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Reducing Compressor Vibrations by Load Torque Compensation from Acoustic Point of View

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ABSTRACT

As heat pumps help meeting the CO_2 reduction goals for residential heating, a fast and steady increase of residential heat pump sales is predicted. To enable the installation of heat pumps in densely populated areas, very low noise targets have to be met, especially in the European markets. One major noise source of heat pumps is the compressor. Many authors have identified load torque compensation (LTC) as promising method to reduce noise and vibrations of compressors. So far, this topic has been approached from a control-perspective with focus on developing suitable controller strategies. This paper, in contrast, analyses LTC from an acoustic point of view. It identifies which compressor excitations and hence vibrations can be reduced by LTC and which cannot. With this knowledge the noise reduction potential of LTC can be predicted. Therefore we performed vibration measurements on the compressor surface with and without LTC. To investigate the effects of LTC inside the compressor and consequentially evaluate the potenial of LTC we performed multibody simulations. These simulations cover the path from the load torque source via the applied driving torque to the compressor vibrations. As input for the multibody simulation, the load torque to be compensated is calculated from thermodynamic and geometrical relationships. With our investigations we deliver the following findings: First, we derive how much certain vibration orders can be reduced by LTC. For the investigated twin rotary compressor, we were able to reduce even orders but not odd orders. Second, we estimate the reduction potential of LTC at different operational points. The reduction potential is increasing with higher pressure differences and decreasing with increasing rotational speed. Third we identify which vibration directions can be reduced by LTC. Based on these results, LTC can be combined more efficiently with other noise reduction measures, leading to quieter heat pumps.

1. INTRODUCTION

Differences between load torque and driving torque in a rotary compressor lead to speed ripples, that cause vibrations and noise. As the name indicates, load torque compensation (LTC) is a method to compensate varying load torque with a well following driving torque to reduce the speed ripples and as a result vibrations and noise. It has been identified as a promising method by many authors. For example, Cho (2006) presented a sensorless control for a single piston rotary compressor using errors between the real currents of the electric machine and the calculated currents of a system model. This control compensates the load torques and reduces the speed ripple. Huang, Yu, and Chen (2007) proposed a sensorless control for systems with load torques, repetitive regarding the absolute rotor position. The repetitiveness is used to reduce the speed variations by estimating the load from the commutation interval. Latham and McIntyre

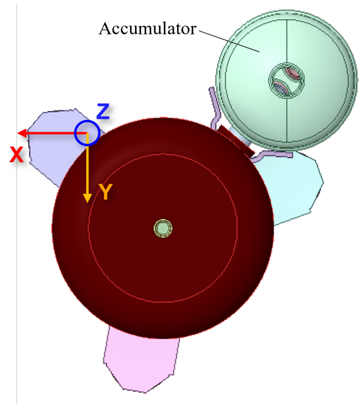
(2018) proposed a controller, which uses a load torque observer as feed forward input to compensate speed ripples of the compressor more effectively. The load torque observer is depending on the compressor geometry and thermodynamic behavior of the refrigerant. All these papers have in common, that they focus on developing a controller strategy for reducing speed ripples by LTC. This paper instead takes an acoustic point of view. Previous investigations in a heat pump have indicated a dominance of the compressor vibrations for overall heat pump noise and that vibration reduction on the compressor surface leads to a quieter heat pump. Therefore, in this paper we investigate the effects of LTC on vibrations of a twin rotary compressor with focusing on the path from load torque and driving torque to vibrations on the compressor surface. This paper shows a measurement approach as well as a simulation approach. In the measurement approach we optimize the driving torque of the electric machine iteratively to find a vibration minimum measured with an accelerometer on the compressor surface. The compressor itself can be considered as a black box for the measurement approach: we do not need speed ripple information nor load torque information. For the simulation approach we calculate the load torque first. If no LTC is applied we assume a constant driving torque, which is the mean value of the load torque over one crankshaft rotation. If LTC is applied, we assume a perfectly compensated load torque at each time. These simulations and resulting findings help to evaluate the benefit of LTC without the necessity to perform measurements with a LTC control first. Consequently, this reduces development time for a new controller and measurement time in the early heat pump development stage when the decision for or against LTC is made. To evaluate the simulations, this paper presents vibration measurements and derives main effects of LTC.

2. MEASUREMENTS

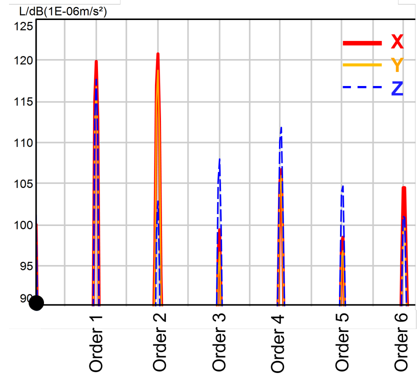
Figure 1 shows our measurement setup. We used accelerometers to measure vibrations on the compressor surface as shown in Figure 1a and minimized the vibrations by iteratively changing the driving torque of the electric machine. This has the advantage of neither needing to know the rotational speed ripple of the crankshaft nor the load torque variation. In this particular case we optimized the driving torque regarding a sensor at the accumulator and evaluate the results on another sensor at “Foot 1” as shown in Figure 1b. Figure 1c shows a typical frequency spectrum of a twin rotary compressor. All even and odd orders, regarding the rotational speed of the crankshaft, get excited. The following describes the observations made from our measurements. The small orders one to six have the highest excitations, thus they are examined in this paper to evaluate LTC.

Table 1 shows the results for even orders two, four and six for a crankshaft rotational speed of 40 rps. The first observation is about which orders can be compensated by LTC. For odd orders one, three and five LTC did not lead to an overall vibration reduction, therefore they are not displayed in the table, and we focus on even orders. The second observation is about the influence of pressure differences for compressor vibrations and for the reduction potential with LTC. Subtable 1a shows a small pressure difference of 7 bar, Subtable 1b a high pressure difference of 15 bar. The suction pressure did not change during all investigations in this paper. Comparing the baseline measurements “LTC off” of both subtables we observe increasing vibrations levels with higher pressure difference especially for second order (117.3 dB to 123.5 dB), but also in weakened form for fourth and sixth order. This behavior can be explained with the increasing loads resulting from higher pressure differences, see Section 3.1. Comparing “LTC off” and “LTC on” we observe a vibration level reduction for all even orders. The difference between “LTC off” and “LTC on” is expressed as “Delta”. For the second order we achieved a vibration level reduction of −9.9 dB at 7 bar pressure difference (Subtable 1a) and −15.7 dB at 15 bar (Subtable 1b). Comparing “LTC on” for Subtable 1a and 1b we find the vibration level with LTC for each order is similar for small pressure differences of 7 bar as well as for high pressure differences of 15 bar. This could indicate, that increasing of the radial compression forces with higher pressure differences effects the even orders insignificantly and the increasing load torque, which leads to higher vibrations without LTC, is compensated completely by LTC. Another observation is about vibration reduction of the higher orders compared to second order. A reduction potential for fourth and sixth order has been observed but tends to be smaller than for the second order. There are three different potential explanations for this observation. First, the load torque ripple, which can be compensated with LTC is dominated by the second order, see Section 3.1. The second potential explanation are optimization possibilities on the LTC controller for higher orders. Third, our approach to determine the driving torque variation for LTC by minimizing the vibration at a certain point on the compressor is limited when we reach a frequency range with local resonances, where the compressor housing does not behave as one rigid body. This can be observed in Subtable 1b, by comparing “4th order” and “4th order**”. The latter shows the vibration minimization regarding “Foot 1” instead of the accumulator. This means in “4th order**”

(a) Test bench



(b) Measurement position “Foot 1” marked by coordinate system



(c) Typical frequency spectrum of a twin rotary compressor, measured at “Foot 1” (Δp = 15bar, 40rps)

Figure 1: Measurement setup

evaluation and optimization point are the same. It leads to an increased vibration level reduction compared to “4th order”, especially in direction x and y see values in brackets in “Delta”. This leads to the last observation in Table 1. Comparing overall vibration level reduction in ”Delta” with the directional ones in the brackets, we observed a high vibration level reduction in x and y direction and a limited effect in z direction. This is explainable as the load torque ripple as well as the compensating drive torque ripple when applying LTC is operating in the x-y-plane, see Section 3.1.

Table 2 displays the same investigations as Table 1 but for higher speeds, namely 60 rps instead of 40 rps. In line with the observations at 40 rps, no overall vibration reduction for odd orders one, three and five, was possible. Therefore, this table contains even orders, two, four and six, only. First this paper describes the differences and similarities of 60 rps rotational speed without LTC (“LTC off”) to 40 rps. As observed for 40 rps also for 60 rps vibration levels increase with higher pressure differences, see Subtable 2a for 7 bar and Subtable 2b for 15 bar, , except order 6. Also the observation, that with LTC, vibration levels with smaller and bigger pressure difference have the similar vibration levels, holds as well for 60 rps as it has for 40 rps, see “LTC on” in Table 2 and 1. Higher rotational speed itself leads to higher vibrations levels, compare 2a and 1a or 2b and 1b. There are different potential explanations for this. First of all, load torque variation is depending on crankshaft rotation angle, see Figure 2a, therefore the variation happens faster at higher rotational speeds. The driving torque of the basic control would probably follow less accurate than at lower speeds. Second, at different rotational speeds same orders are located at different frequencies, which leads to different resonances. This for example can explain the high differences in the fourth order, e.g. “LTC off” vibration levels for fourth order increased by 8.2 dB from 111.6 dB at 40 rps (Subtable 1a) to 119.8 dB at 60 rps Subtable 2a). Third, also the whirling of the crankshaft in its bearings could increase with higher speeds. Comparing “LTC off” and “LTC on” or observing directly “Delta” in Subtables 2a and 2b, even orders can be reduced with LTC, which has already been observed at 40 rps in Table 1. There is one exception which is “6th order” in 2b. On the one hand our optimization approach could be a reason for that. We cannot determine a vibration reduction after the uncoupling of the accumulator after its resonance. On the other hand, the LTC control used at this point of our investigations, still had optimization potential regarding higher speeds, higher orders, and higher mean loads. The improvements at 60 rps are smaller than at 40 rps, e.g. we could reduce the second order in Subtable 2a by −2.5 dB instead of −9.9 dB in Subtable 1a. Possible explanations for this reduced effect of LTC at higher speeds are the same as mentioned before: The optimization potential of the LTC controller at higher speeds, increased whirling of the crankshaft as well as increasing influence of resonances. Again, the latter is indicated by comparing “Delta” of “4th order **” of Subtables 1b and 2b. Here the vibration level reduction is even increasing from −1.6dB at 40 rps (Subtable 1b) to −7.5 dB at 60 rps (Subtable 2b). Also it has been observed by comparing the values in the brackets in Table 2 “Delta” that x and y direction can be reduced by LTC and that there is limited potential of reducing exitation in z direction. This is true for speeds of 60 rps as well as for 40 rps.

Table 1: Measured vibration levels at “Foot 1” in dB (base $10^{-6}\frac{m}{s^2}$) with and without LTC at 40 rps rotational speed, optimized regarding accumulator acceleration, first line displays the sum of all three directions $\sqrt{x^2+y^2+z^2}$, second line in brackets shows each direction (x, y, z)

(a) Δp = 7bar

LTC	2nd order	4th order	6th order
LTC off	117.3	111.6	109.3
	(115.3,112.6,101.5)	(103.0, 104.1, 109.9)	(106.9, 103.6, 101.1)
LTC on	107.4	111.1	104.3
	(102.5, 97.8, 105.0)	(100.6, 104.9, 109.4)	(95.5, 97.3, 102.6)
Delta	-9.9	-0.5	-4.0
	(-12.8, -14.8, 3.5)	(-2.4, 0.8, -0.5)	(-11.4, -6.3, 1.2)

(b) Δp = 15bar, **optimization “Foot 1”

LTC	2nd order	4th order	4th order**	6th order
LTC off	123.5	113.9	114.2	109.9
	(121.4,119.3,103.1)	(107.0, 107.1, 111.5)	(107.2,107.0,112.3)	(107.2,103.6,103.0)
LTC on	107.8	113.1	112.6	105.8
	(101.9,96.6,106.1)	(106.8, 102.1, 111.5)	(99.8,94.7,112.3)	(97.3,97.5,104.3)
Delta	-15.7	-0.8	-1.6	-4.1
	(-19.5, -22.7, 3)	(-0.2, -5.0, 0)	(-7.4,-12.3,0)	(-9.9, -6.1, 1.3)

Table 2: Measured vibration levels at “Foot 1” in dB (base $10^{-6}\frac{m}{s^2}$) with and without LTC at 60 rps rotational speed, optimized regarding accumulator acceleration, first line displays the sum of all three directions $\sqrt{x^2+y^2+z^2}$, second line in brackets shows each direction (x, y, z)

(a) Δp = 7 bar

LTC	2nd order	4th order	6th order
LTC off	122.3	119.8	111.9
	(115.3, 117.5, 119.0)	(110.9, 115.7, 116.7)	(99.5, 111..3, 100.0)
LTC on	119.8	117.4	109.6
	(110.3, 108.8, 118.9)	(111.4, 103.2, 115.9)	(99.1, 108.8, 99.1)
Delta	-2.5	-2.4	-2.3
	(-5.3, -8.7, -0.1)	(0.5, -12.5, -0.8)	(-0.4, -2.5, -0.9)

(b) Δp = 15 bar, **optimization “Foot 1”

LTC	2nd order	4th order	4th order**	6th order
LTC off	127.2	121.9	121.9	110.9
	(120.7, 124.5, 121.1)	(118.8, 117.7, 113.4)	(118.8, 117.7, 113.4)	(105.6, 107.3, 105.2)
LTC on	118.7	116.8	114.4	112.4
	(109.9, 107.5, 117.7)	(110.7, 110.8, 113.7)	(100.3, 97.8, 114.1)	(106.4, 109.8, 105.4)
Delta	-8.5	-5.1	-7.5	1.5
	(-10.8, -17.0, -3.4)	(-8.1, -6.9, 0.3)	(-18.5, -19.9, 0.7)	(0.8, 2.5, 0.2)

To end this section, the listing shows a short summary of our measurement obeservations:

- Even orders can be reduced by LTC, odd orders cannot.
- Vibrations in the x-y-plane can be reduced very well by LTC, vibrations in z cannot
- With increasing pressure difference vibration levels increase without LTC. They can be compensated with LTC so that vibration levels with LTC are similar for higher and lower pressure differences.
- Vibration levels increase with higher rotational speeds. Vibration level reduction with LTC in contrary decreases with higher rotational speeds.

3. SIMULATIONS AND CALCULATIONS

This section is about the simulations performed to understand the measurement observations. Therefore, we use a multibody simulation model of the twin rotary compressor, investigated in the measurements. We apply loads resulting from the compression process as input forces and torques for the multibody model and specify the rotational speed and speed ripple. As torque ripple and speed ripple are depending on each other, the driving torque, which can be influenced by LTC, is an output of the model depending on the load torques and rotational speed. As in the measurements we evaluate the vibration level of the simulation model at ”Foot 1”, see Figure 1b.

3.1 Input parameters: loads and speed ripple

The following briefly describes the calculation of the input parameters for the multibody simulation model. In detail these relationships were investigated for example in Okada and Kuyama (1982). They calculated the radial and tangential forces acting on the crankshaft at the roller position, resulting from gas compression force and centrifugal force. Later Park (2010) used a similar approach to evaluate the radial force and torque on the crankshaft. The force resulting from the pressure difference is the highest force acting on the crankshaft (Park, 2010). As our approach is to use as little data as possible to evaluate LTC in an early development stage, we only used the force resulting from the pressure difference to estimate the load forces and torques for our model. Therefore, in our case the following input data are necessary to estimate the input parameters for our simulation model:

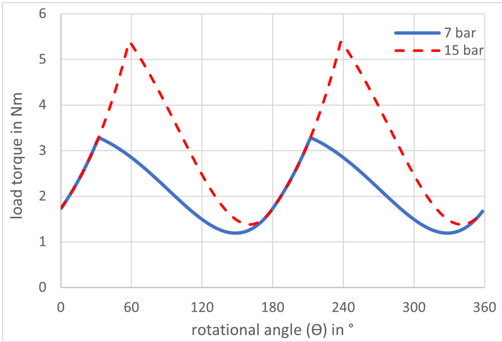
- cylinder height h
- cylinder radius r_c
- roller radius r_r
- eccentricity e
- specific heat ratio κ
- suction pressure p_S and discharge pressure p_D
- moments of inertia of rotating parts J_z (i.e. rotor of electric machine, crankshaft, rollers)

Pressure force $F_p(\theta)$ depends on the crankshaft rotation angle θ and can be calculated using the geometric relations shown in Figure 2c. The detailed calculations can be found in Okada and Kuyama (1982) and Park (2010). For the calculation we need the pressure difference of compression chamber pressure $p(\theta)$ and suction chamber pressure p_s. To calculate the resulting pressure force we need to multiply this pressure difference with the intersection between the two chambers S_{AB} and cylinder height h.

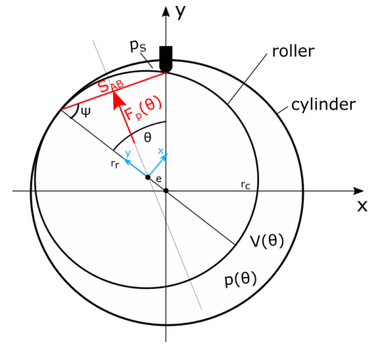
$$F_p(\theta) = S_{AB}h(p(\theta) - p_s) \tag{1}$$

The compression chamber pressure $p(\theta)$ is increasing with decreasing compression chamber volume $V(\theta)$ until the valve opens at $p(\theta) = p_D$

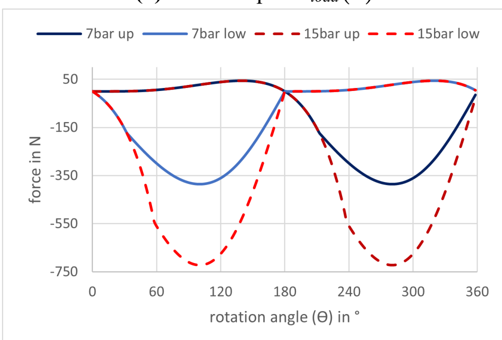
$$p(\theta) = p_s \left[\frac{\pi(r_c^2 - r_r^2)h}{V(\theta)} \right]^{\kappa} \tag{2}$$



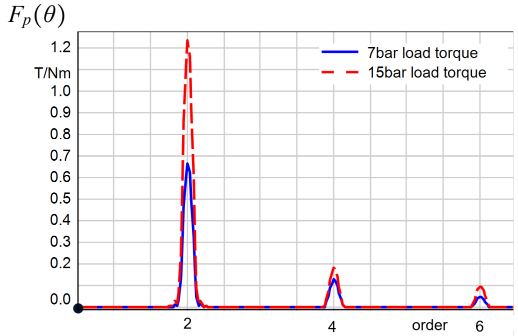
(a) Load torque $M_{load}(\theta)$



(c) Geometry relations for calculating pressure force $F_p(\theta)$



(b) Radial pressure force $F_{p,rad}(\theta)$



(d) Frequency spectrum load torque M_{load}

Figure 2: Loads

With the geometric relationship,

$$\cos(\psi) = \frac{\frac{1}{2}S_{AB}}{r_r} \qquad \psi \geq 0 \text{ for } \theta \leq \pi \text{ and } \psi < 0 \text{ for } \theta > \pi \tag{3}$$

and the eccentricity e the pressure force $F_p(\theta)$ can be separated in load torque $M_{load}(\theta)$ and radial pressure force $F_{p,rad}(\theta)$:

$$M_{load}(\theta) = F_p(\theta)\cos(\psi)e \tag{4}$$

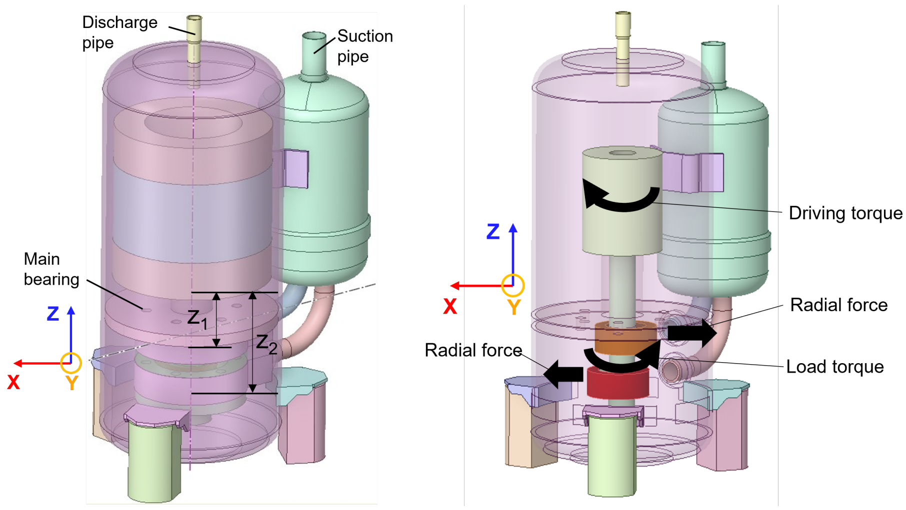
$$F_{p,rad}(\theta) = F_p(\theta)\sin(\psi) \tag{5}$$

The load torque $M_{load}(\theta)$ and radial force $F_{p,rad}(\theta)$ only depend on pressure difference, not on rotational speed. For the two pressure differences investigated in the measurements load torque and radial forces are shown in Figure 2a and 2b. Load torque $M_{load}(\theta)$ and driving torque $M_{dr}(\theta)$ are acting on the compressor assembly as shown in Figure 3b. At a twin rotary compressor there are two identical chambers, which are rotated to each other by 180°. The load torque $M_{load}(\theta)$ is the sum of the two chambers. The radial forces $F_{p,rad}(\theta)$ need to be treated separately for each chamber, as the distance from each chamber to the main bearing z_1 and z_2 is different (Figure 3a). This leads to different torques on the x and y direction of the main bearing for each compression chamber. A deviation of driving torque $M_{dr}(\theta)$ and load torque $M_{load}(\theta)$ leads to a speed ripple $\dot{\omega}(\theta)$ of the crankshaft rotation, depending on moment of inertia of rotary parts J_z:

$$\dot{\omega}(\theta) = \frac{M_{dr}(\theta) - M_{load}(\theta)}{J_z} \tag{6}$$

We assume without LTC driving torque $M_{dr}(\theta)$ is the mean value of the load torque $M_{load}(\theta)$, and therefore constant during the crankshaft rotation.

$$M_{dr}(\theta) = M_{dr} = \overline{M_{load}(\theta)} \tag{7}$$



(a) Twin rotary compressor

(b) Rotational parts

Figure 3: Twin rotary compressor

For evaluating the situation with LTC we assume a perfectly compensated load torque:

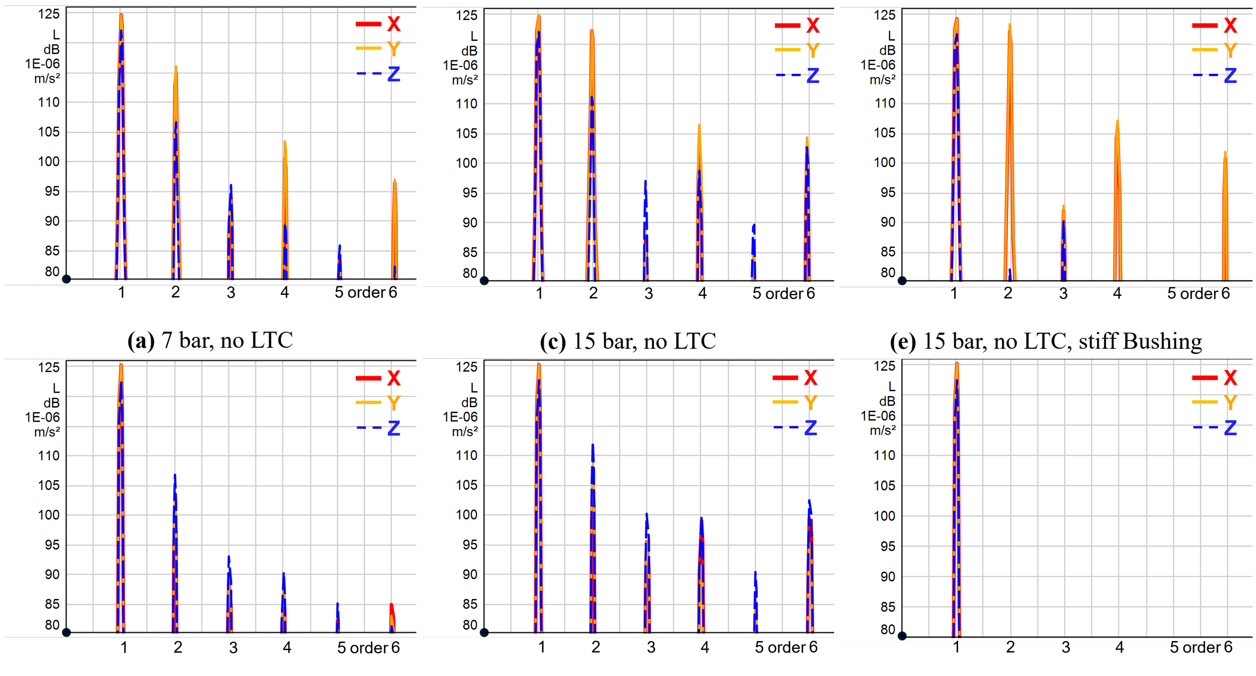
$$M_{dr}(\theta) = M_{load}(\theta) \tag{8}$$

The load torque from Figure 2a transferred to a frequency spectrum is shown in Figure 2d. As this spectrum contains only even orders, this matches with the observations that only even orders can be reduced by LTC.

3.2 Multibody simulation model and results

In a next step, the input loads can be applied to the multibody simulation model to investigate the main effects of LTC on the vibrations on the compressor housing, e.g. "Foot 1" (Figure 1b). Han, Hwang, and Koo (2004) confirmed that multibody simulations are suitable to predict rotary compressor vibrations by comparing their simulations with measurements. As main influences on the vibrations mass imbalances of the rotating parts and compression forces were identified. The isolator stiffness instead had limited effect on the accelerations. All these effects are considered in this paper. The compressor in our multibody simulation model is only connected to the ground via three rubber feet. These three feet are the only flexible parts, that allow motion of the compressor. All other parts are assumed rigid and discharge pipe and suction pipe are free ends (Figure 3a). For simplification of the model, we also assumed that the crankshaft is connected to the housing only via the main bearing, which was modeled as a bushing with stiffness and damping.

The following investigates which measurement observations (Section 2) can be represented and explained by our simulations, for the lower rotational speed of 40 rps. These simulation results are shown in Figure 4. In our measurements we observe a reduction potential for even orders and no reduction potential for odd orders. We also observe this in our simulations, compare Figure 4a, which shows our simulation without LTC and Figure 4b which shows the situation with LTC. The same holds for our simulations with higher pressure difference (Figure 4c and 4d). This can be explained as load torque in twin rotary compressors only excites even orders, see Figure 2d. Load torque M_{load} and driving torque M_{dr} act in x-y-plane, see Figure 3b. This leads to good vibration reduction in x in y direction and limited effects in z direction, which we observe in our measurements as well as in our simulations, see even orders two, four and six in Figure 4a and 4b or 4c and 4d. With increasing pressure difference vibration levels for even orders increase without LTC, from 7 bar (Figure 4a) to 15 bar (Figure 4c), which is in line with our measurements. It is also in line with the increasing pressure torques and forces, see Figure 2a and 2b. Only the first order is not depending on the pressure forces and therefore independent of the pressure difference. This indicates that the first order is dominated by mass imbalance of the rotating parts. In our measurements the vibration level reduction with LTC increased with higher pressure difference and after applying LTC even orders with 7 bar and 15 bar were of the similar vibration level, which



(a) 7 bar, no LTC

(c) 15 bar, no LTC

(e) 15 bar, no LTC, stiff Bushing

(b) 7 bar, LTC

(d) 15 bar, LTC

(f) 15 bar, LTC, stiff Bushing

Figure 4: Simulated vibration levels at "Foot 1" with and without LTC at 40rps

could not be shown with this simulation model. This limitation of our model indicates that the motion of the crankshaft is not modeled precisely enough. There are two possible reasons for that. One is the assumption of all rotating parts as rigid bodies. To exclude this we calculated the first bending mode of the crankshaft with the rotational parts, which is at over 1000 Hz. For our investigations up to maximum 400 Hz the rigid body assumption of the crankshaft seems justified. The second possible reason is the simplified bearing situation in our model, which is mapped to a bushing at the main bearing of the crankshaft. To compare different compressors in an early development stage the target was to build a model with easily accessible data only. From Figure 4d to 4f and Figure 4c to 4e we increased the stiffness of the bushing representing the bearing of the crankshaft. Except from first order, all orders change severely with the bearing situation. This shows that even for a rough comparison of different twin rotary compressors and to estimate the influence of LTC on the vibration levels of different compressors, it is important to describe the bearing of the crankshaft properly. Wang, Feng, Zhang, and Yu (2012) investigated a rotor-journal bearing system of a rotary compressor. They used stiffness and damping of the bearing oil film force and represented it as springs. One conclusion of their paper was that long journal bearings lead to deformation an instability. This paper indicates the importance of a bearing model for vibration analysis of a rotary compressor.

4. CONCLUSIONS

We performed measurements at a compressor test bench and multibody simulations to evaluate LTC from an acoustic point of view. The following summarizes our observations for the investigated twin rotary compressor. When applying LTC, our measurements and simulations show good vibration reduction of even orders and limited reduction of odd orders. Moreover, our measurements and simulations show good vibration reduction by LTC around the rotational axis of the crankshaft, meaning in x and y direction. Whereas, along the rotational axis of the crankshaft, z-direction, vibrations cannot be reduced. Another measurement observation is increasing vibration level and decreasing LTC vibration reduction with increasing rotational speed. Higher pressure differences in our measurements and simulations lead to increasing vibrations in even orders, especially for the second one. In our measurements we were able to compensate these increasing vibrations with LTC and reached the same vibration level of even orders for both investigated pressure differences. In our simulations we could show high influence of the crankshaft bearing model for resulting compressor vibrations. This indicates that a suitable friction bearing model is necessary to predict the compressor vibrations and LTC influence in simulations. This paper also indicates, that investigating LTC from an acoustic point of view is helpful to verify the performance of the control strategy and predict its maximum potential.

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