to the threshold value. The hydraulic system contains a fixed displacement pump supply leading to high energy consumption for damping. Two separate single rod cylinders controlled by two directional control valves are used at the working implement.

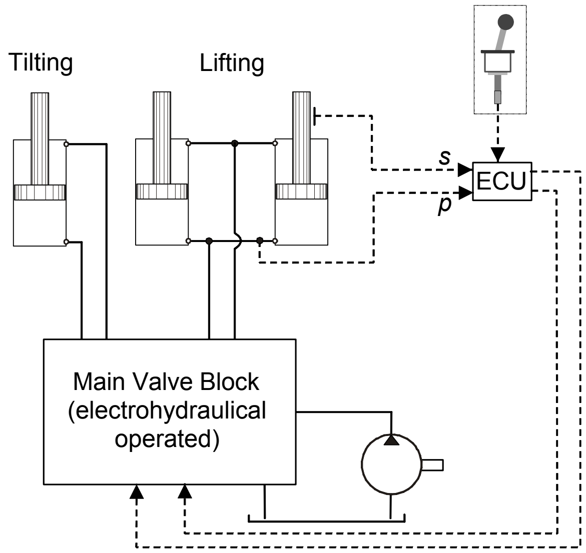

Fig. 14: *Wheel loader active oscillation control with electrohydraulic valve block and micro controller (Hesse, 1995; Lödige and Kaplick, 1995)*

In a tractor hitch system an active damping concept for damping during driving with raised hitch assembly was presented by Orbach and Schubert (1996). The active damping strategy contains a load (pressure) and position measurement with a double acting lifting cylinder and directional electrohydraulic control valve. Again a load threshold is determined and damping action is engaged in case of a high deviation. The aim is to improve especially the driving conditions of the tractor machine system for realizing and improved working condition for the operator (operator feeling) and to increase the productivity.

Next to the tractor application, in the other patent by Berger and Patel (1999) a pressure control for an active ride control for excavators and wheel loaders and other mobile machines was introduced. By measuring the pressure in the piston chamber of the lifting cylinders the pressure is being controlled to be constant, which is, of course, not fully possible. Next to this, a position sensor for the boom res. cylinder is required, e.g. angular position sensor. The idea is to use the standard valve for the lifting cylinder (electrohydraulic valve block). The active ride control is only active in case the machine is driving and no movements of the boom structure are commanded. The position and ride control signals are summed up and commanded to the valve whereby a dead zone for the position control ensures that position control does not disturb ride control in case of small position deviations. In fact, a flow command is transmitted by the ride control to the valve. The static pressure is calculated also with the help of the position sensor in an extensive set point generator which contains a low pass filter with cut-off frequency of 0.1 Hz. In Fig. 15 the basic idea is displayed. Furthermore, the idea of using an acceleration sensor instead of the pressure signal is mentioned in the patent.

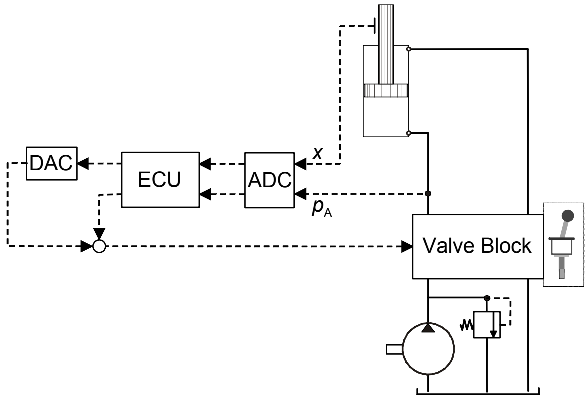

Fig. 15: *Excavator and wheel loader active ride control with electrohydraulic valve block and micro controller (Berger and Patel, 1999)*

Main problem of this active ride control is an extensive model of the valve including friction and deadband, etc. and of the control plant which is behind the strategy. This limits the transfer of this method to other similar machines using different hydraulic hardware. Furthermore, an extensive state estimator is used in order to realize the active ride control within the micro controller what makes the implementation of this concept difficult.

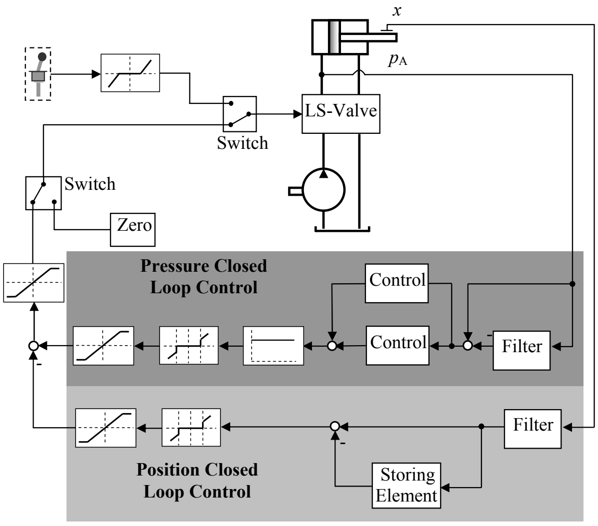

Fig. 16: *Wheel loader active oscillation control with electrohydraulic valve block and micro controller (Latour and Biener, 2002 and 2003)*

Latour and Biener (2002) developed an active oscillation damping concept focussed on wheel loader. The basis idea is a pressure control in combination with a position control as depicted in the block diagram in Fig. 16. The dynamic pressure is again estimated by a low pass filter (static value) and the subtraction with the measured value. The command value for the dynamic pressure is, similar to the concepts above, zero. The parallel position control is designed the same way. The effect is that high position deviations from the moment of activating the active oscillation damping should be avoided (by the position control loop).

In these works (Latour and Biener, 2002) the results of the active oscillation damping are compared to passive oscillation systems according to a wheel loader of 7.5 tons weight. A standard valve block and load-sensing pump for mobile applications were used firstly. It was found out that for high load masses leading to lower dominant oscillation frequencies (1 to 1.5 Hz) the active system could achieve the same damping quality as the corresponding passive system (based on hydro-pneumatic high pressure accumulators). But for low or no load which leads to higher dominant oscillation frequencies (2 to 3 Hz) the passive system behaved better.

Recent results (Latour and Biener, 2003; Kliffken and Geerling, 2003) confirmed these outcomes. The control valves used for working hydraulics in today's mobile machines are only partially suitable for an active oscillation damping as the demands were different in the past especially for earthmoving machines from the dynamic point of view. In the range of most interesting operating parameters the passive system is more efficient regarding damping quality (especially for high frequencies). Especially, the energy consumption of these active systems is not comparable with passive systems.

4.4 Damping by Velocity Control

Bernzen (1999) realized an active oscillation damping of a hydraulically driven large robot by using a force, velocity and position feedback with constant pressure supply. The cylinder was handled as a virtual spring-damper element with adjustable spring and damper constant. And the control elements were appropriate high dynamic servo valves. The principle control block diagram which bases upon a PI (proportional and integral) velocity controller can be found in Fig. 17, where also the virtual spring-damper element and the control plant are indicated. Bernzen (1999) analyzed in a first step the damping of hydraulic robots with flexible arms. The tuning of spring and damper constant is important to avoid a slow system and to realize a high damped system. However, it could be proved that no stability problems can rise by this control concept. In fact, the pressure build-up dynamics were neglected as the equation of motion dynamics is dominant (dominant conjugated pole couple). For the commanded cylinder velocity yields with the basic force equation for a spring damper element:

$$\dot{x}_{\mathrm{com}} = -\frac{F(t) - c(x_0(t) - x(t))}{d} \tag{7}$$

With F as measured cylinder force, c and d as by the designer defined spring and damper constants, x as measured cylinder position.

For larger hydraulic linear drives with differential cylinder a nonlinear compensation was introduced next to the virtual damper element. However, the system damping increase was more limited in this case as the actuator is naturally low damped. Therefore, the achieved damping quality compared to non-damped controller structures was minor. Additionally, the use of three sensors, or minimum two sensors including a force sensor in case the position signal can be differentiated once with sufficient quality, is problematic for a transfer of these techniques to mobile machines. Other aspects are the high effort in controller layout and the required control hardware.

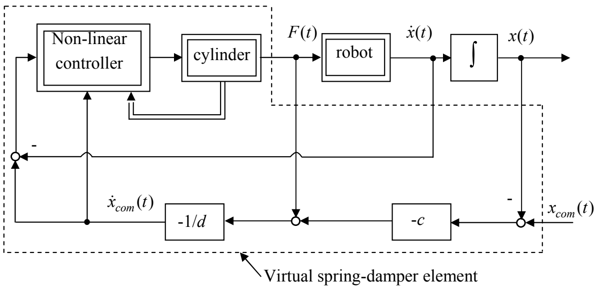

Fig. 17: *Active oscillation control for hydraulic robots (Bernzen, 1999)*

The works of Bernzen (1999) were transferred onto a mobile forge robot by Deckers et al (2000). Again the control element here was a servo valve with high bandwidth. The force measurement was recommended to be replaced by a force calculation with the help of the pressure measurements within the differential cylinder neglecting friction and acceleration force. A major problem with force and pressure feedback here is still the elimination of the static part by estimators. In fact, the static part has to be removed totally in order to assure good positioning and stability during active damping. The coherent conclusion made by the authors according to this concept was that it is too extensive for mobile machine applications.

In the work done by Michalowski and Sobczyk (2003) an active damping oscillation strategy of a mobile offroad manipulator is proposed by velocity control. A valve controlled actuator supplied by a fixed displacement pump is utilized for this purpose and the corresponding active damping control signal is calculated in proportional relation of the linear actuator velocity by:

$$u = -K_v \dot{x} \tag{8}$$

with: K_v as feedback loop gain.

The results showed that this damping concept is especially useful in case the disturbance force is acting with a frequency near the natural resonance (eigenfrequency) frequency of the actuator. However, the results could also indicate that the power consumption for active damping for higher disturbance frequency is over 30 kW for a high gain K_v, what proves the main problem of valve controlled active damping.

4.5 Combined Damping Controls

In the Japanese patent Yamagata (1993) the oscillation damping of the boom of a crawler excavator was investigated, see Fig. 18. In the combined velocity and force control the aim was not to reduce oscillations especially during driving of the machine. In fact, the goal was to reduce oscillations of the end effector or boom caused by starting res. stopping the actuator immediately during working similar as in Kuromoto (1993) and Seichii (1994). The cylinder force of the lifting actuator is calculated by the measured pressures in the two cylinder chambers. A high pass filter is used to guarantee that only interesting frequencies for the damping routine are considered. Next to this, a low pass filter is used in order to cut-off too high frequencies by subtracting its output from the high-pass value. Furthermore, the cylinder velocity and position are measured. The force and the velocity are used for generating the damping signal whereas the position is utilized for adapting the weighting factors (k_v, k_F) for force and velocity as a function of the static force. The commanded flow commanded signal follows as:

$$Q_{com} = -k_v(F_{stat}, x)\dot{x} - k_F(F_{stat}, x)F \tag{9}$$

For the active damping routine a separate control valve with electric input is used to which the flow command is commanded. Parallel to this a hydraulic-mechanical main valve is used which is connected to the joystick (operator). The hydraulic source realizes a constant supply pressure. Concluding, the operator feeling could be improved with this damping procedure. However, the required sensors and the additional hydraulic hardware demand (extra valve with lines) is very extensive for mobile machines. Additionally, the use of a constant pressure supply with fixed displacement pump results in high energy losses for the machine.

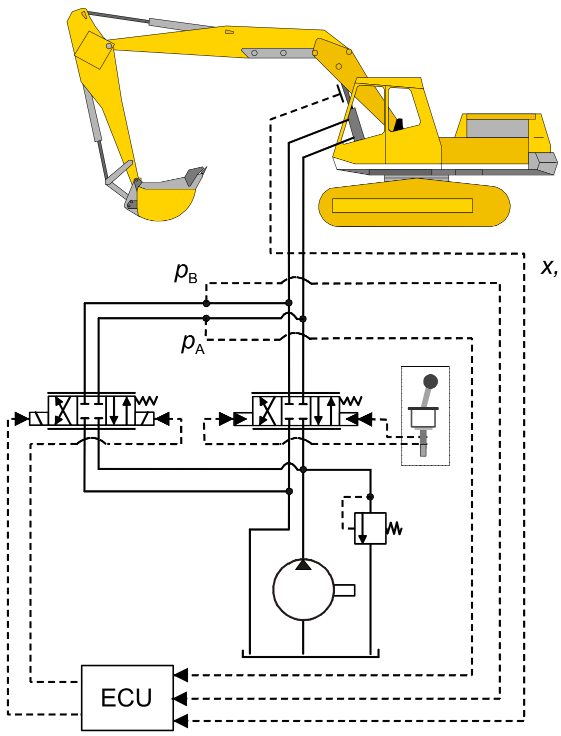

Fig. 18: *Active oscillation damping of an excavator - control concept (Yamagata, 1993)*

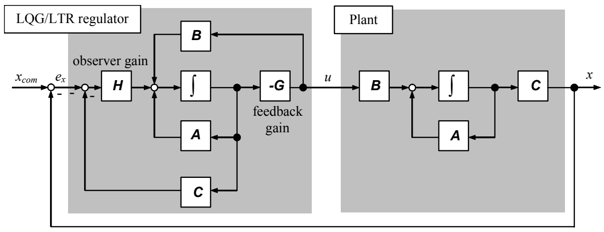

Fig. 19: *Principle of LQG/LTR control as combined Kalman filter with optimal state estimation by observer (Kawaguchi et al, 1994)*

Kawaguchi et al (1994) analyzed also the possible reduction of excavator arm oscillations during working by using an additional high dynamic servo valve next to the main control valve both supplied by a constant pressure source (same hydraulic concept as in Yamagata (1993)). In this investigation a model of the mechanical (with the help of identification) and hydraulic systems was derived in order to find an appropriate state space model. This model was the basis for the LQG/LTR (Linear Quadratic Gaussian/Loop Transfer Recovery, compare to Fig. 19 and similar to Sadri et al, 1999) controller design in order to increase the system damping and reduce oscillations during moving. By the state estimation (optimal state observer) behind the *Kalman* filter only the position of the single rod cylinder has to be measured. Hence, the LQG/LTR control method needs fast control hardware. The result was a reduction in arm vibration of about 50 % in specific arm positions which was researched with the help of impulse responses. Kawaguchi et al (1994) concluded that the use of LQG/LTR controllers is principally not recommended for mobile machines due to high implementation and signal processing effort. The authors finally recommended a gain scheduling controller which would lead to similar results reducing the control and control hardware significantly.

In Kovanen and Handroos (2002) the active oscillation damping of a flexible hydraulic manipulator is investigated, whereby the analysis res. the calculation of the damping signal bases on the concept by Yamagata (1993). The active damping control concept consists of a combined force and velocity control, whereby the force was calculated by the measured pressures of both cylinder chambers. Furthermore, the cylinder velocity was calculated by differentiating the cylinder with the standard difference quotient.

$$F = (p_{\mathrm{A}} - \alpha p_{\mathrm{B}})\, A \tag{10}$$

$$\dot{x} = \frac{x_{\mathrm{i}} - x_{\mathrm{i-1}}}{t_{\mathrm{i}} - t_{\mathrm{i-1}}} \tag{11}$$

The force and velocity values are passed over a high pass filter in order to remove the static parts.

$$G(s) = \frac{s^2}{s^2 + 2\,\omega_0\, D + \omega_0^{\,2}} \tag{12}$$

This filter is a double differentiating system with second order time delay (D2T2).

An extra high dynamic valve was omitted for damping control by transferring the flow command to the standard valve. And the control concept generates a flow command for this valve. Later the damping signal is also combined res. added to the joystick (operator) signal. In contrast to Yamagata (1993) the force and velocity weighting factor was set to 100%. So that for the damping control signal for the valve follows:

$$u = -\frac{K\,\dot{x} + F}{F_{\max}} \tag{13}$$

With K = 1 kg/s. However, during heavy work of the manipulator the results were minor as the operator did not stop the working functions. This is a major problem for the concepts aiming to reduce boom oscillations during working.

In Andersen and Hansen (2003) two different active damping approaches were presented for a tractor hitch system. In the first concept the control strategy is to hold the derivative of the hitch cylinder bottom side pressure to zero leading to a relative movement of the hitch to the machine. The second concept is a cascade structure where in the inner loop the pitching angle of the machine should be held at zero, and the outer loop controls the hitch position in order to hold the hitch in a specific position range. In this case the hitch is moved in opposite phase of the machine as the signal machine pitch angle is physically totally different from the pressure res. force of the hitch attachment. I.e., first concept utilizes the working hydraulics as damper mass and second concept damps the machine by direct measuring the machine oscillations. For the pressure or pitch signal a high pass filter as DT1 system is used as controller.

$$G(s) = \frac{K\, s\, \omega_0}{s + \omega_0} \tag{14}$$

The break point frequency is set below the lowest natural resonance frequency, here chosen to ω_0 = 0.5 Hz. Interesting is the comparison of the two basically different damping concepts which has been done. The pitch angle damping meaning the direct machine damping is the better solution when one takes the presented frequency suppression plots into account. However, the measurement of pitch angle is complicate to realize, and furthermore, the high pass filter has to be combined with further signal processing elements to neglect high frequency stimulations of the system. In a first step only simulation results were presented within this paper.

4.6 Damping by Acceleration Control

Generally, acceleration sensors offer especially in comparison to pressure and force sensors several advantages.

- It is easy to implement these sensors on a machine systems or even more than one sensor in order to consider more information of the oscillating machine structure.
- Due to the developments in the automotive area this sensor type has become very cheap, especially regarding capacitive sensor types.

- Most important is that the acceleration signal is oscillating around a known value: in most cases earth gravity or an earth gravity dependent value which can be simply calculated. In contrast to this, using a pressure of force sensor leads in most cases to extensive load estimators or extended filters in order to determine the dynamic part.

However, the use of this sensor type is still limited at the moment in the area of mobile machines. Moreover, it has to be mentioned that the signal quality and preparation is different for this kind of sensors signals.

Berger and Patel (1999) mention that instead of pressure control, also an acceleration control is possible using an electrohydraulic valve as final control element. Next to this, some further investigations mention an acceleration sensor for the task of active oscillation instead of pressure or force sensor in the area of mobile machines using valve controlled working hydraulics have been published (see section 4.7 and 4.1).

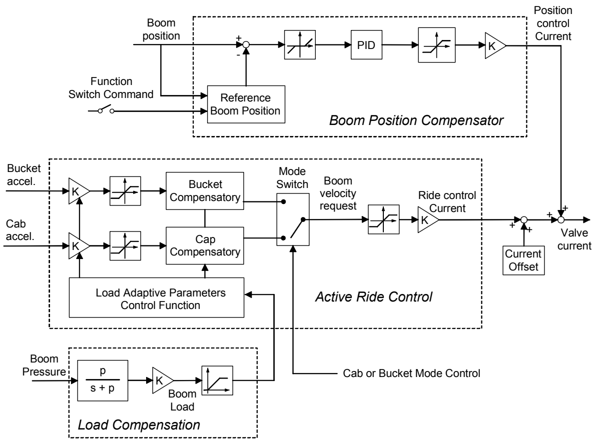

Fig. 19: *Active oscillation damping of a wheel loader based on acceleration and pressure signal (Frediani et al, 2004)*

In the recent concept (Frediani et al, 2004) the use of two one-axis acceleration sensors for the active damping according to wheel loader is utilized. Hereby the aim was to damp the boom (bucket) structure as well as cabin (machine structure) oscillations. The reducing of cabin oscillations is to improve the operator feeling and the bucket oscillations are to reduce material loss and allowing a high speed meaning productivity, respectively. Therefore a cab and a bucket mode have been developed each basing on the signal of one acceleration sensor. Additionally, a pressure sensor is used for adapting the damping control to the load situation of the machine as shown in Fig. 19. The active damping is done in parallel (simple adding of control signals) to the position closed loop control similar to the concepts of Latour and Biener (2002) and Rahmfeld (2002) and Rahmfeld and Ivantysynova (2003). I.e., a new position commanded value is generated by the operator and then the closed loop position control is running.

The achieved results during experimental investigations with a wheel loader are the following. The cabin oscillations could be reduced in comparison to the non-damped and the passive damped machine. The bucket oscillations have been reduced in the same way as with the passive damping system. Obviously, the bucket oscillations having sometimes multiple resonance peaks are harder to damp. Having achieved the same quality as the standard passive system, there is still the higher energy consumption for active damping which was not specified by the authors.

Here, also the concept by Hagemeister and Keuper (2002) should be mentioned (for a tractor) whereby next to a hitch sensor also a front axis sensor is utilized for active damping control.

4.7 Displacement Controlled Damping

A major problem of active oscillation damping using valve controlled actuators is the required additional energy consumption compared to passive systems. In fact, the machine boom has to be lifted and lowered in short time intervals. The potential load and brake energy is not used and always converted into heat which has to be cooled. Another main problem is given by the dynamic behaviour of today´s used standard control valves. The use of a load-sensing pump together with today´s electrohydraulic load-sensing valves requires a multi variable control concept and leads often to a low dynamics control system. Next to this, in case of parallel actuator movement, the load independency in load-sensing systems can only be guaranteed by energy wasting pressure compensators (Zähe et al, 1993; Gu and Wang, 2001). Obviously, the use of the electrohydraulic load-sensing (Esders, 1996) leads to complicated multi-variable control systems with an extended sensor and signal effort, and is therefore not in serial industry use up to now. Therefore, for this advanced task a simpler actuator technology is desired.

Principally, the use of displacement controlled (valveless) linear actuators raises the problem of volume flow compensation of the nearly exclusively used single rod cylinders. Next to this, most of the damping concepts from above can be transferred onto displacement controlled actuators when the servo pump is considered as final control element.

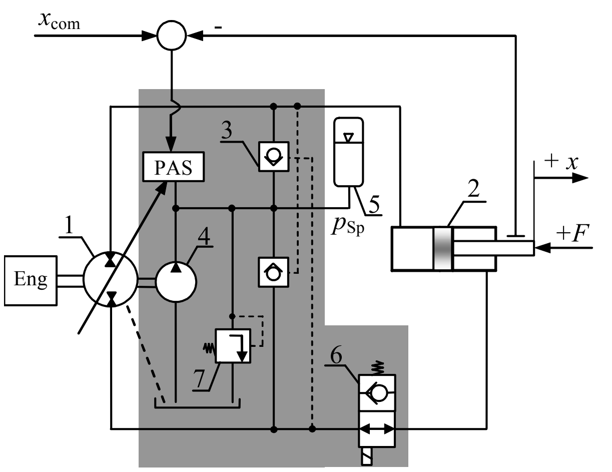

Fig. 20: *Displacement controlled linear actuator with single rod cylinder (Rahmfeld and Ivantysynova, 2000)*

Rahmfeld and Ivantysynova (2003) introduced the task of active oscillation damping with displacement controlled linear actuators based on single rod cylinder drive of a wheel loader (Rahmfeld et al, 2004). In earlier investigations a new simple displacement controlled circuit solution for a single rod cylinder was found an analyzed (Rahmfeld and Ivantysynova, 2000; Rahmfeld, 2002), see Fig. 20, and then firstly applied to a wheel loader working hydraulics (Rahmfeld et al, 2004). This simple circuit solution uses only one servo pump (1) as final control element. The proof of function in four quadrant operation was successfully done by experiment on a laboratory test rig and additionally on a full functional mobile machine.

Here, differential volume and volumetric losses are balanced on the low pressure side, whereby two pilot operated check valves (3) always connect the cylinder low pressure side with the pressurized low pressure level p_{Sp}. In Fig. 1 a charge pump (4) together with an accumulator (5) are used for pressurization of low pressure. Generally, the size of the low pressure source (pump (4) and accumulator (5)) is optimized according to typical working cycles of the actuator. Low pressure is given by the characteristics of the accumulator (5) and limited by the pressure relief valve (7). In this case the servo pump control system is also supplied by the low pressure level.

The grey box in Fig. 20 indicates that all additional elements inside the box can be usefully integrated into the servo pump housing. Today´s servo pumps for closed circuit mostly contain several components like for example relief valves, check valves and a charge pump. I.e., for one linear actuator there are only three individual components required (servo pump, accumulator and cylinder). By the use of a shut-off valve (6) loads can be hold also in case of an engine or pump failure. Further, a simple open loop operation of the actuator can be realized by the help of this valve.

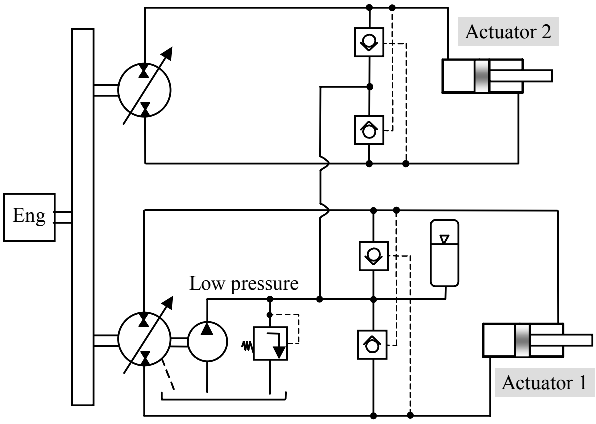

Fig. 21: *Coupling of displacement controlled linear actuators with single rod cylinder (Rahmfeld and Ivantysynova, 2000)*

Main advantage of this actuator concept is that in case of more than one actuator in the machine working hydraulics, the low pressure lines can be coupled. Figure 21 explains the resulting circuit for two linear actuators with single rod cylinder. Each cylinder is controlled by the individual servo pump as final control element. Due to a coupled pressurized low pressure again only one accumulator and one low pressure charge pump are required. In fact, in case of opposite movement directions of the cylinders the differential volume is directly balanced between the two cylinders allowing a reduced low pressure source in many cases.

Summarizing, following advantages are combined with the new displacement controlled linear actuator technology compared to valve control, especially load-sensing:

- less energy dissipation due to omitting throttling losses,
- use of load and brake energy for other drives in the machine and in fact, this energy has not to be cooled in contrast to valve controlled systems,
- less cooling power required,
- simplification of hydraulic system and
- easier control and operating.

Figure 22 includes the displacement controlled actuator control concept with active oscillation control. The control concepts bases upon a mixture of position and acceleration control. Normally, the active oscillation damping is switched off and the actuator position control bases upon a first order compensator and a quasi integrator (Rahmfeld, 2002) which avoids certainly integrator wind-up and limit cycles. Next to this, non-linear compensations for pump speed and kinematics are part of the controller as gain scheduling which is very important in order to achieve a high bandwidth. The control plant consists of the servo pump (position controlled system), the pump flow gain and an IT2 system for the mechanical cylinder res. boom part. By a switch it can be decided between OL (open loop) and CL (closed loop) control.

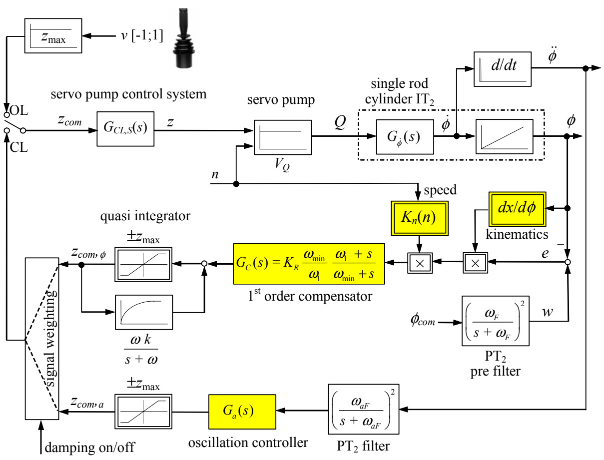

Fig. 22: *Active oscillation damping with displacement controlled linear actuators with single rod cylinder (Rahmfeld and Ivantysynova, 2003)*

The transfer function for the active oscillation res. acceleration control with desired value zero is:

$$G_{\mathrm{a}}(s) = \frac{\ddot{\phi}(s)}{z_{\mathrm{com}}(s)} = G_{\mathrm{CL,S}}(s)\, V_{\mathrm{Q}}\, G_{\dot{\phi}}(s)\, s \qquad (15)$$

$G_{CL,S}(s)$ is the dimensionless closed loop control transfer function of the pump, multiplied by the pump flow gain V_Q the pump volume flow results. And after the PT2 cylinder velocity/flow transfer function $G_{\dot{\phi}}(s)$ follows the boom acceleration by differentiating. I.e., only a D system needs to be applied to receive the DT2 cylinder acceleration transfer function. Note that this means that there is no static influence possibility on the acceleration. Concluding, an integral acceleration control corresponds theoretically to a proportional velocity controller.

The active oscillation control utilizes in a first approach an acceleration sensor signal at the boom of a wheel loader as example system. This signal is low-passed so that only interesting frequencies below 3 Hz are considered. As said above, the simplest useful acceleration controller is an integrator leading to the same control behaviour as a proportional controller in the actuator speed control. However, a proportional controller can also be used and is easier to implement, but the acceleration gain must be reduced then. Further investigations led to the result that a similar first order compensator as for position control is well applicable. One possibility is just to add the position and acceleration control signal. Another way is to implement a mixture control with the weighting factor W[0,1] leading to the commanded servo pump position, displacement volume, respectively:

$$z_{\mathrm{com}} = W\, z_{\mathrm{com},\phi} + (1-W)\, z_{\mathrm{com,a}} \tag{16}$$

The results with displacement controlled linear actuators for active oscillation control could indicate the strong potential of this actuator concept. The acceleration oscillations could be reduced by 60 % with this simple and easy to implement control concept. Furthermore, the energy consumption was analyzed also leading to a reduction of 25 % compared to LS when only considering the continuous energy recovery during damping (in a first step for low loads). For more details according to the use of valveless linear actuators for the task of active oscillation damping at a wheel loader machine refer to Rahmfeld and Ivantysynova, 2003.

4.8 Comparison

The last section has shown that a large variety of active oscillation damping concepts (according to strategy II) in the area of mobile machines has been developed by industry and universities. All methods are different and have pros and cons described above. Table 1 summarizes the concepts together with the individual parameters whereby only the basic concepts from above are included due to overview reasons. In Table 1 the concepts are judged among the following criteria:

- required sensors for the damping concept,
- control effort in terms of software implementation,
- use of principle hydraulic control meaning also the used electrohydraulic final control element,
- additional hydraulic hardware for the damping concept and
- energy consumption as far as this information was given by the authors or by the hydraulic principle.

The comparison clearly outlines that the different concepts differ in their advantages and disadvantages. The most important properties for each concept are marked bold. Problematic for the use in mobile machines are the concepts which use more than two sensors or a force sensor. Furthermore, some concepts require a high control implementation effort in combination with difficult reliability of the control. The hydraulic control system relates directly to the primary energy consumption of the machine system also when using active damping. This is still the most critical point of view according to active oscillation damping compared to passive oscillation damping. This is also one of the main reasons why the passive systems have not been replaced by the active systems in many machines up to now. The displacement control of actuators offers many advantages here.

Next to this, it is listed if the damping concept needs additional hydraulic components like pumps and valves for the damping control regarding the standard electrohydraulic machine configuration. Furthermore, the principal energy use for the damping control is judged considering the hydraulic control and additional hydraulic components. This judgement is done quantitatively based on experience with basic hydraulic circuits and by the author´s statements.

Obviously, following the criteria in Table 1 the displacement controlled actuator technology offers the most advantages. Moreover, these advantages can be mostly seen in the minimized energy consumption and the easy controllability (also easy realization of appropriate dynamic characteristics) according to the active oscillation damping and the simple hydraulic system. However, in case of displacement control a fast pump control system is required. Other concepts offer advantages which are focussed more on the maximized damping performance.

5 Conclusions

This review paper summarizes the developments of active oscillation damping concepts in the area of mobile machines by using the machine working hydraulics. Several concepts have been presented in the past basing on different sensor, control and hydraulic strategy.

There are, however, also other concepts partially dealing with active oscillation damping within these machines which have been neglected due to the limited length of this paper.

In fact, considering mobile machine in series production active oscillation damping is used more or less only in agricultural tractor machines. Due to additional costs for components and high control and implementation effort only minor construction machines use active damping today. One reason is surely the inefficient damping quality with today´s valve controlled actuators and another reason is the additional high energy consumption. This opens the way for other actuator solutions like the displacement controlled actuator technology as described in section 4.7. This development, meaning the step from valve control to direct pump control was done in electric systems many years ago (resistance control to direct motor control) and could offer new application fields for fluid power or even strengthen the position of hydraulics against electrics.

Table 1: Comparison of active oscillation damping concepts for mobile machines

Damping Concept	**sensors**	**control effort**	**hydraulic control**	**additional hydraulic hardware**	**energy use**
Damping by Force Control Hesse (1991), Aichele et al (1989)	position, force	filter, cascade	open centre (valve)	**no**	high
Damping by Force Control Maichle (1989)	position, force	var. filter, cascade	fixed displacement pump and valve	**no**	very high
Damping by Force Control Kuromoto (1993), Seiichi (1994)	2 pressures	filter	open centre (valve)	**no**	high
Damping by Force Control Patel et al (1999)	position, force	special filter	variable pump	one valve	moderate
Damping by Pressure Control Hesse (1995), Lödige and Kaplick (1995)	position, pressure	filter, cascade	open centre (valve)	extra valve block (and pump)	high
Damping by Pressure Control Hausman et al (1996)	4 pressures	threshold value	fixed displacement pump and valve	pilot control	very high
Damping by Pressure Control Orbach and Schubert (1996)	position, pressure	threshold value	fixed displacement pump and valve	**no**	high
Damping by Pressure Control Latour and Biener (2002, 2003)	position, pressure	filters, mixture control	load-sensing	**no**	moderate
Damping by Pressure Control Berger and Patel (1999)	position, pressure	filters, observer	load-sensing	**no**	moderate
Damping by Velocity Control Bernzen (1999), Deckers et al (2000)	position, velocity, force	filter, non-linear controller	constant pressure	**no**	very high
Damping by Velocity Control Michalowski and Sobczyk (2003)	position, velocity	filter	fixed displacement pump and valve	**no**	very high
Combined Controls with Valve Yamagata (1993)	position, velocity, 2 pressures	filter	constant pressure	extra control valve	very high
Combined Controls with Valve Kawaguchi et al (1994)	**position**	LQG/LTR control	constant pressure	extra control valve	very high
Combined Controls with Valve Kovanen and Handroos (2002)	position, 2 pressures	filter, differentia-tion	constant pressure	**no**	high
Combined Controls with Valve Andersen and Hansen (2003)	position, pressure or pitch	filter	fixed displacement pump and valve	**no**	high
Damping by Acceleration Control Berger and Patel (1999)	position, accelera-tion	filters, observer	load-sensing	**no**	moderate
Damping by Acceleration Control Frediani et al (2004)	position, accelera-tion, pres-sure	filters, load compensa-tion	load-sensing	**no**	moderate
Damping by Displacement Control (pump control) Rahmfeld and Ivantysynova (2003)	position, accelera-tion	filter	**closed hydraulic circuit**	**no**	**low**

Nomenclature

a	acceleration	[m/s^2]
A	cylinder piston area	[m^2]
c	spring rate	[N/m]
d	damper parameter	[kg/s]
D	damping	[-]
F	force	[N]
G	transfer function	[ ]
h	length	[m]
k_F	force gain	[m^3/s/N]
k_v	velocity gain	[m^2]
J	moment of inertia	[kgm^2]
K	gain	[ ]
l	boom length	[m]
l_c	length	[m]
m	mass	[kg]
M	torque	[Nm]
n	pump speed	[rpm]
p	pressure	[Pa]
Q	volume flow rate	[m^3/s]
s	hitch position	[m]
t	time	[s]
u	control variable	[-]
V_Q	pump flow gain	[m^3/s]
w	commanded position value	[m]
W	weighting factor	[-]
x	cylinder position	[m]
z	pump position	[%]
α	single rod cylinder area ratio	[-]
θ	constant boom angle	[rad]
ϕ	angular position	[rad]
ω	break-point frequency	[rad/s]
ω_0	eigenfrequency	[rad/s]

References

Aichele, H., Hesse, H., Höfer, F.-W., Mohaupt, H. and **Schumacher, W.** 1989. Fahrzeug mit einem System zur dynamischen Stabilisierung seiner Bewegungen. *EP Patent No. 0.299.223; DE Patent No. 3.723.488*. (in German)

Altmann, U., Hesse, H. and **Lödige, H.** 1992. Landwirtschaftlich nutzbare Zugmaschine mit angelenktem Hubwerk. *EP Patent No. 0.518.226; DE Patent No. 4.119.414.*

Andersen, T. O. and **Hansen, M. R.** 2003. Active Damping of Oscillations in Off Road Vehicles. *The 8th Scandinavian International Conference on Fluid Power (SICFP'03)*, pp. 1073-1085, Tampere, Finland.

Anderson, W. R. 1988. *Controlling electrohydraulic systems*. New York: Marcel Dekker Inc.

Asokan, T. and **Singaperumal, M.** 1995. Damping variation through electrohydraulic servosystem: an application to active impedance control. *Mechatronics*, Vol. 6, No. 6, pp. 613-630.

Backé, W. 2000. Concerning: Research and Development in Fluid Power Technology. *1st FPNI-PhD Symposium Hamburg 2000*, pp. 9-21, Hamburg.

Beck, H.-P. 1999. Aktive Schwingungsdämpfung in Triebachsen mittels selbsteinstellender Zustandsregelung. *eb – Elektrische Bahnen*. 97, Nr. 12, pp. 393-401. (in German)

Berg, H. and **Ivantysynova, M.** 1999. Design and testing of a robust linear controller for secondary controlled hydraulic drive. *Journal of Systems and Control Engineering. Special Issue: Control in Fluid Power Systems*, Vol. 213, Part I, pp. 375-385.

Berger, A. D. and **Patel, K. B.** 1999. Active Ride Control System for Offroad Vehicles. *US Patent No. 5.897.287.*

Bernzen, W. 1999. *Zur Regelung elastischer Roboter mit hydrostatischen Antrieben.* Dissertation Gerhard-Mercator Universität Duisburg. (in German)

Bernzen, W. 1998. Active Vibration Control of Flexible Robots Using Virtual Spring-damper Systems. *Journal of Intelligent and Robotic Systems*, Vol. 24, No.1, pp. 69-88.

Bindel, R. and **Mähner, K.** 1997. Rechnergestützte Dimensionierung des Hydraulikantriebes von Großmanipulatoren mit Knickarmkinematik. *Ölhydraulik und Pneumatik*, 41, Nr. 6, pp. 417-420. (in German)

Bittner, O., Leidinger, G. and **Weber, J.** 2001. Arbeitsmaschine. *International Patent No. WO 01/69089 A2.*

Blackburn, J. F., Reethof, G. and Lowen Shearer, J. 1960. *Fluid power control.* Technology Press of M.I.T. & John Wiley and Sons, USA.

Bonefeld, R. 2000. Einsatz elektrohydraulischer Antriebssysteme zur aktiven Dämpfung seilgetragener Bauwerke. *2.IFK (2. Int. Fluidtechnisches Kolloquium)*, pp. 81-96. Dresden. (in German)

Bonefeld, R. 2002. *Einsatz elektrohydraulischer Antriebssysteme zur aktiven Dämpfung mechanischer Strukturen.* Dissertation TU Dresden.

Bonefeld, R. 2003. Einsatz elektrohydraulischer Antriebssysteme. *Ölhydraulik und Pneumatik*, 47, Nr. 9, pp. 592-598. (in German)

Brönner, G. and **Kolb, W.** 1983. Hydraulikanlage für mit Arbeitsgeräten versehene Arbeitsmaschinen. *DE Patent No. 4.221.943.* (in German)

Burrows C. R., Edge K. A. and **Vaughan, N. D.** 1990. Some applications of hydraulics for control of vibrations and machines. *Proceedings of the Second International Technology-Transfer Congress*, pp. 68-72, Bremen, Germany

Burrows, C. R. and **Keogh, P. S.** 1994. The active control of vibration. *International Union of Theo-*

retical and Applied Mechanics: IUTAM Symposium, Bath, UK.

Campos, J., Davis, L., Lewis, F. L., Ikenga, S., Scully, S. and **Evans, M.** 1999. Active Suspension Control of Ground Vehicle Heave and Pitch Motions. *The 7th IEEE Mediterranean Conference on Control and Automation*, Haifa, Israel.

Deckers, J., Nissing, D. and **Schwarz, R.** 2000. Erprobung neuer Regelungskonzepte zur aktiven Schwingungsdämpfung am Beispiel eines Schmiederoboters. *Mechatronik – Mechanisch/Elektrische Antriebstechnik*. Vol. 1533, pp. 465-484. (in German)

Crolla, D. A. 1996, Vehicle dynamics - theory into practice. *Journal of Automobile Engineering*, *IMechE, Part D,* Vol. 210, No. D2, pp 83-94.

Cundiff, J. S. 2001. *Fluid Power Circuits and Controls - Fundamentals and Applications.* CRC Press.

Drake, F. J. and **Jaecks, B. K.** 1991. *Hydraulic System for a Wheel Loader.* US Patent No. 5.195.864.

Edge, K. 2003. On the Control of Electrohydraulic Systems - Some Recent Research Contributions. *The 8th Scandinavian International Conference on Fluid Power (SICFP'03)*, pp. 31-59, Tampere, Finland.

Esders, H. 1996. *Elektrohydraulisches Load Sensing für mobile Anwendungen*. Dissertation TH Braunschweig, VDI-Fortschritt Berichte Vol. 14, No. 75, ISSN 0178-9570, Düsseldorf: VDI-Verlag. (in German)

Franklin, G. F., Powell, J. D. and Workman, M. 1997. *Digital Control of Dynamic Systems.* Menlo Park: Addison Wesley Longman.

Frediani, S., Gianoglio, R. and **Weber, J.** 2003. Electro Hydraulic Active Ride Control (Aktive elektrohydraulische Niveauregelung). *4. IFK (4th Int. Fluid Power Conference)*, pp. 129-138, Dresden, Germany.

Freedy, A. L. and **McIndoo, R. G.** 1964. Hydraulic Circuit for Tractor mounted Loaders. *US Patent No. 3.122.246.*

Fuller, C. R., Elliott, S. J. and **Nelson, P. A.** 1996. *Active Control of Vibration.* London: Academic Press.

Gaudreault, M. L. D., Liebst, B. S. and **Bagley, R. L.** 1993. Simultaneous Design Of Active Vibration Control And Passive Viscous Damping. *Journal of guidance, control and dynamics*. Vol. 16, pp. 654-661.

Göhlich, H. and **Hesse, H.** 1986. Landwirtschaftlich nutzbare Zugmaschine mit angekoppeltem Arbeitsgerät. *DE Patent No. 3.446.811.* (in German)

Gu, L. and **Wang, Q.** 2001. Research on Noninterference Control of Multiple Actuators Load Sensing System with High Inertia in Construction Machinery. *5th International Conference on Fluid Power Transmission and Control (ICFP'2001),* Hangzhou, China, pp. 373-377.

Guglielmino, E., Stammers, C. W. and **Edge, K. A.** 2000. Robust force control in electrohydraulic friction damper systems using a variable structure scheme with non linear state feedback. *Proc. of the 2nd International Fluid Power Conference (2. IFK),* Vol. 1, pp. 163-176, Dresden, Germany.

Hagemeister, W. and **Keuper, G.** 2002. Fahrzeug mit einer Regeleinrichtung zur Schwingungsdämpfung. *DE Patent DE10039219.* (in German)

Harms, H.-H. 2001. Mobile Hydraulic Technology – Current Problems and Future Challenges. *The 7th Scandinavian International Conference on Fluid Power (SICFP'01)*, pp. 21-36, Linköping, Sweden.

Hansen, M. R., Andersen, T. O. and **Conrad, F.** 2002. Control of oscillations in hydraulically driven off-highway vehicles. *Bath Workshop PTMC2002*, Bath, UK, pp. 81-94.

Hansen, M. R. and **Andersen, T. O.** 2002. Active Damping of Oscillations in Off Road Vehicles. *15th Nordic Seminar on Computational Mechanics*, pp. 163-166, Aalborg, Denmark.

Hansen, M. R., Andersen, T. O., Nielsen, B. and **Conrad, F.** 2002. Electro Hydraulic Hitch Control. *The 18th International Conference on Hydraulics and Pneumatics (ICHP03),* pp. 326-337, Prague, Czech Republic.

Hauck, M. 2001. *Geregelte Dämpfung für Traktor-Fahrersitze (Controlled damping for driver's seats of agricultural tractors)*. Dissertation Technical University of Berlin. (in German) http://edocs.tu-berlin.de/diss

Hausman, D. J., Krone, J. J. and **Cline, M. I.** 1996. Method for Controlling Bounce of Work Implement. *US Patent No. 5.622.226.*

Heiskanen, K., Hakola, J., Laitinen, L. and **Pietola, M.** 2003. Vibration Suppression with Hydraulic Actuator. *The 8th Scandinavian International Conference on Fluid Power (SICFP'03)*, pp. 523-531, Tampere, Finland.

Heiskanen, K., Juhala, J. and **Pietola, M.** 2003. Active Structural Control of Hydraulically Supported Machinery. *The 1st International Conference on Computational Methods in Fluid Power Technology (sim2003)*, Melbourne, Australia.

Heiskanen, K., Hakola, J., Laitinen, L. and **Pietola, M.** 2003. Active structural control using hydraulic actuator. *Bath Workshop PTMC2003*, pp. 119-129, Bath, UK.

Helduser, S. and **Bonefeld, R.** 2001. Active Damping In Civil Engineering Using Hydraulic Actuators. *5th International Conference on Fluid Power Transmission and Control (ICFP'2001)*, pp. 259-263. Hangzhou, China.

Hesse, H. 1991. Digitale elektronische Hubwerksregelung für Ackerschlepper. *Ölhydraulik und Pneumatik*, 35, Nr. 11, pp. 828-835. (in German)

Hesse, H. 1995. Active oscillation damping for off road vehicles. *The 4th Scandinavian International Conference on Fluid Power (SICFP'95)*, pp. 1152-1172, Tampere, Finland.

Hirowatari, Y. and **Ichiryu, K**. 2001. Active Vibration Control By Tracking Filter. *5th Interntional Conference on Fluid Power Transmission and Control (ICFP'2001)*, pp. 33-36 Hangzhou, China.

Höfer, F.-W., Schumacher, W., Mohaupt, H. and **Hesse, H.** 1991. Landwirtschaftliche Zugmaschine mit einem System zur dynamischen Stabilisierung seiner Bewegungen. *DE Patent No. 3.942.057.* (in German)

Hosseini, J. 1992. Automatic Ride Control. *US Patent No. 5.147172.*

Imanishi, E. 1988. Controller for Attachment of Oil-Pressure Shovel. *JP Patent No. 6.313.802.4.*

Ivantysyn, J. and **Ivantysynova M.** 2000. *Hydrostatic Pumps and Motors - Principles, Designs, Performance, Modelling, Analysis, Control and Testing.* Academica Books International, New Delhi.

Ivantysynova, M. 2002. Displacement Controlled Actuator Technology - Future for Fluid Power in Aircraft and other Applications. *Proc. of the 3rd International Fluid Power Conference (3. IFK),* Vol. 2, pp. 425-440, Aachen, Germany.

Jelali, M. and Kroll, A. 2002. *Hydraulic Servosystems*. Heidelberg: Springer Verlag.

Kauss, W., Mucheyer, N. and **Wittich, K.** 1984. Vehicle with hinged Attachment Implement. *US Patent No. 4.667.744.*

Kauss, W., Mucheyer, N. and **Wittich, K.** 1985. Fahrzeug mit einem gelenkig befestigten Anbaugerät. *DE Patent No. 3.346.892.*

Kawaguchi, M., Maekawa, A., Kobayashi, N. and **Yoshimura, T.** 1994. Study on Active Vibration Control of Arm for Construction Machinery (Modeling and Linear Control Simulation). *JSME International Journal*, Vol. 37, No. 1, pp. 678-684.

Kliffken, M. G. and **Geerling, G.** 2003. New Developments and Trends in Hydraulics for Mobile Applications. *The 8th Scandinavian International Conference on Fluid Power (SICFP'03)*, pp. 77-88, Tampere, Finland.

Kummer, W. 1996. Reduzierung der Schwingungsbelastung von Radladerfahrern durch die Schwingungsdämpfung der Hubhydraulik. *Tiefbau: amtliches Mitteilungsblatt der Tiefbau Berufsgenossenschaft*. Ausgabe 1, pp. 16-18. Tiefbau-Berufsgenossenschaft Berlin. (in German)

Kovanen, J. and **Handroos, H.** 2002. Active Vibration Reduction Method for a Hydraulic Driven Flexible Manipulator. *ICMA2002*, Tampere, Finnland.

Kurohashi, M., Torii, M. and **Kunieda, Y.** 1992. Vibration Suppression Device for Wheeled Construction Equipment. *US Patent No. 5.116.188.*

Kuromoto, K. 1993. Oscillation Suppressing Device for Working Machine. *JP Patent No. 5.163.746.*

Latour, Ch. and **Beck, J.** 1999. Fahrantrieb und Arbeitshydraulik für einen Radlader. *VDI-MEG Tagung Landtechnik*. pp. 379-384. Braunschweig. (in German)

Latour, Ch. and **Biener, R.** 2002. Comparison of active and passive oscillation suppression systems for wheel loaders. *3. IFK (3rd Int. Fluid Power Conference),* T4 Vol. 2, pp. 101-112, Aachen.

Latour, Ch. and **Biener, R.** 2003. Schwingungstilgung in Radladern – Vergleich von aktiven und passiven Systemen. *Ölhydraulik und Pneumatik*, 47, No. 3, pp. 171-174. (in German)

Leidinger, G. 1992. Laststabilisator steigert die Umschlagleistung von Radladern. *Fördern und Heben*. 42, No. 6, pp. 476-478. (in German)

Lizell, M. 1988. Semi-Active damping. *Advanced Suspensions Conference*, pp. 83-91, London, UK.

Lizell, M. 1994. An Actuator For Semi-Active Damping System. *Mechatronics*. Vol. 4, No. 2, pp. 207-215.

Long, T., Brennan, M. J. and **Elliott, S. J.** 1998. Design of Smart Machinery Installations to Reduce Transmitted Vibrations by Adaptive Modification of Internal Forces. *Journal of Systems and Control Engineering*, Vol. 212, No. 3, Part I, pp. 215-228.

Lödige, H. and **Kaplick, Ch.** 1995. Aktive Schwingungsdämpfung für ungefederte Arbeitsmaschinen. *Ölhydraulik und Pneumatik*, 39, No. 3, pp. 188-192. (in German)

Maczynski, A. and **Wojciech, S.** 2003. Dynamics of a Mobile Crane and Optimisation of the Slewing Motion of Its Upper Stucture. *Nonlinear Dynamics 32 (Kluwer Academic Publishers),* pp. 259-290.

Maichle, J. 1989. Landwirtschaftlich nutzbare Zugmaschine mit angekoppeltem Arbeitsgerät. *DE Patent No. 3.816.166.* (in German)

Maichle, J. 1990. Agricultural Pulling Machine with Automatic Pitch Vibration Damping Mechanism. *US Patent No. 4.924.943.*

Merkel, M., Gojny, M. H. and Carl, U. B. 2002. Enhanced Eigenstructure Assignment for Aeroelastic Control Application. *Deutscher Luft- und Raumfahrtkongress 2002*, DGLR-2002-007, Stuttgart, Germany.

Merritt, H. E. 1967. *Hydraulic Control Systems.* New York: John Wiley & Sons.

Michalowski, St. 2000. Application of Fluid Power Devices in active vibro-control systems. *International Scientific Forum on Developments in Fluid Power Control of Machinery and Manipulators*, pp. 281-328, Krakau, Polen.

Michalowski, St. and **Sobczyk, A.** 2003. Dynamic Effects in Mobile Manipulators with Hydraulic Systems. *The 1st International Conference on Computational Methods in Fluid Power Technology (sim2003)*, Melbourne, Australia.

Motohashi, K. and **Ichiryu, K.** 2003. Research of Active Vibration Control by Hydraulic Device. *4th International Symposium on Fluid Power Transmission and Control (ISFP2003)*, pp.565-568, Wuhan, China.

Murrenhoff, H. and **Wallentowitz, H.** 1998. *Fluidtechnik für mobile Anwendungen.* RWTH Aachen, Verlag Mainz, Wissenschaftsverlag Aachen. (in German)

Noak, S. 2001. *Hydraulics in Mobile Equipment*. Bosch-Rexroth Group Publication, Ditzingen (Germany): OMEGON Fachliteratur; Warendale (USA): SAE Publication.

Norvelle F. D. 1999. *Electrohydraulic Control Systems*. New Jersey: Prentice-Hall.

Orbach, A. and **Schubert, W. L.** 1996. Active Roadability Control for Work Vehicles. *US Patent No. 5.884.204.*

Patel, K. B., Lin, H. C., Berger, A. D., Farag, W. and **Khan, A. A.** 1999. EDC Draft Force Based Ride Controller. *US Patent No. 6.196.327.*

Pietola, M., Kajaste, J., Heiskanen, K., Ellman, A., Porkka, E., Valkonen, A. and **Järviluoma, M.** 2003. Active Vibration Control of a Large Scale Rotor with a Servohydraulic Control System. *The 8th Scandinavian International Conference on Fluid Power (SICFP'03)*, pp. 1201-1208, Tampere, Finland.

Rahmfeld, R. 2002. *Development and Control of Energy Saving Hydraulic Servo Drives for Mobile Systems*. Doctoral thesis, Vol. 12, No. 527, Düsseldorf: VDI-Verlag.

Rahmfeld, R. and **Ivantysynova, M.** 2003. New Displacement Controlled Linear Actuator Technology – a Suitable Control Element for Active Oscillation Damping. *The 8th Scandinavian International Conference on Fluid Power (SICFP'03)*, pp. 1139-1154, Tampere, Finland.

Rahmfeld, R., **Ivantysynova, M.** and **Weber, J.** 2003. IBIS – Advanced Multi-functional Machinery for Outdoor Applications. *The 18th International Conference on Hydraulics and Pneumatics (ICHP03),* pp. 137-149, Prague, Czech Republic.

Rahmfeld, R., Ivantysynova, M. and **Weber, J.** 2004. Displacement Controlled Wheel Loader – a simple and clever Solution (Verdrängergesteuerter Radlader – die einfach bessere Lösung). *4. IFK (Int. Fluid Power Conference)*, pp. 183-196, Dresden, Germany.

Saotome, Y. 1991. Mechanism for suppressing vibrations of travelling crane. *US Patent No. 4.995.517*

Sasaki, K. 2003. Semi-active Suspension System. *Railway Technology Avalanche*, No. 1, Jan 2003, pp. 5.

Seiichi, M. 1994. Device and Method for Suppressing Vibration of a Working Machine. *EP Patent No. 0.747.797.*

Stecki, J. S. and **Garbacik, A.** 2002. *Design and Steady-state Analysis of Hydraulic Control Systems.* Fluid Power Net Pty Ltd.

Stecki, J. S. and **Garbacik, A.** 2002. *Development of Fluid Power Control of Machinery and Manipulators.* Fluid Power Net Pty Ltd.

Streiter, R., Riege, B., Boller, M., Schneider, R. and **Himmelstein, G.** 2001. Active Lateral Suspension for High-Speed Trains. *World Congress on Railway Research 2001*, Cologne, Germany.

Sunamura, K., Tomikawa, O. and **Hirata, T.** 1990. Hydraulic system for boom cylinder of working apparatus. *EP Patent No. 0.378.129.*

Sunamura, K., Tomikawa, O. and **Hirata, T.** 1990. Anti vibration apparatus in a hydraulic system for boom cylinder of working apparatus. *US Patent No. 5.048.296.*

Ufheil, S. and **Rector, S. W.** 1996. Programmable Ride Control. *US Patent No. 5.520.499.*

Ulrich, A. 1980. Fahrzeug mit angekoppelten und/oder festangebauten Heck- und/oder Frontanbaugeräten und/oder seitlich angebaute Geräte. *DE Patent No. 2.856.583.* (in German)

Walters, R. B. 2000. *Hydraulic and electric-hydraulic control systems*. Norwell, Dordrecht: Kluwer Academic Publishers.

Weber, J. 2000. Funktionelle Flexibilität durch Elektrohydraulik am Radlader. *2. IFK (2nd Int. Fluid Power Conference)*, Dresden, Germany, pp. 65-73. (in German)

Yamagata, E. 1993. Vibration Suppressing Control Device for Operation Device of Hydraulic Working Machine. *JP Patent No. 5.321.297.*

Zähe, B., Linden, D. and **Gawlikowski, R.** 1993. Load-Sensing als Mehrgrößenregelung. *Ölhydraulik und Pneumatik* 37 (1993) No. 8.

Zhang, Y. and **Alleyne, A.** 2003. A Simple Novel Approach to Active Vibration Isolation with Electrohydraulic Actuation. *ASME Journal of Dynamic Systems, Measurement and Control*, Vol. 125, pp. 125-128.

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