Vibration Damping System for ALMA Antenna Transporters

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ABSTRACT

The two antenna transporters of the ALMA Observatory are used to relocate 12-meter antennas and to move them between low (3000m) and high altitude site (5000m). When analyzing the dynamic response of the transporters to road ripple of critical height and inter-distance, it turned out that transporter accelerations exceeding the seismic accelerations cannot be excluded. This is considered as a remote risk however with possibly catastrophic consequences for the equipped antennas. The problem was analyzed by ESO experts for dynamic simulations and an additional damping system was designed to limit the loads to acceptable values.
This paper describes the design of the additional damping system including its concept, the model-based design using dynamic simulations and verification tests on site.

Keywords: Damping, oscillations, antenna transporter

1. INTRODUCTION

The ALMA Project is an international observatory currently in construction on the high-altitude Chajnantor site in Chile, and composed of 66 high-precision telescopes, operating at sub- to millimeter wavelengths [1]. The ALMA antennas will be electronically combined and provide astronomical observations, which are equivalent to a single large telescope of tremendous size and resolution. The telescopes can be moved across the high-altitude desert plateau, covering antenna configurations as compact as 150 meters to as wide as 15 kilometers. Changing the relative positions of the antennas allows for different observing modes, comparable to using a zoom lens on a camera.
ESO is responsible for the supply of two antenna transporters [2] of the ALMA Observatory. They are used to relocate antennas and to move them between the low site at 3000m, where they are assembled and maintained, and the high altitude site at 5000m altitude, where the observation takes place. The transport is carried out on unpaved roads, which are regularly maintained. Each transporter has a mass of app. 130 tons and is able to lift and transport antennas of up to a weight of 115 tons. The transporters have to position the antennas on the docking pads with precision in the millimetre range. On the other hand, the transporters must be powerful enough to reliably and safely climb up to Chajnantor plateau with their heavy and valuable load, putting extraordinary demands on the vehicles and the two 500 kW diesel engines.

Figure 1 ALMA transporter on the unpaved road towards the Chajnantor Plateau.

The design and manufacturing was contracted to the company Scheuerle that is specialized in heavy load transport

vehicles [3]. At final design of the transporters in 2007, when analyzing the dynamic response of the ALMA transporters to road ripple of critical height and inter-distance, it was found that transporter accelerations exceeding the seismic accelerations (used as critical design parameters) cannot be excluded. This was considered as a remote risk however with possibly catastrophic consequences for the transporter and the equipped antennas.
The problem was analyzed by ESO experts for dynamical systems and a solution was found to limit the loads to acceptable values. For this purpose the transporter dynamics and the non-linear transporter hydraulic system were simulated and a possible modification of the hydraulic suspension system was proposed, specifying the characteristics of the critical components. After some breadboard tests of the main components, both transporters were equipped with an additional damping system by Scheuerle in their factory, consisting of nitrogen charged accumulators and throttle valves for each wheel pair as well as an automatic emergency brake system that brings the vehicle to a safe stop in critical situations independent of the driver. End of May 2009 both safety systems where successfully adjusted and tested at ALMA OSF in Chile. Since then both transporters are in daily operation with the activated damping systems.
This paper describes the additional damping system from its concept to the model-based design using dynamic simulations and verification tests. Additionally, the paper summarizes the possible additional hazards. Even though the damping concept is introduced in the context of the ALMA antenna transporters, it is applicable to other heavy load damping applications.

2. PROBLEM STATEMENT

2.1 Concern

At final design review of the transporter a concern was raised that critical stress values could be reached or exceeded by transporter oscillations cause by road imperfections. The main area of concern was in the antenna interface. Therefore, the configuration with loaded antenna was analyzed. The unloaded vehicle can absorb higher accelerations which are assumed to be noticed by the driver and the speed is reduced by operator's corrective actions.

2.2 Requirements

The damping system design described in this paper is derived from the following top level requirements:

Acceleration Limit The positive vertical transporter (and antenna) accelerations shall be limited to the Maximum-Likelihood-Earthquake level (ca. 0.4g) in all operational conditions.

Worst Case Perturbation The design should be compatible with the worst case road profiles to be considered, which have amplitudes of 1.29cm (2.54 com as a goal) and excite the main vehicle resonant mode at 2 Hz.

Design Constraints (Parts) Passive components shall be used if possible, mainly for maintainability and reliability, but also in order to minimize the risk of instabilities.

Design Constraints (Operation) The impact on regular transporter operation shall be kept as low as possible (e.g. as few emergency stops as possible, simple recovery from emergency situations).

Design Constraints (Volume) Additional parts needed for the damping system shall fit the interfaces and reserved volumes foreseen in the transporter design.

2.3 Assumptions

The following vehicle properties were considered for the damping system analysis:

Weight The vehicle has a weight of 128 tons. ALMA uses 3 types of 12m-antennas with different weights in the range of 85 up to 115 tons. The damping system was conceived to work with the same settings for all of them.

Hydraulic Support System The transporters are equipped with 14 identical running gears with hydrostatic drives and suspension equally spaced at a distance of about 2.3 m on the vehicle chassis. During antenna transport the hydrostatic suspensions are linked within 3 groups, 2 groups of 4 wheel pairs in the front and 1 group of 6 wheel pairs in the back on the vehicle, respectively. Since all support points within one group are equally charged the transporter can absorb road ripples on individual wheel pairs. The overall support systems acts like a 3 point support.

Figure 2 Transporter with 14 running gears, ©ESO

Structural Dynamics The mechanical properties and kinematics were made available in form of a complete FE model. Usually, tire characteristics have to be estimated. However, in this particular case data from the tire manufacturer, concerning tire stiffness and damping, were available. The elastic behavior of the vehicle including tires was analyzed and it was recognized that in the load case, where all wheels are synchronously excited in vertical direction in the vehicle main resonant mode of 2Hz, the critical accelerations and thus the admissible stresses in some parts are largely exceeded.

Road Characteristics Although the transporter roads on Chilean site were already partly constructed, no reliable data about typical washboard profiles were available. The typical road use and maintenance as can be expected during ALMA observatory operation had not been started at the time of the design review. As a substitute the requirements from the NASA road roughness classification "3510 NASA SP-8077" table II [4] have been selected.

Table 1 NASA Road Roughness Classification

Type of road encountered	Low probability of occurrence (a)		High probability of occurrence (b)	
	Sinusoidal washboard amplitude, cm (c) (d)	Single bump amplitude, cm (c) (d)	Sinusoidal washboard amplitude, cm (c) (d)	Single bump amplitude, cm (d)
Primary and secondary roads with rigid or flexible pavement that are well constructed and maintained	1.91	3.81	0.95	2.54
Paved primary and secondary roads with average maintenance; or well constructed unpaved roads with good maintenance	2.54	5.08	1.29	3.81
Secondary roads that are flexibly paved and poorly maintained, and unpaved roads	5.08	10.16 (chuckhole)	3.81	7.62 (chuckhole)

The amplitude value 1.29 cm corresponding to "high probability" and "well constructed unpaved roads with good maintenance has been selected as requirement. The amplitude value 2.54 cm corresponding to "low probability" and "well constructed unpaved road with good maintenance" has been selected as goal. Assuming an average daily drive distance of a loaded transporter of 50 km, the "high probability" washboard amplitude would occur approximately once per year per transporter, and the "low probability" washboard amplitude would occur less than every ten years per transporter.

A washboard of critical wavelength and length is assumed, which means in case of the transporter that all 28 wheels move synchronously up and down. In this case the compensating effect of the hydrostatic support is neglected, which would reduce the vehicle amplitude compared to the road amplitude and provide additional damping. The assumption that the washboard is completely synchronous to the 28 wheels reduces further the probability of occurrence of a critical resonance case. Integrating effects of the ca 40 x 40 cm tire footprint are neglected and the amplitude is assumed as vertical movement at the axle centreline.

2.4 Load Limit

With the FE model from the antenna manufacturer Scheuerle a dynamic analysis of the vehicle was performed imposing sinusoidal vertical movements (large mass approach) with different frequencies and amplitudes corresponding to the typical road profile defined above. The calculations were performed with excitation of the wheels of one side only to generate lateral movements and also with all the wheels excited synchronously on both sides.
For lateral excitation the accelerations remain below the MLE limits and additional damping is not required.
For vertical excitation in the eigenmode (2Hz) the vehicle accelerations reach values larger than 1g, which are too high and need to be reduced. Additional damping is required in order to bring the level of acceleration down to 0.4g.
All calculations are based on a tire damping ratio of 5% as given by the tire manufacturer. However, even for a tire damping of 20% additional damping would still be required.

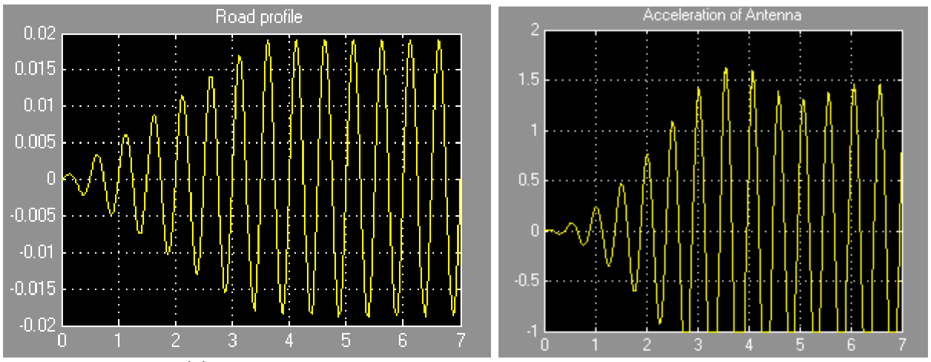

Figure 3 Transporter Response without additional damping system.
Left figure: Road profile in [m], right figure: antenna accelerations exceeding specification in [g]

2.5 Critical Drive Speed

The critical resonance case (2 Hz) can be reached at different vehicle speeds.

Table 2 Critical Combination of Road Geometry and Drive Speed for Resonance Case.

Washboard Wavelength	Vehicle Speed
2.35 m	16.92 km/h
1.18 m	8.46 km/h
0.78 m	5.64 km/h
0.59 m	4.23 km/h

The last 3 cases are inside the envelope of the transporter operational parameters and observed washboard characteristics.

2.6 Assessment of Hazardous Situation

As mentioned above, in the 2Hz resonance case the allowable accelerations are largely exceeded and major damage of a transporter with antenna cannot be excluded. This drives the need for an additional damping system. Additionally, it can be assumed that the acceleration profile will disturb the driver and by braking and decelerating he would probably reduce the oscillations. However, since catastrophic consequences cannot be excluded an automatic stopping system was conceived as an additional safety mechanism to protect against 2 independent hardware or human errors.

3. CONCEPT OF THE ADDITIONAL DAMPING SYSTEM

A damping system was proposed consisting of additional passive components in the hydraulic supporting system of the transporter and an automatic vehicle stopping system, which senses critical oscillations and initiates an automatic stop of the vehicle before critical acceleration values are reached.

3.1 Working Principle of the Hydraulic Damping System

The main components of the transporter and the additional damping system are shown schematically in Figure 4. The dynamical behavior of the transporter without additional damping system is dominated by the tire stiffness, the tire damping and the overall mass of the vehicle. The finite stiffness of the hydraulic cylinder can be neglected.

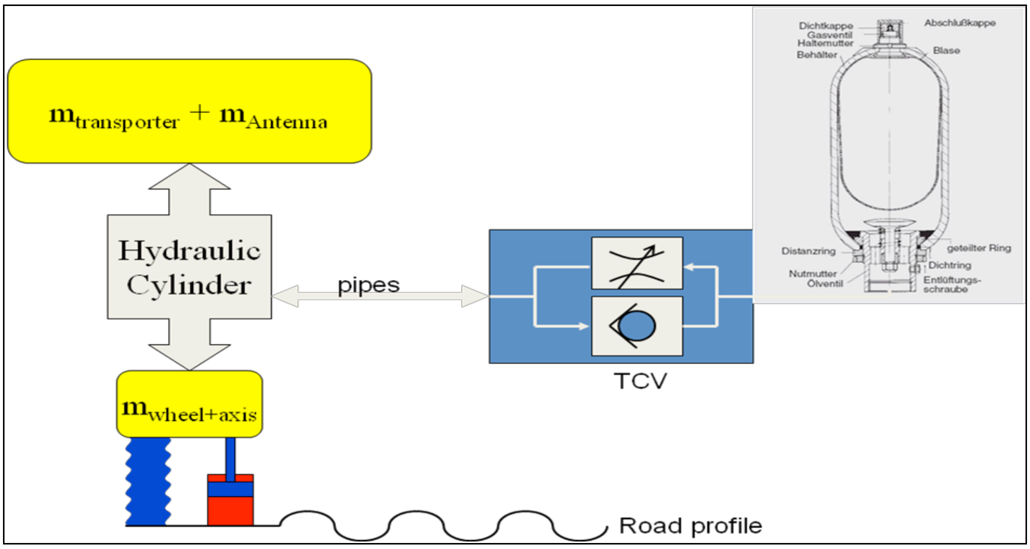

Figure 4 Working principle of the damping system (accumulators [5] and throttle-check valves [6] are added to the transporter)

The principle of the additional damping systems is to limit the forces transmitted from the road to the antenna by changing the oil volume in the hydraulic cylinders that connect wheels and transporter chassis. Oil is released when the force threshold is reached. Technically, the force threshold is imposed by the nitrogen pressure preload of accumulators, which are connected to the hydraulic circuit by check valves (TCV = throttle-check valve). It was shown by simulation that without feeding the released oil back into the hydraulic system the vehicle would be lowered down to the hard range limits of the wheel cylinders in only a few seconds. Therefore, the accumulated oil is returned to the vehicle hydraulics during the low pressure (=low acceleration) regimes. The return flow is limited by the throttle of the TCV.

From a dynamic point of view releasing oil into accumulators is equivalent to a change of stiffness of the hydraulic cylinder (non-linear spring).

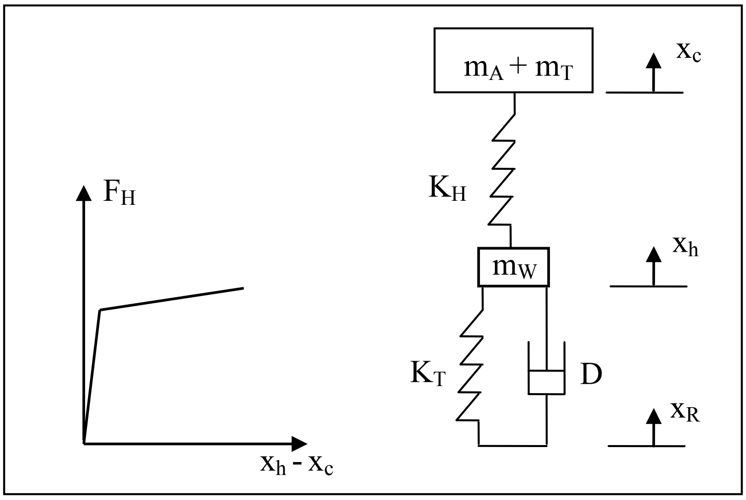

Figure 5 Simplified mechanical model (releasing oil into accumulators changes K_H)

Figure 5 shows a simplified mechanical model of the transporter with the damping system, in which the stiffness of KH is toggled between

I. the stiffness of the closed hydraulic system (if a(mA) < amax) and

II. the stiffness of the nitrogen reservoir of the accumulators (if a(mA) > amax).

As larger the accumulators as smaller are their equivalent spring stiffness. Therefore, the increase in forces acting on the transporter (and thus the acceleration) is very little with respect to differential movement of wheel and transporter.

This effect can also be interpreted as a change of resonance frequency. The original system has its main resonance frequency at about 2Hz. When the damping system becomes active at critical speed the resonance frequency is lowered and brings the system into a non-resonant regime with a significantly smaller gain. Therefore, the system response to the originally resonant road profile is attenuated.

3.2 Dynamic Simulations of the Hydraulic Damping System

The transporter damping system was simulated in Matlab/Simulink. The used models can be split in 3 parts:

Model of the mechanical system Only the vertical piston mode of the transporter was analyzed, because it was the only mode identified to be critical concerning material stresses. The mechanical system is modeled as a 2-mass system, which is excited by the vertical road displacement. All damping and stiffness is concentrated in the tires. The interface between hydraulic cylinders and masses are forces generated by the hydraulic piston model described below. The tire stiffness and damping as well as the relevant inertias are derived from the FE model described in Chap. 2.3. The corresponding Simulink implementation includes an additional saturation management to prevent from negative (pulling) wheel forces.

Model of the vehicle hydraulics The vehicle hydraulics are modeled by one large piston with the surface of all hydraulic wheel cylinders and the whole oil volume of the system. In addition the pipes are represented by one straight pipe for each wheel. Hydraulic wheel cylinder dimensions, oil volumes and estimated pipe dimensions (length and diameter) were obtained from the manufacturing documentation of Scheuerle. For the simulations the oil properties were computed for an ambient temperature of 20 degC and a sensitivity analyses for other temperatures was done for some specific load cases.

The piston model uses the oilflow, the piston position and the piston velocity in order to compute the resulting force transmitted to the interfaced mechanical system. The physical relation used is the compressibility of oil.

The used pipe models consider 3 physical effects:

- The compressibility of oil to model the effect of pressure waves.
- The pipe friction to limit the flow or to generate a pressure drop, respectively.
- The inertial properties of the oil volume in order to simulate dynamic effects of varying flow.

Model of the additional (non-linear) components of the damping system Accumulators are modeled as gas volumes, which can be compressed. The expansion is constraint by the initial volume of the accumulator. As suggested by the supplier's data sheet [5] the pressure change was modeled by the adiabatic gas equation, since the cycle times are well below 2 min.

For the inlet and outlet flow characteristics (flow = f(delta_p)) the throttle-check valve characteristics from the TCV data sheet [6] was applied in form of lookup tables (Figure 6).

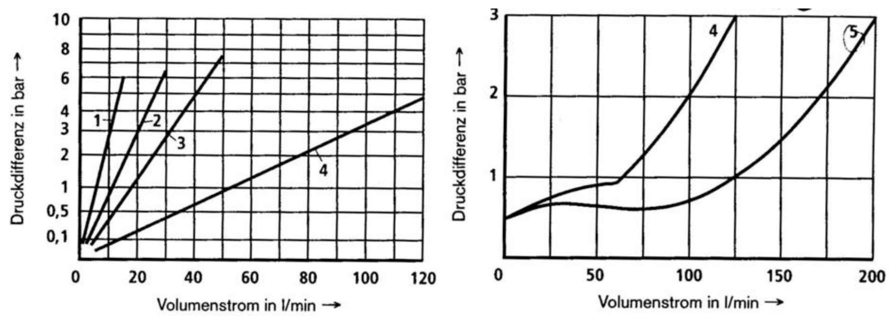

Figure 6 TCV Characteristics from Bosch Rexroth (size 4 was chosen, left throttle, right check valve)

Originally, no special care was taken concerning the hard volume constraint where all the oil is released from the accumulator and the membrane suddenly closes the accumulator outlet. This nonlinearity turned out to be critical in terms of induced hydraulic oscillations and thus raised a major concern. Assuming a linearized characteristics for the last 1% of the accumulator volume seemed to be physically reasonable and reduced these oscillations significantly. This modification was in line with the supplier's expertise and it was confirmed by specific breadboard tests (see Chap. 3.3).

Model of the Road Profile For the simulations the critical washboard profile is generated and stored in a lookup table (LUT) as a time series. In order to simulate build-up and breaking effects the time vector of this LUT is stretched in time.

Tuning parameters The damping system contains 3 tuning parameters:
1. Accumulator volume: It influences the equivalent accumulator stiffness in the damping regime. As larger the accumulator volume as smaller the influence of road amplitudes on the peak accelerations. However, note that the amount of oil volume, which can be absorbed by the accumulators, is limited by the overall oil volume in the system, and there are constraints on the piston dynamical range and dimensional constraints at accumulator interface.
2. Nitrogen pressure: It defines the threshold, where the damping system becomes active. As lower the pressure as lower are the resulting accelerations in the resonant case. However, note that the minimum pressure is limited. Setting this value too low will influence the operational range (e.g. breaking, turns) and can cause oscillation problems at lower frequencies (see section about additional harzards below)
3. Throttle Valve setting: The throttle restricts the amount of return flow during the low-pressure regimes. Therefore, it is influencing the damping of the negative accelerations. As smaller the opening as larger the damping. However, the lower value is limited by the need to return all the oil back into the hydraulics during one period of the road profile. Otherwise, the accumulators will build up a higher pressure threshold.

Performance Analyses System simulations were done in Matlab/Simulatink with stiff solvers in order to handle the non-linearities of the hydraulics. First a basic setting of the tuning parameters described above was evaluated, which could match to commercially available components. These settings were then tested in different load cases in order to verify the robustness of the system with respect to changes in tire damping, variations in the pipe characteristics, antenna mass variations and the sensitivity of the throttle valve settings. Additionally, the impact of an emergency braking system and wrong nitrogen pressure settings were simulated.

Simulation Results It could be shown in simulation that a damping system composed of preloaded accumulators and throttle check valves can limit the transporter accelerations to the specified value and allow to leave the resonant regime safely (see Figure 7).

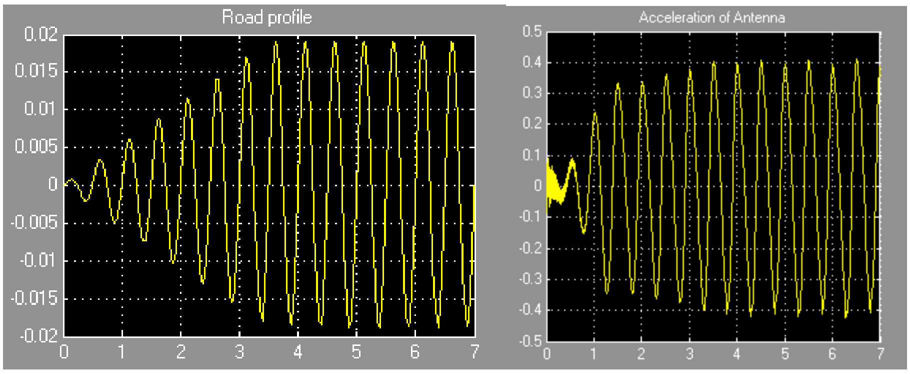

Figure 7 Transporter Response with additional damping system.
Left figure: Road profile in [m], right figure: antenna accelerations below MLE in [g]

It was also shown that an automatic stopping system improves the damping effect by leaving the resonant mode. However, it would not be sufficient by itself. All component parameters used in the simulations were obtained from system documentation and data sheets of the selected components. The simulations showed no major limitations due to the pipe system. Fixed settings for the tuning parameters were working for all operational cases. The throttle valve settings proved to be robust and uncritical.

Additional hazards introduced by the damping system The hydraulic parts of the additional damping system increase the risk of failure. Several load cases were studied, e.g. total loss of oil in one group of wheels at extreme driving conditions (max. downhill, max. acceleration, max. curve) or wrong tuning of the nitrogen pressure (lower than stationary pressure). For all load cases there is no risk of instability, i.e. the transporter will rather slide than fall over. The stresses in all major structural elements remain below critical values.

3.3 Breadboard Test for Accumulator Characterization

Triggered by the concern about induced oscillations mentioned above a bread board was built, where the flow characteristics of the specified accumulators could be studied. Two auxiliary accumulators were used to generate a controlled flow. The flow characteristics turned out to be smooth enough that there is no major concern for the suggested damping system concept.

4. DESIGN OF THE ADDITIONAL DAMPING SYSTEM

For the additional damping system at every boogie an in-line assembly consisting of a manual shut-off valve, a throttle check valve and a preloaded accumulator is connected to the existing junctions (see Figure 8). The hydraulic damping system uses only passive devices.

Figure 8 Additional hydraulic components at each wheel pair

The dynamic analysis described in Chap. 3.2 suggests accumulators of 9.3l nominal volume with a preload of 222@20degC. This nitrogen pressure is higher than the nominal pressure in the loaded system (app. 160 bar). Therefore, no oil is stored inside the accumulators during normal operation. For the throttle check valves a throttle setting of 10-15% was suggested by simulation in order to damp high frequency vibrations in the hydraulic lines. The simulated levels of the preload pressure and the throttle valve settings were confirmed during transporter tests (see Chap. 5).

The manual shut off valves allow isolating the damping system components from the vehicle support hydraulics in cases where the vehicle shall be used without damping system, or for maintenance on the system. In the running system the handles of the isolation valves have been removed. They are part of the maintenance tools of the damping system.

An Automatic Vehicle Stopping System is part of the additional damping system. It detects excessive vehicle oscillations via acceleration sensors on the chassis and pressure sensors in the hydraulic support system and requests an immediate stop of the vehicle.

The automatic vehicle stopping system consists of 3 main types of devices, fast and slow multi-axes acceleration sensors, pressure sensors and the evaluation electronics. Accelerometers from Althen [7] are permanently installed on both sides of the antenna loading system close to the antenna brackets (2 vertical, 1 lateral sensor, marked as Accelerometers 3-4, 5 in Figure 9). They are used to measure vertical and lateral accelerations and to transmit them to the evaluation electronics. In addition, 2 SHOCKLOG acceleration data loggers [8] are installed in the transporters' arms close to the other accelerometers. Each of them logs 3-axis acceleration data in a non-volatile internal memory. Their reaction time is much too slow for the antenna damping system (app. 20s). Therefore, they are only used to record data during antenna transport.

Additionally to the pressure sensors used for transporter control 4 additional identical sensors (one for each group A, B, C, D) were installed in the supporting hydraulic system. These sensors measure the pressures in the support circuits, which are equivalent to the forces introduced into the vehicle via the wheels or road, and transmit them to the evaluation electronics.

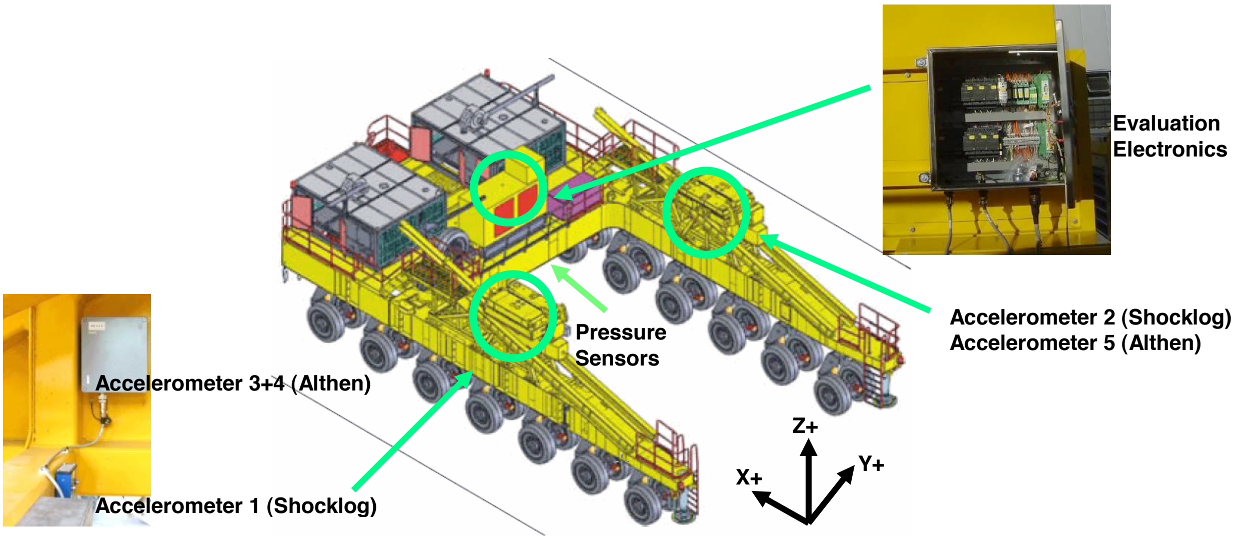

Figure 9 Main components of the Automatic Stopping System

The evaluation electronics for the damping system has been installed in an IP65 sealed housing in the upper central part of the transporter (see Figure 9). The evaluation electronics is based on 2 Siemens LOGO PLCs with flash memory [9], which read in the signals at the interfaces to vehicle and sensors and triggers emergency stops if necessary. The trigger conditions are defined as exceeding individual pressure and acceleration thresholds more than 3 consecutive times within a 1.5 seconds running window for a loaded transporter. This time window is based on the known transporter resonance frequency and prevents from undesirable false alarms at single shock events. The system is only active when an antenna is loaded (automatic detection) and can also be used to force an emergency stop (function used during the test procedures). The evaluation electronics starts automatically after it is connected to the vehicle interface. After an emergency stop has been triggered and the hazardous condition has vanished, the evaluation electronics reset automatically.

5. DAMPING SYSTEM TESTS

During the commissioning of the antenna transporters several tests of components and the complete damping system were performed in order to characterize subsystems, adjust critical parameters and to evaluate the proper functioning of the damping system. The tests had to be planned carefully, because the transporters were operated in conditions where a damping system failure or procedural mistakes could cause major damage.

The first series of tests took place in November 2007 at Scheuerle premises in Pfedelbach, Germany, where critical components were characterized and the test setup for the on-site tests in Chile was prepared. This included the breadboard tests for the accumulators described above, but also measurements of the vehicle dynamics (eigenfrequency and damping) in order to validate the assumption taken for the simulations. The eigenfrequency of 2 Hz in piston direction could be confirmed. However, the damping of the tires turned out to be only 3% instead of the 5 % predicted by the manufacturer, which increases the sensitivity to road ripple.

The second series of tests took place in May 2009 at ALMA OSF, Chile. For the functional tests of the additional damping system the transporters had to be excited in the critical piston resonance with the appropriate amplitude. Two scenarios were considered for the excitation, one driving with the transporters over properly spaced and sized obstacles attached to the ground with the appropriate transporter speed or second imposing a displacement profile to the hydraulic wheel cylinders equivalent to the expected road excitation, using the built-in proportional valves of the transporter hydraulics. The excitation in stand-still was selected as the preferred solution for reasons of safety, simplicity and repeatability.

Figure 10 Antenna Transporter Test Area at ALMA Operation Support Facility, Chile

For the tests the transporters were equipped with additional sensors to analyse the transporters' dynamic response: displacement sensors to measure the elongation of the wheel cylinders, acceleration sensors attached to the transporter chassis and interface electronics to drive the transporter proportional valves and read the transporter pressure sensors. The measurement setup allows a synchronous excitation of the 3 groups of hydraulic cylinders with signals of the same period, but individual characteristics (duty cycle, offset, amplitude) to overcome the differences of the valve characteristics when exciting the transporter in piston direction. For safety reasons in all tests at least 3 persons were involved, one for the measurement system, one for transporter status monitoring outside and one for inside.

Like in Europe the eigenfrequencies and damping of both loaded transporters were measured first by exciting them with the additional measurement setup introduced above close to the expected frequency and then measure the free oscillation. The 2Hz, 3% damping from the pretest could be confirmed.

The main goal of the Chile tests was the demonstration of the positive effect of the additional hydraulic damping. For this purpose the different groups of hydraulic cylinders were excited with synchronous rectangular signals of adapted duty cycles using the measured piston resonance frequency stepwise increasing the amplitude. At the excitation level, where the critical transporter acceleration was reached, the additional damping system started to operate. The excitation level was carefully increased and the transporter acceleration remained limited to the specified limit of 0.4g. In order to further validate the effect of the accumulators individual units were shut off and the effect on the hydraulic pressure was measured. Closing accumulators the damping performance clearly got worse.

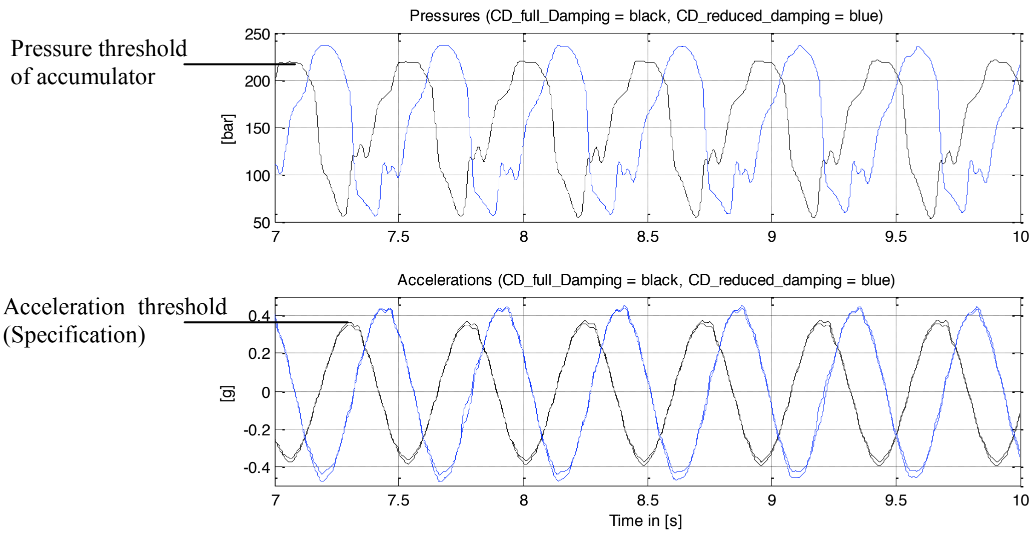

Figure 11 Increase of hydraulic pressures due to partial disable of accumulators, Black = All on, Blue = Some off

Figure 12 shows the saturation effect of the transporter accelerations when approaching and exceeding the critical excitation level. Note that due to the lower tire damping and the averaging effect of the tires the critical accelerations are already reached for excitation amplitudes lower than 2cm. This level of excitation was considered to be sufficient to demonstrate hydraulic damping. Larger excitations were not applied in order to reduce the risk of damage in the transporter hydraulic components. Note that also the excitation was delivered by the transporter.

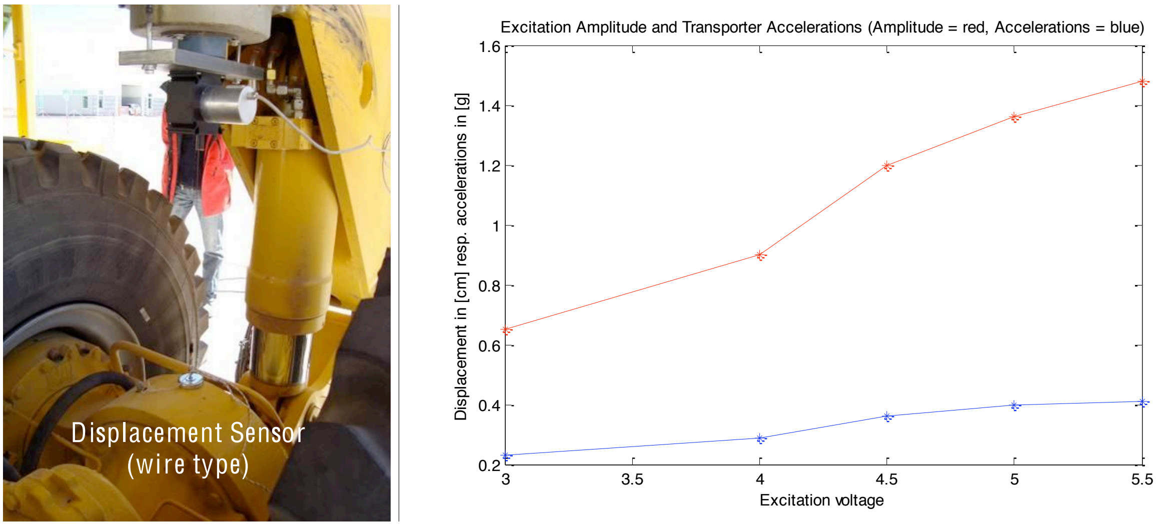

Figure 12 Wheel Cylinder Displacement and Acceleration of Transporter #1 (Otto) as a Function of Excitation Amplitude (Saturation due to Damping System)

During the tests of the hydraulic damping system, the acceleration and pressure values measured by the Automatic Vehicle Stopping System were validated and showed a good match. In measurements with values exceeding the programmed thresholds, emergency stops were generated at the transporter interface. They could be recognized by alarm messages on the transporter operation panel.

In order to analyze the impact of the additional damping system on the transporter characteristics, the system was monitored during a test drive on the road between ALMA high and low site. The antenna dummy was loaded, hydraulic pressures and accelerations were logged. On the way down two brake stops were done, one with the operational brake and one with the park brake.

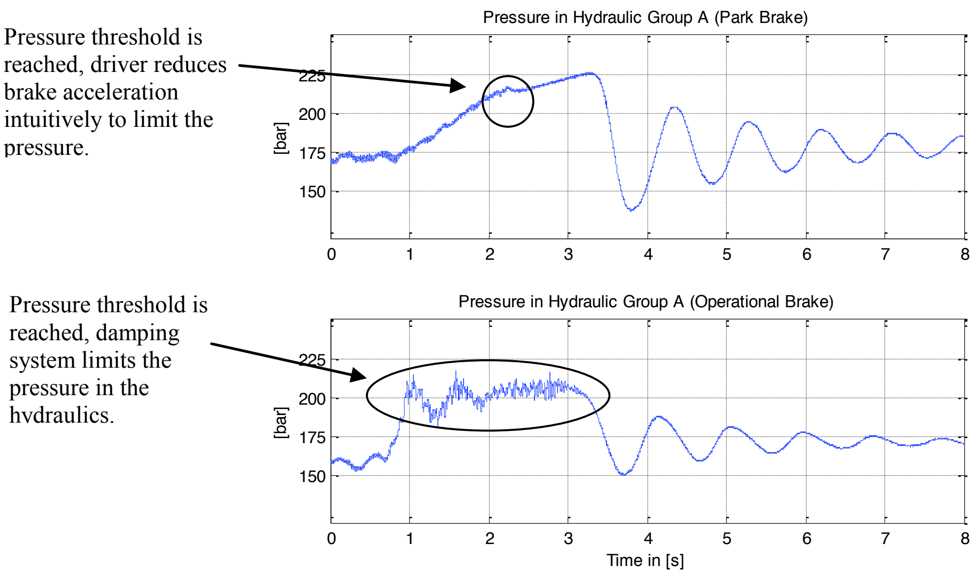

Figure 13 Pressure and Acceleration History during Transporter Brake Test (note: the pressures are measured a few meters away from the running gears, therefore quasi-static pressure changes seem to be higher than fast ones)

No washboard profile with sufficient amplitude could be found on the well maintained ALMA road. The maximum accelerations were well below the activation threshold. No impact of the additional damping system on the transporter characteristics could be recognized during driving and turning as expected.

As shown in Figure 13, during both brake manoeuvres the accumulator threshold is reached. With the operational brake (footbrake) the driver intuitively limited the braking force to maintain the braking acceleration level (the driver reacts mostly to the change of jerk than the acceleration as such). For the park brake, the critical pressure level is exceeded and the system is lowered in group A and B. The amplitude of the level change is estimated to app. 1.5 cm (app. 0.3l of oil). This value is uncritical concerning transporter stability as shown in the Hydraulic Failure Analysis. On the contrary, the additional damping system introduces an extra level of safety to limit excessive pressures, and thus loads, during extreme manoeuvres.

6. CONCLUSIONS

The paper provided an overview of the additional damping system for ALMA transporters. The initial analysis showed the need of a damping system as an additional safety against transporter oscillations caused by typical road disturbances. A design based on passive hydraulic dampers and an additional automatic vehicle stopping system based on acceleration and pressure measurements was proposed. The system was simulated and analyzed in non-linear dynamic simulations using dynamic models of transporters, tires and hydraulics. The transporters were equipped with the necessary components and the parameters were adjusted as predicted by simulations and test beds. The equipped transporters were tested under realistic conditions, using the transporters' own hydraulic system for excitation.

The damping system was working properly limiting the transporter accelerations to MLE level (0.4 g). At critical acceleration and pressure levels a transporter emergency stop was triggered by the Automatic Vehicle Stopping System.

The driving characteristics of the transporters are not affected by the additional damping system.

Since the successful test of the Additional Damping System it is activated for all antenna transports and maintenance drives.

For the first year of operation the acceleration dataloggers are read out regularly. As soon as the roads develop washboard ripples the ripple height and frequency will be monitored and hydraulic pressures in the supporting system will be logged in addition. These data are used for system diagnostics.

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