Noise and vibration reduction for an aerospace secondary controlled hydraulic motor

Emmanuel Viennet*, Anton Gaile**, Tobias Röben**

HES-SO University of Applied Sciences and Arts Western Switzerland, 1700 Fribourg, Switzerland*
Liebherr-Aerospace Lindenberg GmbH, Flight Controls and Actuation Systems,
Pfänderstraße 50-52, Germany**
E-Mail: anton.gaile@liebherr.com

During flight, passenger comfort is affected by noise emissions from various aircraft systems. Apart from jet engines one of the main sources of noise within the fuselage is the power control unit (PCU) for high-lift actuation. In preparation for take-off and landing this hydraulic motor is responsible for the extension and retraction of the slats and flaps. Along with the increase in operating pressure from 206bar (3,000psi) to 345bar (5,000psi) noise and vibration induced by fluid power systems became more striking. Consequently the aim of the BMWI founded research project “Move On” was to reduce the emissions of Liebherr’s power control unit. The results of these research activities are presented within this paper. It is shown how the noise emissions could be reduced in a secondary controlled hydraulic motor by means of a valve-plate and structure optimization. In addition the results of a noise measurement campaign, conducted by Airbus on an A350, are presented.

Keywords: Noise, vibration, hydraulic motor, fluid power systems, axial piston motor, valve-plate optimization
Target audience: Aerospace application, commercial aircraft, pumps & motors

1 Introduction

Fluid power systems are used on Aircraft (AC) since many decades. They drive actuation systems, such as primary flight controls, secondary flight controls, landing gear and utility actuation, e.g. door actuation. Fluid power systems on AC are well proven for excellent reliability, low weight and reasonable cost.

In order to further benefit from its high power density, the working pressure of fluid power systems within some commercial AC was recently raised form 206bar (3,000psi) to 345bar (5,000psi). This shift has made two of the main drawbacks of fluid power systems, namely noise and vibration, even more noticeable. At the same time the noise requirements have become more restrictive. Therefore jet engines as the main source of noise on AC have made advancements in noise reduction. As a consequence, the noise emitted by the fluid power system has become more relevant as regards passenger comfort. One of the components responsible for cabin noise is the hydraulic motor driving the high-lift system. Slats extension and retraction are driven with Slat Power Control Unit (SPCU). The investigated SPCU is a hybrid design and comprises an electric and a hydraulic motor.

Within the BMWI founded research project “Move On” Liebherr-Aerospace Lindenberg GmbH and Liebherr Machines Bulle SA together with Airbus Deutschland GmbH have developed a noise reduced variable displacement hydraulic motor. As a boundary condition for this project the weight of the hydraulic motor should not increase, the envelope should not be affected and the performance characteristics should be equal or better compared to the reference motor.

Accordingly in section 2 the considered type of hydraulic motor and possible means for noise reduction are described in detail.

In a next step several concepts for noise reduction have been analysed with the aid of a simulation models. Results of the simulative analysis are given in section 3.

Subsequently the most promising concepts have been realized within a prototype PCU /2/. This prototype was then tested in a laboratory environment at Liebherr-Aerospace, Lindenberg GmbH in comparison with a reference PCU. Test data is provided in section 4.

Finally a noise and vibration test was conducted inside an AC cabin, which exemplifies the improvements within a real AC environment. Airbus performed these vibro-acoustic measurements on an A350 test AC in July 2015. Respective results are depicted in section 5.

2 Means for noise reduction

In axial piston pumps and motors the timing and the geometry of the swash plate are crucial for noise generation. A careful design of the swash plate is therefore one of the most important ways of reducing noise /1/. However, the performance that can be achieved by only optimizing the swash plate is highly sensitive to any variation in operational conditions. One way to mitigate this sensitivity is to use a design with a cross-angle. Johanson et al. /2/ describe the cross-angle as “a fixed small (~1-4°) displacement angle of the swash plate in the direction perpendicular to the traditional displacement [implying] that the piston top and bottom dead-centres […] vary as functions of the displacement angle, which gives varying cylinder pre-compression and decompression”. While cross-angle designs have been mainly used in pumps, it is also mentioned in the literature as a mean to reduce noise of hydraulic motors /3/.

The swash plate axial piston motor developed by Liebherr takes advantage of both a cross-angle and an optimized valve plate geometry to achieve significant noise reduction. It features a maximum displacement of 16cc/rev that can be adjusted in both positive and negative directions.

Figure 1a : picture of the 16cc/rev Liebherr aerospace hydraulic motor

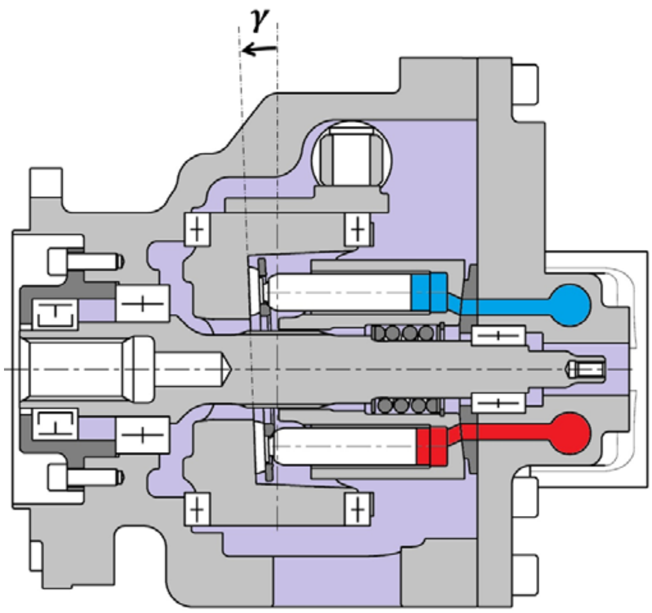

Figure 1b : cross-angle γ machined in the bearing supported swash plate

As shown in Figure 1b, the cross-angle is machined directly in the swash plate of the Liebherr motor. The cross-angle is compatible with both pumping and motoring modes, this as long as the high- and low-pressure ports remain fixed. Actually, if the high and low pressure ports were reversed, the cross-angle direction would have to be reversed as well. The Liebherr motor is therefore particularly well suited for secondary control with one port being supplied by a constant pressure network.

3 Simulation of motor-induced excitations

3.1 Simulation model

The goal of the simulation is to model motor-induced excitations that may lead to noise emissions in order to minimize them. Due to fluid compressibility and dynamic effects taking place in the piston chambers, the pressure profiles are not trivial to describe analytically, therefore a simulation model is used, based on a control volume approach that is common in literature; see for instance /4/ and /5/. The simulation model has been developed with the commercial simulation package Amesim /6/.

From the kinematics of the hydraulic motor, the model first computes the piston stroke y_i and the fluid volume V_i of the ith piston chamber as a function of angular position ϕ_i.

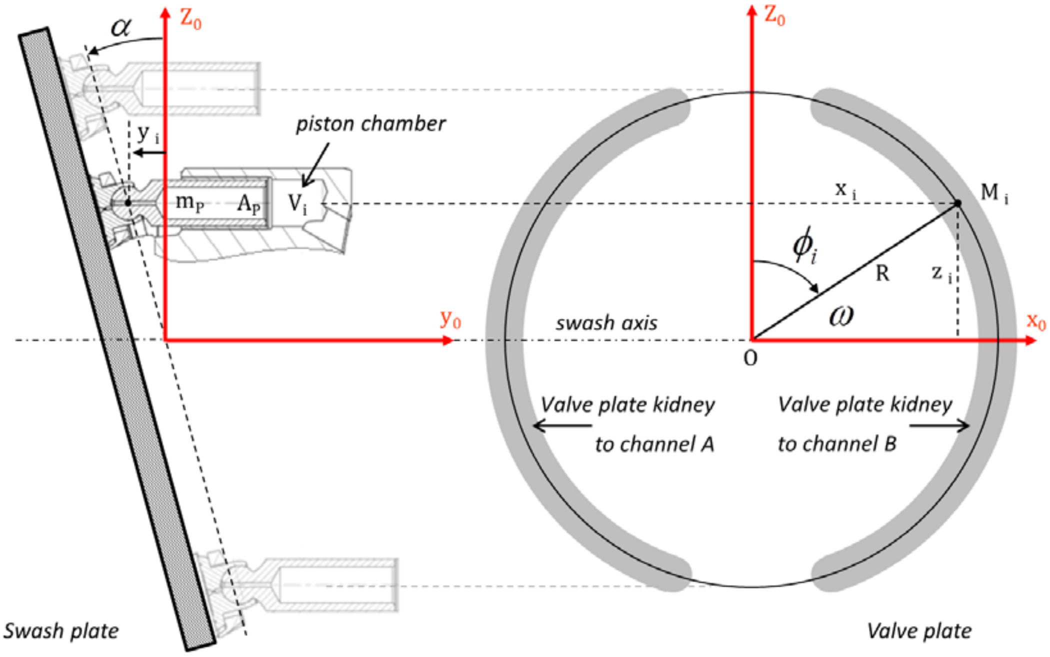

Figure 2: Sketch of motor kinematics (cross-angle not represented)

The displacement angle is referred to as the angle α around the x_0 axis. This first rotation transforms (x_0, y_0, z_0) into an intermediate coordinate system (x_1, y_1, z_1). Then the cross-angle is referred to as the angle γ around the axis z_1 and transforms (x_1, y_1, z_1) into (x_2, y_2, z_2).

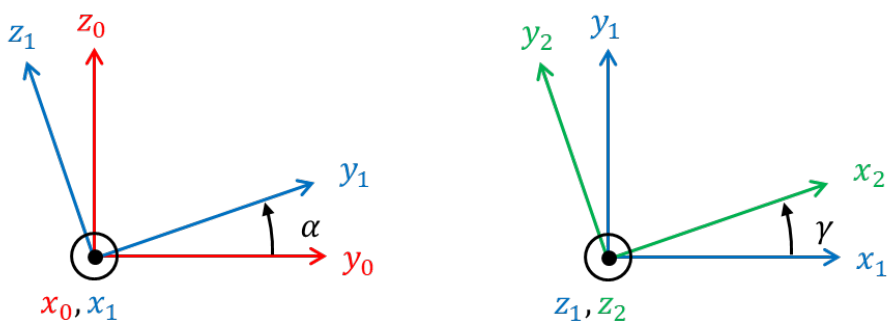

Figure 3: displacement angle (left) and cross-angle (right)

The piston stroke, i.e. the displacement of the piston along the y_0 axis, follows as:

$$y_i = R \cdot \frac{tan\gamma}{cos\alpha} \cdot sin\phi_i - R \cdot tan\alpha \cdot cos\phi_i \tag{1}$$

Then the model computes the pressure-rise rate in the piston chamber as:

$$\frac{dP_i(\phi_i)}{dt} = \beta \cdot \frac{Q_{iA} + Q_{iB} + Q_{iKIN}}{V_i(\phi_i)} \tag{2}$$

This quantity is integrated to get the pressure profile for each piston at each simulation time step. It is then possible to compute Q_{iA} and Q_{iB}, the flow rates exchanged between the control volume of the ith piston and the channels A and B respectively. It is assumed that these flow rates occur in turbulent regime and can be computed as:

$$Q_{iA} = sign\,(P_A - P_i) \cdot C_q\, A_{iA} \sqrt{\frac{2\,|P_A - P_i|}{\rho}} \;\; and \;\; Q_{iB} = sign\,(P_B - P_i) \cdot C_q\, A_{iB} \sqrt{\frac{2\,|P_B - P_i|}{\rho}} \tag{3}$$

Q_{iKIN} is the piston motion-induced flow rate that relates to the piston axial velocity $\dot{y}_i$ according to

$$Q_{iKIN} = A_p\, \dot{y}_i \tag{4}$$

In Equations (1) to (4):

P_A and P_B are the pressures in the channels A and B respectively.

β and ρ are the bulk modulus and the density of the hydraulic fluid respectively.

C_q is the flow coefficient of the considered flow rate.

A_{iA} and A_{iB} are the flow areas between piston chamber and channels A and B.

3.2 Optimization objectives

We are primarily concerned about the audible noise in the cabin of the aircraft. However this sound can be generated through two main paths:

- Through pressure pulsations generated by the hydraulic motor and transferred to the hydraulic circuit, also known as "fluid borne noise"
- Through the vibrations of the motor structure, transferred to the rest of the aircraft structure, also known as "structure borne noise"

Since the noise transfer path is typically difficult to identify in an aircraft and because the developed hydraulic motor should not be aircraft-specific, both pressure pulsation and vibrations have to be optimized at the same time.

Unfortunately, both pressure pulsation and vibrations are system dependent. Exactly like positive displacement pumps, positive displacement motors generate a flow rate pulsation, the pressure pulsation being only the consequence of the flow pulsation for one given hydraulic circuit. Similarly, the motor generates forces and moments pulsation. The vibration of the motor itself is depending on its own structure but also on the impedance of the structure it is mounted on.

Therefore the value to be monitored for assessing the general tendency of a hydraulic motor to generate pressure pulsation is the flow pulsation at its inlet port (high pressure port). Assessing its tendency to generate vibration is slightly more complicated because forces and moments in all three spatial directions are involved. Nevertheless Skaistis has shown that the swash plate moment M_{X_0} typically shows the greatest potential as a noise source /1/ and to the best of our knowledge, this proved to be true for many practical cases.

Choosing which quantity to monitor is still not enough and precise optimization objectives should be determined both for flow pulsation and swash plate moment pulsation. Again, since the quietness of the motor should not depend on the system it is used in, it is not possible to target amplitude reductions at specific frequencies and we chose two equally weighted optimization objectives consisting of the normalized peak-to-peak amplitude of time signals simulated for the consumed flow-rate Q_A and the swash plate torque M_{X_0}.

$$objective = \frac{\left(\max(Q_A(t)) - \min(Q_A(t))\right)}{mean(Q_A(t))} + \frac{\left(\max\left(M_{X_0}(t)\right) - \min\left(M_{X_0}(t)\right)\right)}{mean\left(M_{X_0}(t)\right)} \tag{5}$$

Following constraints have been added to the optimization objectives to avoid cavitation resp. low efficiency:

- Minimum acceptable pressure in the piston chamber
- Maximum acceptable internal leakage due to valve plate underlap

3.3 Simulation results

A set of optimal parameters for both the cross-angle value and the valve plate geometry has been obtained by running the optimization on the simulation model described earlier.

In the figures below, simulation results show the comparison between the reference motor initially used for the high lift actuation and the Liebherr motor presented in this document in terms of flow rate and swash plate moment pulsations.

Figures 4a and 4b show that the choice of the flow rate peak-to-peak amplitude as an optimization function has given very good results in the time domain with a reduction of a factor 2.0 of the peak-to-peak amplitude compared to the reference. In frequency domain, it is interesting to see that it has led to significant improvements at harmonic frequencies but not necessarily at the fundamental frequency.

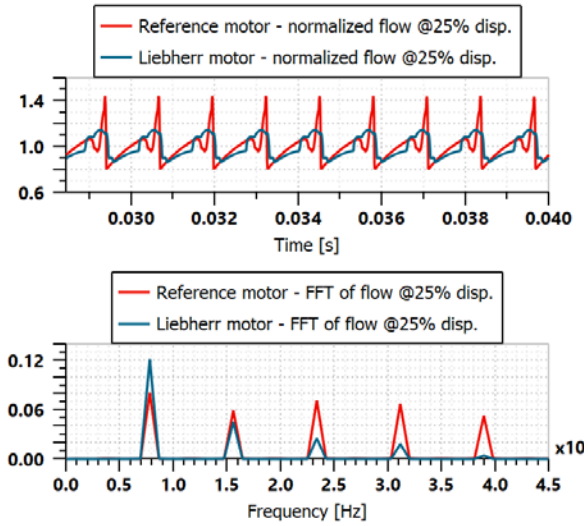

Figure 4a: Normalized flow rate generated by motors at inlet port in time and frequency domain for 25% of max. displacement

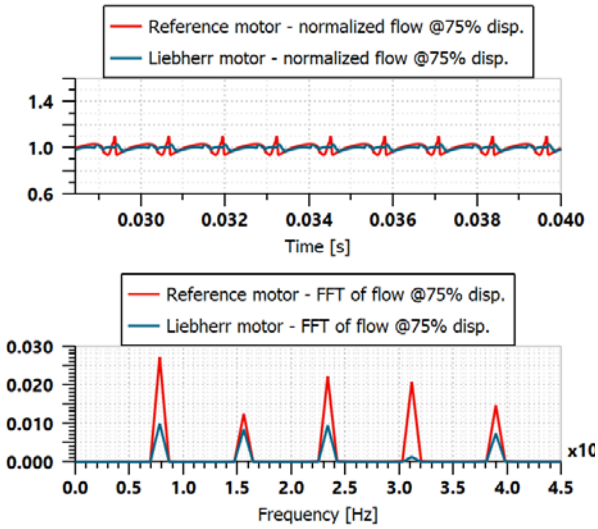

Figure 4b: Normalized flow rate generated by motors at inlet port in time and frequency domain for 75% of max. displacement

The figures 5a and 5b show a massive improvement for the peak-to-peak amplitude of the swash plate moment with a reduction of the peak-to-peak amplitude by a factor 4.0 to 7.0. It indicates that the swash plate moment was most probably not considered when the reference motor was designed, thus probably leading to important vibration of the surrounding structure and consequent audible noise emission.

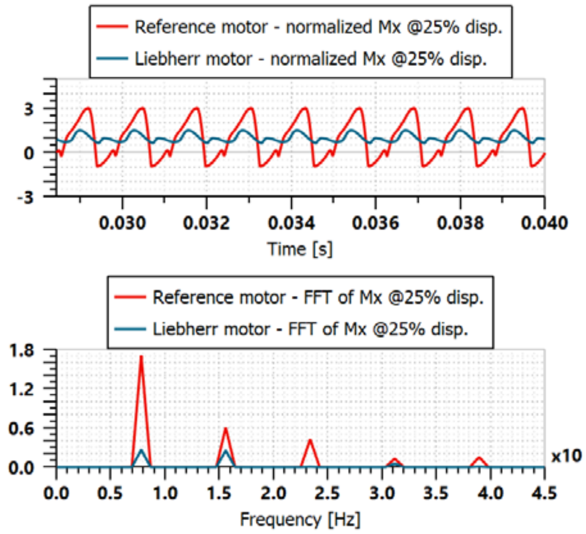

Figure 5a: Normalized swash plate moment generated by motors in time and frequency domain for 25% of max. displacement

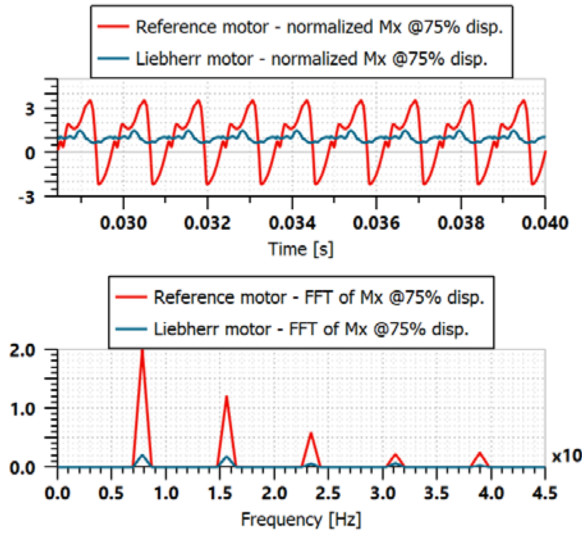

Figure 5b: Normalized swash plate moment generated by motors in time and frequency domain for 75% of max. displacement

4 Noise and Vibration Reduction in Laboratory Environment

In the following test data is presented which demonstrates the achieved reduction regarding noise and vibration. In a laboratory environment the noise emission of the PCU is compared to the reference device /7/. It is further clarified to what extend the improvements affect passenger comfort.

4.1 Structure borne noise

In order to measure the structure born noise emitted during operation, both PCU have been equipped with six tri-axial acceleration sensors. In figure 6 the positioning of these sensors is illustrated. The spectrum of vibration was then measured in various load conditions /8/.

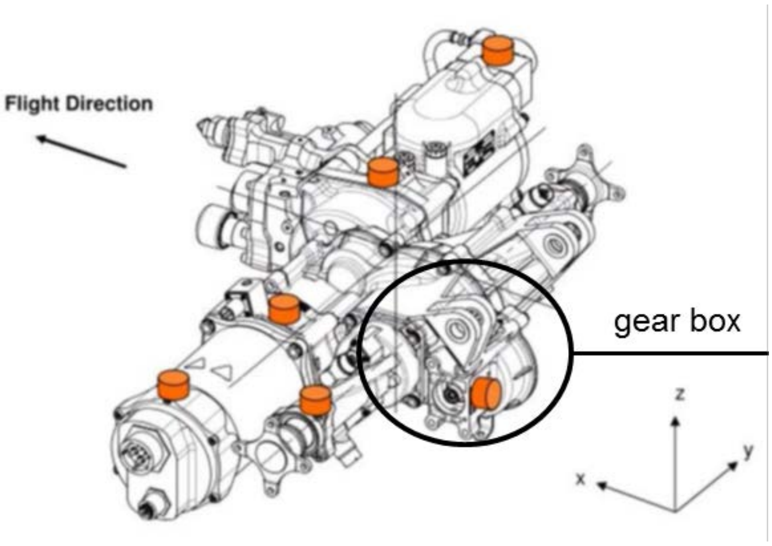

Figure 6: Positioning of acceleration sensors.

Basically all sensors indicated similar characteristics regardless of the sensor position and axis of motion. Therefore the results shall be discussed using the example of the y-axis of the sensor located on the gearbox (refer to figure 6).

The frequency spectrum that occurs during operation of both PCU is depicted in figure 7. While the characteristic frequencies of both units are similar, the vibration amplitude of the Reference PCU (red) exceeds the Liebherr PCU (blue). Both devices are characterized by a fundamental frequency of 790Hz and a first harmonic at 1590Hz. In general the Liebherr hydraulic motor has significantly decreased the structure borne noise for the fundamental and the following harmonics.

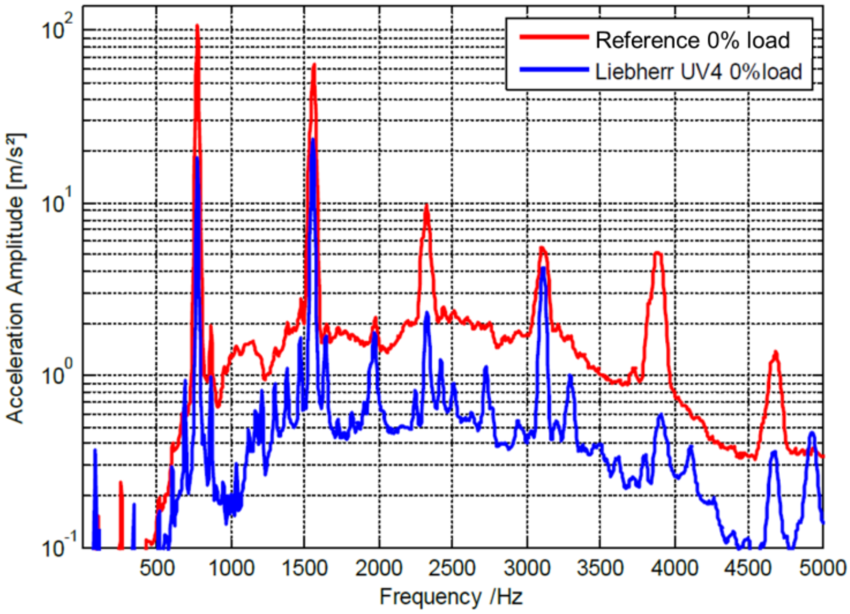

Figure 7: Acceleration amplitude over frequency range of 10Hz-5kHz for varying load conditions.

4.2 Air borne noise

In addition to the structural vibration the sound pressure amplitude spectrum emitted by the PCU was recorded. Microphones were installed at a distance of 1m from the unit. The results are illustrated in figure 8. Thereby the sound pressure was A-weighted according to the international standard IEC 61672:2003, in order to account for the relative loudness perceived by the human ear, which is indicated by the unit dB(A).

Similar to the vibration data a fundamental frequency of 779Hz and a first harmonic at 1558Hz are apparent. Again there is a reduction in the signal amplitude of the Liebherr PCU in comparison to the reference unit.

It is further noted that a sound pressure amplitude increase of 10dB is equivalent to a doubling of the subjective sound level. Therefore according to figure 3 the fundamental frequency with 103dB(A) and its first harmonic with 90dB(A) will have the most disturbing impact to the human ear. The sound pressure of the subsequent harmonics does not increase past 75dB(A). Hence the subjective sound level will be less than half compared to the first harmonic frequency.

Consequently in figure 9 the focus is on the fundamental frequency and its first harmonic. Here the A-weighted sound pressure level of both PCU is depicted in dependence on the load condition. Apparently the Liebherr hydraulic motor has significantly decreased the air borne sound pressure for both the fundamental and the first harmonic frequency. Corresponding to the difference of more than 10dB(A) a subsequent reduction of the sound level in AC environment is expected.

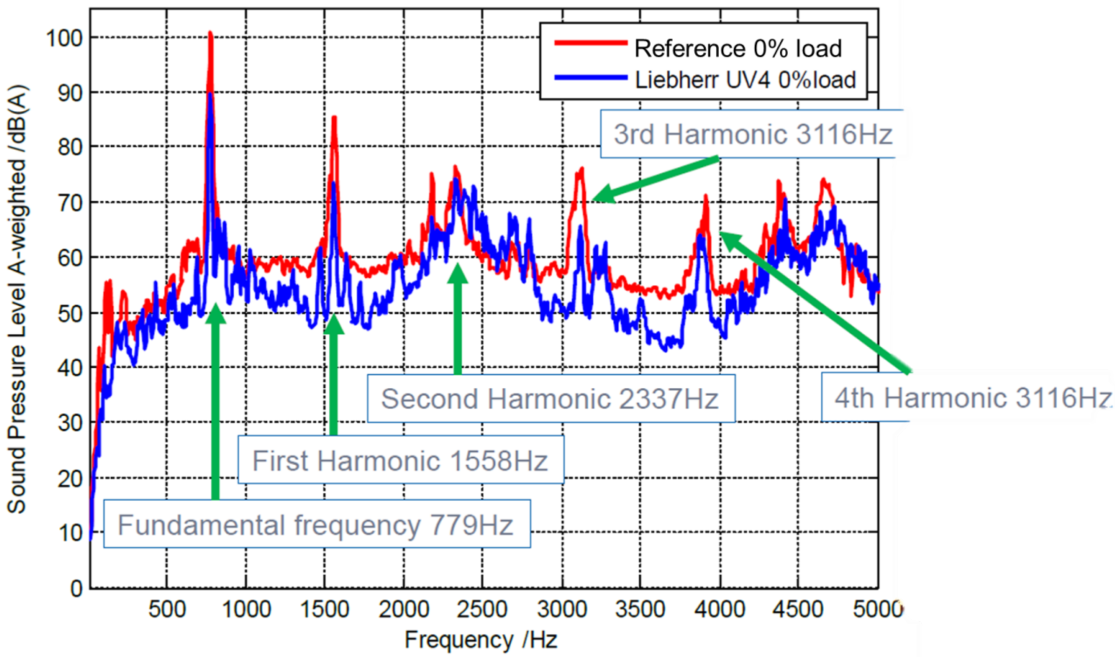

Figure 8: Sound pressure level (A-weighted) over a frequency range of 10Hz-5kHz for varying load conditions.

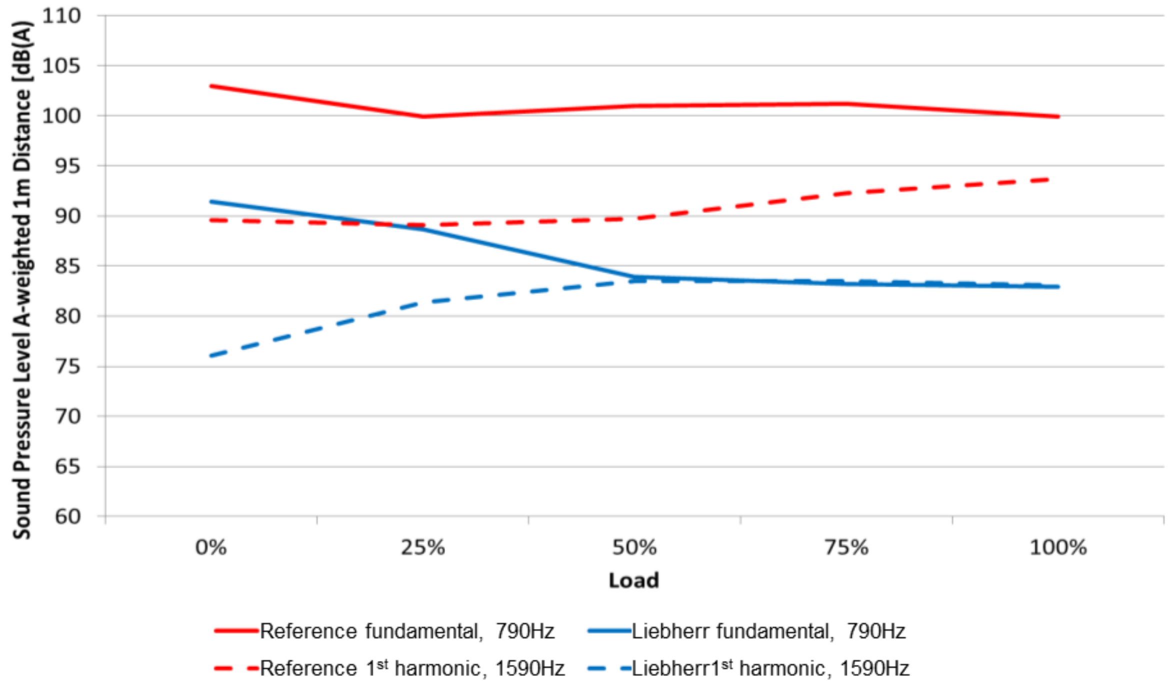

Figure 9: Comparison of sound pressure levels (A-weighted) for different load conditions.

5 Noise and Vibration Reduction in Aircraft Environment

Based on the results in laboratory environment in July 2015 Airbus performed vibro-acoustic tests on an A350 test AC comparing the Liebherr hydraulic motor with the reference unit /9/. Therefore nine microphones have been installed in the cabin for this test campaign (refer to figure 10). Additionally one rotating boom microphone is positioned in the aisle between row 10 and 11.

Figure 10: Microphone test setup for sound pressure level measurement within an A350.

Afterwards several configurations have been tested featuring slat extension and retraction in both a stepwise and complete progression /9/. Thereby the PCU has been powered alternatively by the ground cart and an electromotor pump (EMP). In the following test data with ground cart is presented.

In figure 11 the A-weighted frequency spectrum of the cabin sound pressure level recorded with the rotational boom microphone is illustrated for full slat extension. Again the fundamental frequency in the region of 800Hz and a first harmonic in the region of 1600Hz are apparent. Beyond 500Hz there is a difference in sound pressure level of about 5dB(A) between the Liebherr PCU and the reference unit. This visualizes the subjective perception of noise within the cabin, which is lower when the Liebherr PCU is used.

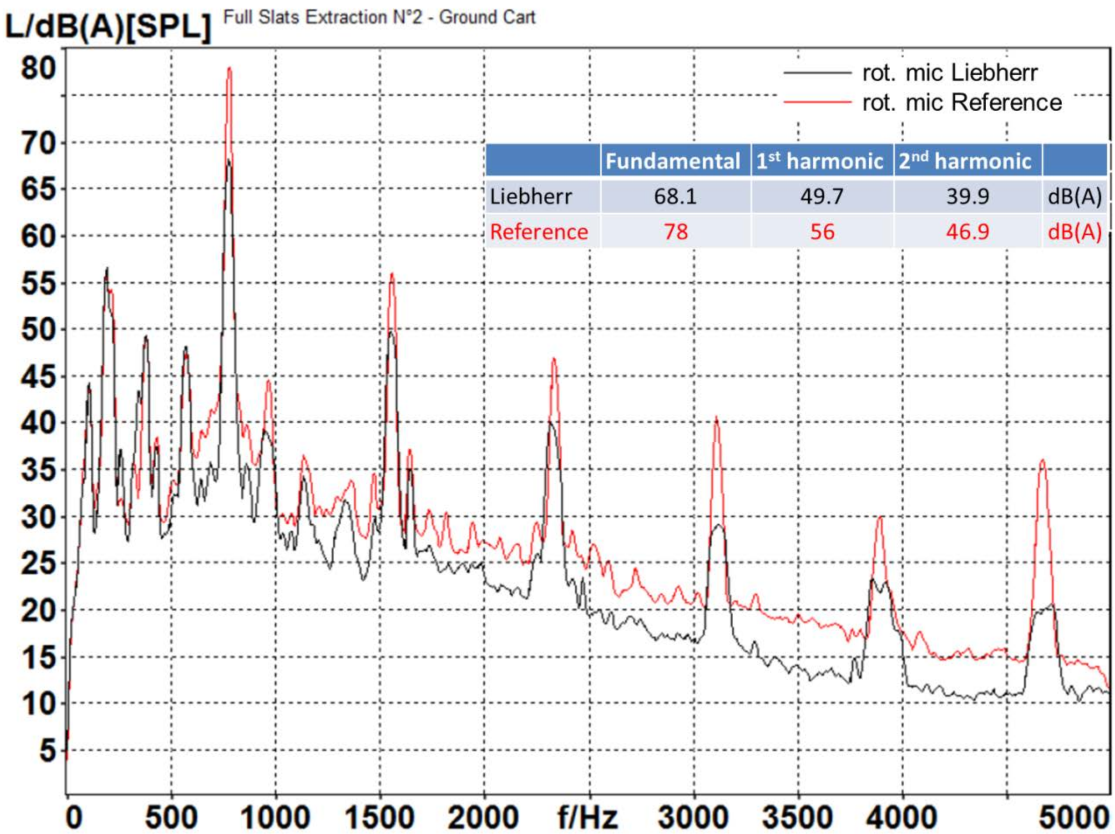

Figure 11: Static sound pressure level (A-weighted) for the rotating boom microphone during slat extension.

Furthermore compared to the laboratory measurements, which have been recorded at a distance of 1m from the source, the sound pressure in the cabin is reduced. Especially higher frequencies are naturally damped, while low frequencies are better transmitted though the fuselage. As mentioned above the subjective perception of sound level is logarithmic to the sound pressure level. Hence again the sound pressure at the fundamental frequency was considered for a detailed comparison between the Liebherr PCU and the reference. The resulting difference of 10db(A) correspond to a subjective sound level reduction by a factor of two.

In figure 12 a surface map illustrates the noise reduction in the A350 cabin and the specific seats. The sound pressure recorded by the rotating boom microphone is indicated by the circle in the aisle between row 10 and 11. The colour scheme represents a pressure between 55dB(A) and 80dB(A) by colours of blue to red. Except for seats 10A and 11D a significant reduction of noise of 5-10dB(A) is apparent. The numeric difference in sound pressure is given in table 1.

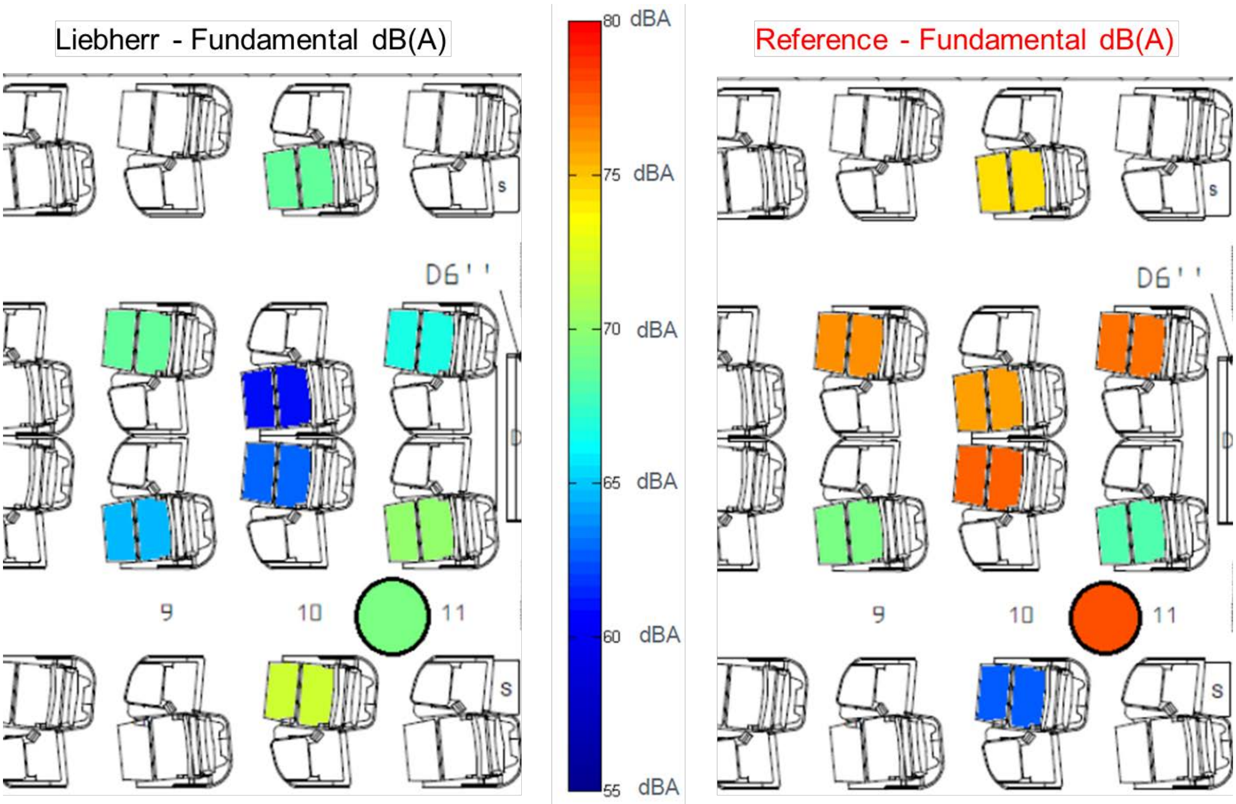

Figure 12: Surface map: Comparison of cabin sound pressure level (fundamental, A-weighted, slat extension).

Microphone	Difference - dB(A)
Seat 11D	0.4
Seat 10A	8.6
Seat 11H	-10.5
Seat 10G	-15.5
Seat 10E	-14.9
Seat 10L	-6.4
Seat 9D	-5.0
Seat 9H	-8.4
Rot. boom mic	-9.9

Table 1: Difference in cabin sound pressure (fundamental, A-weighted, slat extension).

6 Conclusion

It has been demonstrated that effective noise reduction of a hydraulic motor is possible without affecting weight, cost or performance. As the vibration level of the motor was also reduced together with the reduction of the noise emission, the reliability of the unit was improved.

This improvement was not achieved by the typical secondary means, for instance rubber isolation or thicker or more noise absorbing housings; it was achieved by reduction of hydraulic pressure variation between the primary parts of the motor, which are cylinder-block, swash plate and pistons. Thereby the optimized motor produces less noise and vibrations compared to the reference motor. Since no additional parts are necessary, this is the favoured method of noise reduction.

Such an improvement is only possible if the physics inside the motor is properly understood and a verified simulation model is available to support the development activities. Therefore appropriate simulation models are the key for the next generation of low noise, high efficiency hydraulic pumps and motors.

7 Acknowledgements

The noise reduced variable displacement hydraulic motor, whose test data is presented within this paper has been developed within the BMWI founded research project “Move On”. The project was realised in cooperation with Liebherr-Aerospace Lindenberg GmbH and Liebherr Machines Bulle SA together with Airbus Deutschland GmbH. Airbus was mainly responsible for the test campaign within the A350.

Nomenclature

Variable	***Description***	***Unit***
γ	Cross-angle	[rad]
α	Displacement angle	[rad]
x_i	Displacement of the ith piston on x axis	[m]
y_i	Displacement of the ith piston on y axis	[m]
z_i	Displacement of the ith piston on z axis	[m]
V_i	Fluid volume of the ith piston chamber	[m^3]
ϕ_i	Angular position of the ith piston	[rad]
m_P	Mass of piston	[kg]
A_P	Area of piston	[m^2]
R	Cylinder-block pitch radius	[m]
P_i	Pressure in ith piston chamber	[bar]
Q_{iA}	Flow rate between piston chamber and port A of motor	[m^3/s]
Q_{iB}	Flow rate between piston chamber and port B of motor	[m^3/s]
P_A	Pressure in port A	[bar]
P_B	Pressure in port B	[bar]
β	Bulk modulus of fluid	[bar]
ρ	Density of fluid	[kg/m^3]

C_q	Discharge coefficient of valve plate opening	[-]
A_{iA}	Valve plate opening area between piston chamber and port A of motor	[m^2]
A_{iB}	Valve plate opening area between piston chamber and port B of motor	[m^2]
M_{X0}	Swash plate moment around the X_0 axis	[N.m]

References

/1/ Skaistis, S., *Noise Control of Hydraulic Machinery*, Marcel Dekker Inc., New York, 1988.

/2/ Johansson, A., Övander, J., Palmberg J-O., *Experimental verification of cross-angle for noise reduction in hydraulic piston pumps*, in Proc. of the Inst. of Mech. Eng., Part I: Journal of Systems and Control Engineering, Vol 221, Issue 3, pp. 321 – 330, 2007.

/3/ Ericson L., Ölvander J., and Palmberg J-O., *Flow Pulsation Reduction for Variable Displacement Motors Using Cross-angle*, in Proc. of Power Transmission and Motion Control (PTMC 2007), pp. 103-116, Bath, UK, 2007.

/4/ Manring, N.D., *Hydraulic Control Systems*, John Wiley & Sons, New York, 2005.

/5/ Ivantysyn, J. and Ivantysynova, M., *Hydrostatic Pumps and Motors: Principles, Design, Performance, Modelling, Analysis, Control and Testing*, Tech. Books International, New Delhi, India, 2001.

/6/ LMS Imagine.Lab AMESim, User Manual rev 14

/7/ Sinz, G., *VD PCU Noise Reduction - Concepts to Reduce PCU Noise*, internal document, Liebherr-Aerospace, Lindenberg , Germany 2013.

/8/ Ayme, F., Parisot-Dupuis, H., Villard, M., *A350 MSN5 – Vibro-acoustic ground test on FPCU* , technical report, Airbus, France, 2015.

/9/ Heise, U., *Move.On - Comparision of hydraulic motors regarding fluid borne, structure borne & air borne noise*, technical report, Airbus, Germany 2015.