

Consideration of the pulsation as a design criterion for a newly developed oil-injected process gas screw compressor

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Abstract

For the newly developed oil-injected process gas screw compressor an extensive sound analysis has been performed during the prototype testing. The findings can help to design components that are exposed to pulsation. Thus an appropriate design may prevent serious consequence of resonances like excessive noise or any mechanical damage. Based on the measured external radiated sound power level this paper introduces a practical and feasible approach which allows estimating the internal sound pressure level which is introduced into the discharge pipe by the compressor referring to the calculation method of the VDI guideline 3733.

1. Introduction

MAN Diesel & Turbo is known for its reliable oil free process gas screw compressors. Various types of compressors are successfully operated in a wide range of industrial applications. Due to a continuous extension of the product portfolio a series of oil-injected screw compressors (CPO) was developed within recent years and has now reached a market-ready status.

Pulsation typically occurs at screw compressors and leads to high radiated sound power levels which have to be considered during the design process. The consequence could be the installation of noise reducing devices (noise hoods, silencers) or the prevention of any resonances which could have a negative effect. Oil-injected screw compressors usually have a lower external sound power level than dry ones because of the lower rotational speed and the damping characteristic of the injected oil.

The prototypes of the newly developed CPO have been tested mechanically and thermodynamically. In addition an extensive sound analysis has been performed as well. Only real testing brings enough information to be able to evaluate the acoustic situation of the system in dependence of several operating conditions. This information is needed to predict the radiated sound power level as well as to design components that might be affected by the

pulsation. In case of resonances excessive noise or any mechanical damage can be the consequence.

Different measurement methods are deployed to investigate the acoustic relations. The internal sound pressure level within the discharge piping is directly measured. By applying a certain calculation method one can compute the external radiated sound power level based on the internal sound pressure level and under consideration of the operational conditions. If this method is applied the other way round one could compute the internal sound pressure level from the external radiated sound power level. This paper will verify this concept by comparing the gathered data of different measurement methods.

The findings of the sound analysis will be presented. Based on this data and the presented measurement method a practical and feasible approach shall be established which allows estimating the internal sound pressure level despite possible uncertainties. Applying this approach, complex measurement techniques can be avoided because the gathered data of simpler methods is sufficient for the prediction of the acoustic situation within the pipe. Regarding the pulsation as a design criterion this approach helps to receive and use both the internal and external sound spectrum as input data for the design process of the affected components. This may result in oil-injected screw compressor systems with a tolerable level of emitted sound and with less risk of resonances.

2. The newly developed oil-injected process gas screw compressor

Changing market demands and a rising variance of complex processes need a continuous adjustment and extension of the product portfolio. Thus MAN Diesel & Turbo newly developed the oil-injected process gas screw compressor series (CPO) to offer a product which enables additional areas of operation.

Contrary to dry process gas screw compressors oil is injected directly into the compression chamber which cools the compressed gas immediately. The oil also has a lubricating nature which makes a separate timing gear unnecessary, since the rotors directly mesh.

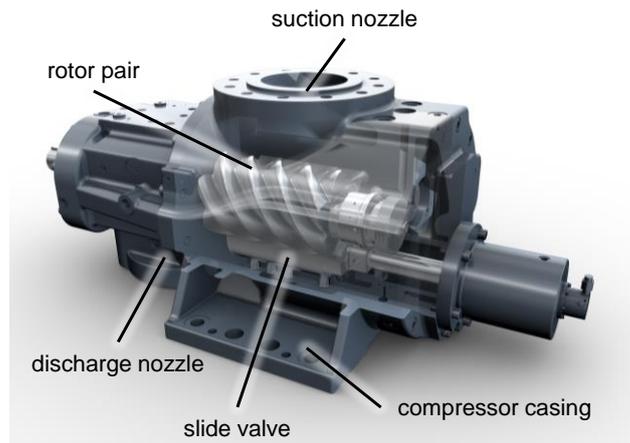


Figure 1: CPO series [source: MAN Diesel & Turbo SE]

The market for oil-injected screw compressors is growing since this type of screw compressor can be utilized for certain operation conditions more flexibly. Compared to dry process gas screw compressors one advantage is to achieve higher pressure ratios (up to 25) with only one stage. The in-built slide valve allows to continuously adjust the compressed capacity while operating. That can be realised since the position of the slide valve influences the size of the effective compression chamber between 10% and 100% of capacity.

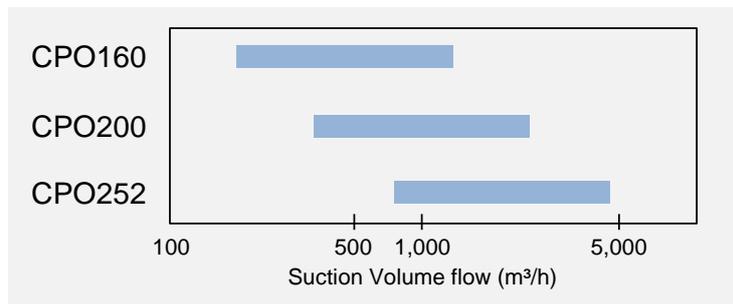


Figure 2: Suction volume flow of the CPO series [source: MAN Diesel & Turbo SE]

MAN Diesel & Turbo has developed an oil-injected screw compressor which is exclusively dedicated for the compression of process gases. It is conform to the API 619 (American Petroleum Institute) and can be equipped with a comprehensive instrumentation. If required, probes to monitor shaft vibration, axial shaft position, casing vibration and bearing temperature can be installed. The CPO series operates with a 5/7 rotor profile. The main rotor is directly driven without a separate gear box. Three different frame sizes are realised which can cover a wide range of the suction volume flow (Figure 2). Journal bearings are used to carry the radial and axial loads. This type of bearing allows long maintenance intervals respectively a long period of use.

MAN Diesel & Turbo is known for its reliable complete train solutions which allow realising different processes with many combinations of gas flow and pressure.

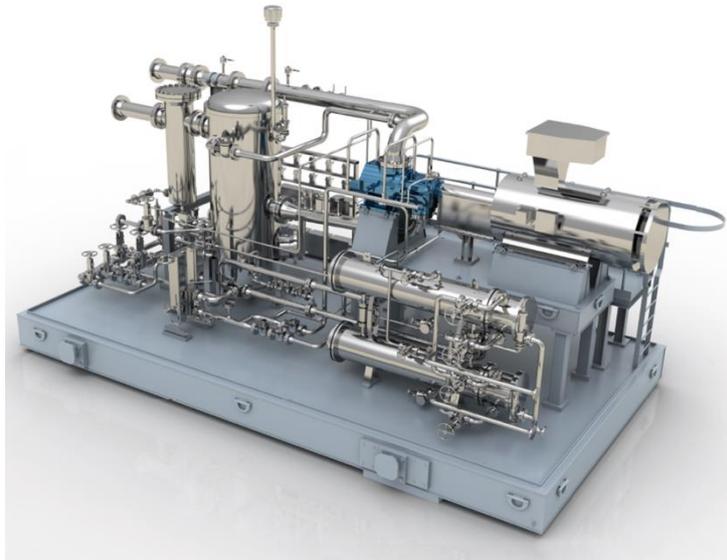


Figure 3: Package of a complete train [source: MAN Diesel & Turbo SE]

This package includes the electric drive which is directly connected to the compressor. The entire oil supply (for oil injection, bearing, sealing) is also included into the package which contains the oil pumps, the cooler and the filter. Utilising the oil separation system the injected oil can be recovered from the gas stream before entering into the customer's process. The recovered oil is reused for the oil supply.

3. Pulsation as a design criterion for the CPO

Pulsation is a typical characteristic of positive displacement machines. It is a crucial aspect which influences the design and dimensioning of several components. It occurs because of the repeating release of compressed gas when the compression chamber opens on the discharge side of the compressor. The characteristic of the opening of the compression chamber depends on the type of compressor [1]. Screw compressors operate with an internal compression which might be different from the desired discharge pressure. There is an over-compression if the compressor has a higher internal compression than the following process demands. That means every time the compression chamber opens a pressure surge moves into the following system. An under-compression takes place, if the internal pressure of the compressor is lower than the following process demands. When the compression chamber opens, a pressure surge moves against the direction of flow into the compressor. It is commonly known that these pulsation effects of over- or under-compression can be considered as source of sound. Especially the discharge port is the major source of pulsation since the described effects take place here [2].

The excitation frequency can be determined straight forward since it directly depends on the rotor profile and the rotational frequency. The number of lobes of the male rotor describes the

number of compression chambers which release compressed gas into the outlet port per rotor rotation. The number of lobes multiplied by the rotational frequency is called pocket passing frequency (PPF) [1].

$$n \left[\frac{1}{\text{min}} \right] \cdot \left[\frac{\text{min}}{60\text{s}} \right] \cdot \text{lobes} = PPF$$

As mentioned before the CPO series operates with a 5/7 profile. That means 5 lobes are located on the male rotor and 7 lobes on the female one. Since the CPO series is directly driven by an electric motor and no additional gear box influences the rotational speed, one can compute four pocket passing frequencies for the four possible rotational input speeds.

Table 1: Rotational speed and resulting pocket passing frequency (PPF) [MAN Diesel & Turbo SE]

Rotational input speed (male rotor)	Pocket passing frequency (PPF)
1500 1/min	125 Hz
1800 1/min	150 Hz
3000 1/min	250 Hz
3600 1/min	300 Hz

Regarding the pulsation as a design criterion the pocket passing frequency and its multiples can be considered as the excitation frequencies for the entire system. The rotational speed of one system does not change while operating that is why only one main excitation frequency for one observed process is effective.

The knowledge about the actual pulsation of the CPO helps to reliably predict the radiated sound of the system as well as to design components that might be affected by the pulsation. The real prototype testing allows to collect accurate data of the actual pulsation which is caused by the compressor. The pulsation can either be measured directly or estimated based on alternative measurement methods.

Since oil-injected screw compressors only have a lower level of acoustic emission there is no dedicated silencer included in the system layout. This function is realised by the oil separator. The intersection from the discharge pipe to the separator vessel provides a sudden change of the volume and cross sectional area which leads to a change of the acoustic impedance and damping characteristics. The oil separator (vessel and internals) is directly affected by the pulsation because it is connected to the discharge nozzle of the compressor just by a pipe. While designing the separator it has to be ensured that the occurring pulsation does not stimulate any harmful vibrations which could lead to an increased acoustic emission or even some damage. Some realistic examples are given in [3]. Because of the huge surface of the separator it has to be avoided that it becomes a source of sound itself which significantly

contributes to the overall sound power level. If the damping characteristic of the first stage separator is known, the estimated remaining pulsation can be used as input data for the design of the second stage separator namely the coalescing stage.

The similar aspect applies to the interconnecting piping within the system. It connects the compressor stage with the oil separator. If the excitation frequency which originates from the compressor matches the natural frequency of the pipe intense solid-borne sound is generated which lets the pipe become a separate source of sound. Within the pipe different fluid vibration modes may be present (transversal and longitudinal). These modes depend on the pipe geometry, the wavelength of the fluid vibration as well as the condition at the pipe terminations. The modes of a vibrating pipe wall also depend on its geometry but additionally depend on the pipe material [5].

But it is not only the obvious components which might be affected by the pulsation. There are also different kinds of valves or small bore piping and tubing which can be influenced. Fluid-borne pulsation or even solid-borne pulsation can excite the small bore piping of the oil system. If the natural frequency of the compressed gas matches the excitation frequency the resonance may lead to stationary waves accompanied by high frequent noise which contributes to the overall sound power level of the system.

The CPO series' slide valve is one component that might be influenced by the pulsation but which could also cause a pressure impulse itself. The slide valve limits the compression chamber so it is directly influenced by the effects of the over- or under-compression. The gas forces act on the valve which results in reacting forces acting themselves on the hydraulic system which is operating the slide valve. So the entire hydraulic system is directly influenced. Thus the pulsation needs to be considered for the design of the piping and the valves. Another effect may occur when the slide valve moves from any part load position to the zero load position. The compression chamber suddenly becomes ineffective since there is a direct bypass between the suction side and discharge side. This bypass results in a pressure impulse which affects both the gas stream and the hydraulic system.

Since it is not only the frequency but also the amplitude which has a significant impact, a real analysis becomes essential. The determined sound spectrum can be used as input data for the design of any connected component. The knowledge about the real sound power level is interesting to everybody involved in the engineering of a screw compressor system. This might be the manufacturer of the pressure vessel, the systems engineers planning the overall layout as well as the customer who needs to know how much remaining pulsation will be introduced into his piping system.

The screw compressor packages are usually a part of a complex assembly. Since human beings might be present it is important to reduce the radiated sound power. To evaluate the sound situation and its physiological influence on the human body the A-weighted sound pressure level is used. Based on the radiated sound power level of the system the sound pressure level is usually calculated at a distance of one meter from the system's border. If this level exceeds a certain threshold an additional sound reducing provision becomes necessary. Dry screw compressors are often installed with a noise hood and silencers on the suction and discharge side. The radiated sound power level of oil-injected screw compressors is lower than at the dry screw compressors, which allows an assembly without a noise hood or any dedicated silencer.

4. Sound analysis

During prototype testing the stages of the CPO are thermodynamically and mechanically tested. Stationary operation points with respect to suction pressure, discharge pressure and rotational speed have been analysed. The capacity has been continuously varied by operating the slide valve from full load operation to part load operation. Besides the stationary operation points transient conditions of the compressor have been measured as well. With respect to the thermodynamic and mechanic analysis the power consumption and the volumetric efficiency have been investigated in particular.

Another aspect of the prototype testing was an extensive analysis of the radiated sound power. All stationary and transient conditions have been acoustically measured. The findings of this analysis can be utilised in different ways. Reliable predictions as to the radiated sound can be given to the customer. Ideally these predictions are made as radiated sound power level since it describes the sound emission of the compressor independently from its point or situation of erection. With respect to its physiological effect on the human body the description of the radiated sound is usually done as A-weighted sound pressure level in one meter distance of the system's boundary.

Figure 4 qualitatively shows the range of operation of the oil-injected process gas screw compressor considering the suction pressure and the discharge pressure. The coloured area indicates possible operation points where pressure ratios between 3 and maximal 25 can be realised.

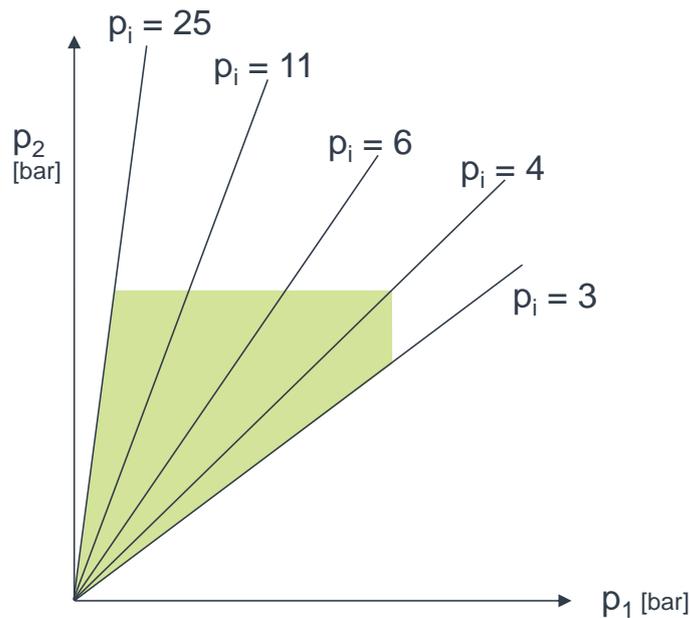


Figure 4: Operation range of the CPO [source: MAN Diesel & Turbo SE]

4.1 Test setup

Figure 5 shows the test setup for the prototype of the newly developed oil-injected process gas screw compressor. The compressor is driven by a speed-controlled electric drive.

The gas stream within the setup is realised as a circular flow. The suction pressure and discharge pressure can be adjusted independently. The amount of injected oil as well as the oil supply of the bearings and sealing are also adjusted electronically. The slide valve piston is operated by a separate oil supply unit which allows more flexibility during the testing. The injected oil is separated from the gas stream by an oil separator, which is similar to the setting of the future compressors. During the testing air was used as substitute gas.

The operation points have additionally been modified with respect to rotational speed and slide valve position which allows to influence the delivered capacity while operating with a constant pressure ratio [4]. Changing the rotational speed influences the delivered volume directly. The variation of the slide valve position leads to a change of the volume of the compression chamber which also influences the capacity.

For the thermodynamic and mechanic testing typical measurement systems have been installed like temperature sensors, pressure sensors as well as rotational speed sensors and torque sensors. To measure the emitted sound different kinds of measurement techniques have been deployed.

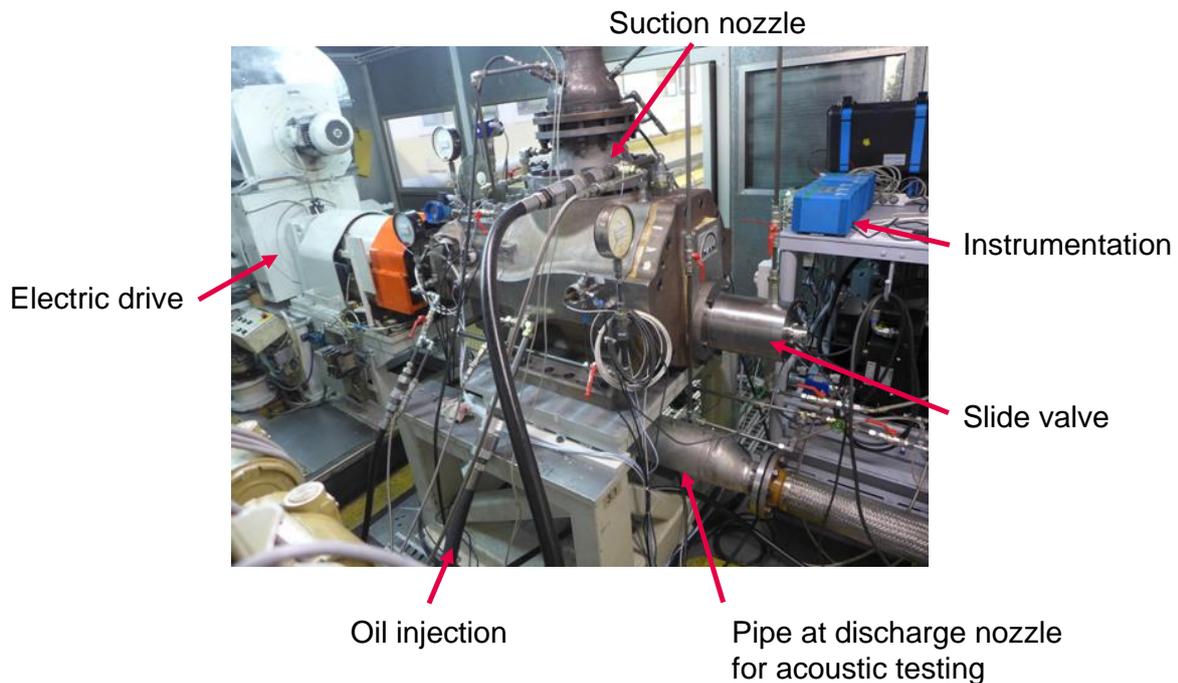


Figure 5: Test setup of CPO [source: MAN Diesel & Turbo SE]

4.2. Measurement of the sound emission

By means of the measurement of the sound emission the dependencies between the sound levels and variable operating conditions shall be analysed. Furthermore by investigating the pulsation considered as a design criterion one can provide input data for downstream components. Another aim of the investigation is to verify if the direct measurement of the internal sound pressure level can be substituted, under certain assumptions, by a measurement technique which can be deployed more easily. If comparable results can be gained without receiving an unacceptable failure more complex methods can be avoided.

4.2.1. Measurement of the internal sound pressure level

Several dynamic pressure probes are installed in the discharge pipe at the discharge nozzle of the compressor (Figure 6). Using these sensors one can measure the internal sound pressure level directly. The tips of the sensors are installed wall-flush with the internal surface of the pipe to prevent the generation of additional sounds which might be caused by the overflow of the fluid. Since the length of the discharge pipe is relatively short the decrease of the sound pressure level due to internal friction and damping can be neglected [5]. Hence the sound pressure level of the compressor which is introduced into the pipe remains constant along the pipe. There are five probes installed along the pipe wall with equal distances. Optionally additional probes can be installed along the circumference. This assembly helps to analyse the accurate modes (transversal or longitudinal) in case one discovers any suspicious acoustic

phenomena like resonances. The actual internal pulsation can be measured by using dynamic probes while the signal is gathered as a sound pressure level in dB.

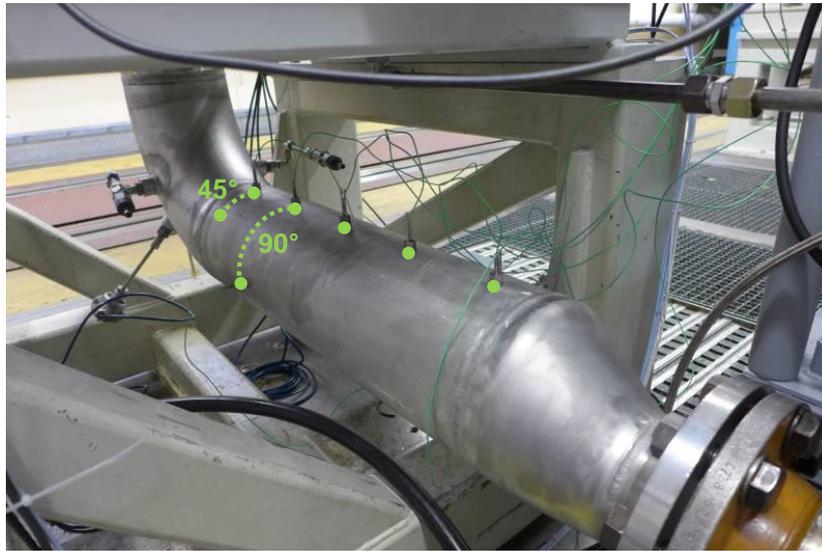


Figure 6: Discharge pipe with pressure probes [source: MAN Diesel & Turbo SE]

Assuming that in later installations of the CPO the discharge pipe is also relatively short the measured sound pressure level can be considered as input data of the criterion pulsation. This data will be used to define the layout of following components like the oil separator.

By applying the calculation method which is introduced in 4.3.1. one can calculate the external sound power which is radiated by the pipe based on the internal sound pressure level.

4.2.2. Measurement of the external sound power level

The actual external sound power level can be gathered by the use of an intensity probe. Usually this is done by the two microphone method where the probe measures the actual sound intensity (product of sound pressure and sound velocity) which leads to the sound power level when it is integrated about a certain area. The probe continuously scans a defined area with a certain distance to the object surface. By performing these near field measurements one can acoustically survey separately the compressor stage and the discharge piping as major acoustic source. In this case it is assumed that the compressor and the piping radiate the acoustic power evenly through a half sphere respectively through a cylinder. Here the sound power level can be measured more easily at fewer positions. The gathered actual sound power level can now be compared to the calculated sound power level as described in 4.3.1. and 4.3.3.. In case this comparison shows similar levels for certain frequencies one can assume that the calculation method can be reliably deployed.

4.2.3. Measurement of the solid-borne sound

In addition to the pressure probes mentioned in 4.2.1 acceleration sensors are mounted on the discharge pipe wall and measure the solid-borne sound. According to [5] the radiated sound power level can be calculated based on the sensor signal (see 4.3.3). This level can be compared with the calculated sound power level which is calculated based on the measured internal sound pressure level (see 4.3.1.). As already mentioned in 4.2.2. here applies the same. If the measured values match the calculated values for certain frequencies the calculation method can be reliably deployed both ways. The method of measuring the solid-borne sound might be very helpful in measurement environments which contain a lot of background noise or sound reflections. The measured acceleration is clearly assigned to its source of sound.

4.3. Calculation methods

The VDI Guideline 3733 describes a calculation method which allows to calculate the external radiated sound power level which is originated from an internal sound pressure level which is introduced from an external source of sound into a pipe. If this calculation method is inverted one can measure the external radiated sound power level and can calculate back to the actual internal sound pressure level that is responsible for it. The external sound power level can be measured directly deploying an intensity probe (see 4.2.2.) or it can be calculated from the signal of the mounted acceleration sensor on the pipe wall (see 4.2.3.). The internal sound pressure level is gathered directly by dynamic sound probes.

4.3.1. Calculation of the sound emission (forward calculation)

Based on the internal sound pressure level L_{pi} which is introduced into the pipe by an external source of sound one can compute the internal sound power level L_{wi0} by adding the measuring area dimension L_S . The measuring area dimension L_S is calculated from the cross section of the pipe. Depending on the type of pipe and kind of fluid further correction factors might be subtracted. K_m takes into account that in real applications the signals of wall-flush installed pressure probes have to be corrected due to acoustic effects caused by pipe form pieces and due to reverberation [5]. The impedance correction K_0 needs to be subtracted if the characteristic impedance of the compressed fluid is different from the impedance of air [5].

$$L_{wi0} = L_{pi} + L_S - K_m - K_0$$

The considered pipe at the discharge nozzle of the compressor is relatively short, that is why the decrease of the sound pressure level due to internal friction and damping can be

neglected. So one can assume that the sound power level L_{wi0} of the compressor which is inducted into the pipe remains constant along the pipe.

$$L_{wi0} = L_{wil}$$

The external sound power level L_{walm} which is radiated from the considered pipe can be calculated by subtracting the pipe sound deduction index R_R from the internal sound power level. Under certain circumstances a geometric auxiliary quantity H_g and a correction index K_α will be added.

$$L_{walm} = L_{wil} - R_R + H_g + K_\alpha$$

The pipe sound deduction index R_R quantifies the decrease of the sound power level from the internal level which is introduced into the pipe wall to the external level which is radiated from the pipe [5]. Certain material characteristics and geometrical characteristics as well as frequency-dependent effects are considered by the sound deduction index R_R .

It is known that there are uncertainties for lower frequencies [5] which might lead to failures in the estimation of R_R which subsequently might result in discrepancies between the theoretically calculated sound power levels and the actual measured sound power levels.

Furthermore the pipe sound deduction index R_R is influenced by the following factors [5]:

- Forms of vibration of the sound source supplying the pipe
- Distance between the pipe section and the sound source
- Discontinuity points within the pipe
- Type of pipe connection
- Type of pipe fixing
- Flow rate
- Introduction of solid-borne sound

With respect to the prototype testing the distance between the compressor and the pressure probes is relatively short. Thus a significant excitation of the pipe can be assumed. The influence of pipe fixing and the flow rate can be neglected since no additional pipe fixing is installed and the Mach number within the pipe is notable lower than 0.1 Ma. All other factors are system specific [5] and may influence the pipe sound deduction index R_R in this described case.

The calculated external sound pressure level which is radiated from the considered pipe will be A-weighted L_{wAalm} which allows to use the calculated values practically. By defining a distance from the system's boundary one can compute a certain measuring area dimension L_s . Assuming an even radiation of the acoustic power through a cylinder the external sound level pressure L_{pAa} is calculated.

4.3.2. Calculation of the internal sound pressure level (backward calculation)

The entire mathematical function as described above can be inverted. This allows to conclude on the internal sound pressure level based on an existing external sound power level. If there is a measured A-weighted sound power level L_{wAalm} which is radiated from a pipe it has to be converted to the linear sound power level L_{walm} before the described pipe sound deduction index R_R will be added. If necessary the geometric auxiliary quantity H_g and the correction index K_a will be subtracted which will lead to the internal sound power level L_{wilm} within the pipe.

As already mentioned in 4.3.1. due to the relatively short pipe section the influence on the sound pressure level due to internal friction and damping can be neglected. As the pipe cross section is known the measuring area dimension L_S can be calculated and will be subtracted from the internal sound power level L_{wilm} . Depending on the type of pipe and kind of fluid further correction factors might be added to finally end up at the introduced internal sound pressure level L_{pi} [5].

4.3.3. Calculation of sound emission based on solid-borne sound

The fluid-born sound energy within the pipe introduces sound energy into the piping wall. The resulting solid-born sound can be measured by acceleration sensors which are mounted outside onto the pipe wall (see 4.2.3.). The gathered acceleration level L_a has to be converted into the velocity level L_v [6].

$$L_v = L_a - \left(20 \lg \frac{f_m}{f_0} - 10 \right) dB$$

f_m – Terz-band centre Frequency, $f_0 = 1$ Hz – reference frequency

Furthermore the radiation efficiency L_σ and the measuring area dimension L_S are added to the velocity level L_v which leads to the external emitted sound power level L_{wa} of the considered pipe [5].

$$L_{wa} = L_v + 10 \lg \sigma + 10 \lg \frac{S_R}{S_0} dB$$

The radiation efficiency L_σ considers similar to the pipe sound deduction index R_R component specific and frequency-dependent characteristics [5]. To enable an easy comparison between the measured and calculated levels the external emitted sound power level will be A-weighted L_{wAa} .

4.4 Evaluation of the measurements

Since the discussion of the thermodynamic and mechanic prototype testing is not part of this paper the following explanations will only focus on the evaluation of the sound analysis. One operation point will be used as an example to discuss the observed characteristics and findings regarding the sound analysis. At this operation point the compressor has been operated with high rotational speed, maximum capacity (slide valve position 100%) and a pressure ratio of 11.

One of the findings with respect to the overall sound power level is that this level is mainly dependent on the rotational speed. It is less dependent on the discharge pressure or the position of the slide valve.

4.4.1. Description of the results

On the basis of one exemplary operation point the measured and calculated levels will be compared and discussed.

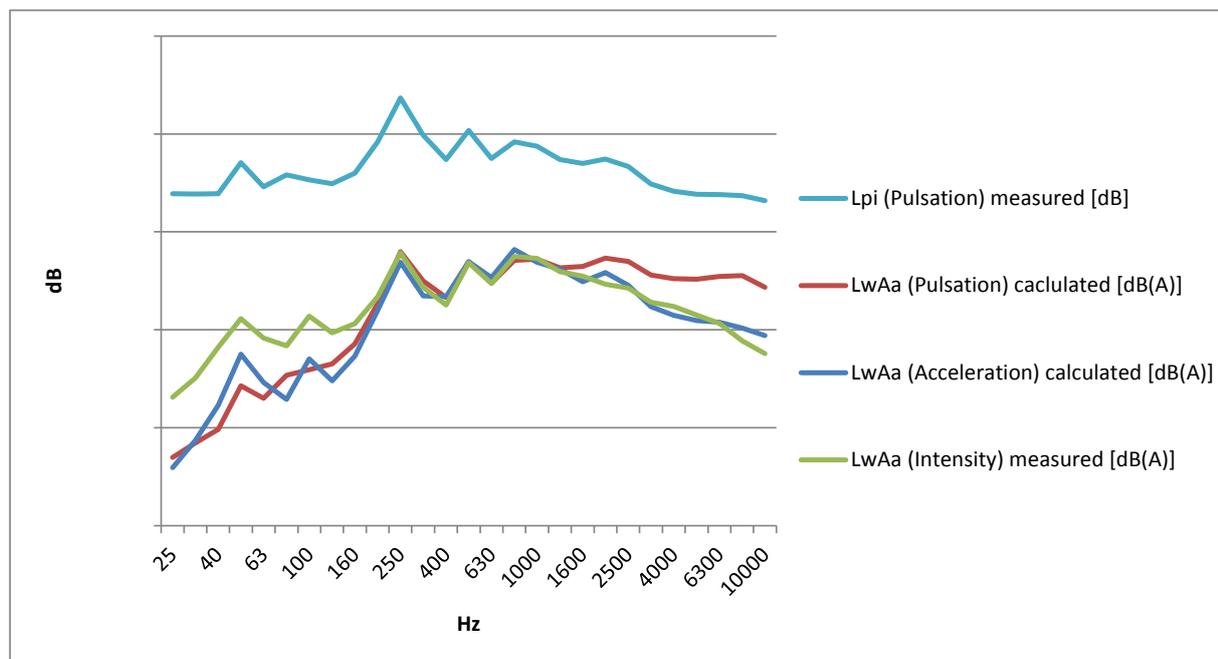


Figure 7: Comparison of measured and calculated levels [source: MAN Diesel & Turbo SE]

Figure 7 shows the various measured and calculated sound levels as frequency spectrum with its level amplitudes in dB. This is the internal sound pressure level L_{pi} (Pulsation) which is introduced by the compressor into the pipe. It is measured by the mentioned dynamic pressure probes. According to the calculation method and based on this internal sound pressure level in 4.3.1. the A-weighted external sound power level which is radiated from the pipe is calculated L_{wAa} (Pulsation) and shown. Furthermore the A-weighted sound power level L_{wAa} (Intensity) which has been measured directly by the intensity probe is shown as well. The remaining

graph represents the A-weighted external sound power level L_{wAa} (Acceleration) which has been calculated from the measured signal of the acceleration sensors. So the graph of the sound pressure level L_{pi} represents the situation inside the pipe while the three other sound power levels L_{wAa} represent the situation outside the pipe where discrepancies between these graphs might be investigated.

For the further description of the calculated and measured data the sound spectrum is divided into three ranges. There is one low frequency range ($f < 200$ Hz), one mid frequency range ($200 \text{ Hz} < f < 2000$ Hz) and one high frequency range ($f > 2000$ Hz). The borders of the frequency ranges might shift a little bit depending on other operational parameters like the rotation speed for instance.

Within the range of low frequencies there is a quite good equality between the calculated sound power level L_{wAa} (Acceleration) and L_{wAa} (Pulsation). At the same Terz-band centre frequency both are considerably lower than sound power level L_{wAa} (Intensity) which has been measured by the intensity probe. One can conclude that within the low frequency range more sound power is radiated than one might expect when deploying the calculation method.

Within the mid frequency range ($200 \text{ Hz} < f < 2000$ Hz) the calculated and measured sound power levels are almost equal. When deploying the calculation method one would compute fairly well the measured levels.

Coming to the high frequency range ($f > 2000$ Hz) the measured sound power level L_{wAa} (Intensity) and the calculated sound power level L_{wAa} (Acceleration) seem to be equal. The calculated sound power level L_{wAa} (Pulsation) is considerably higher than both the latter. This means that less sound power is radiated than one might expect when deploying the calculation method.

4.4.2. Interpretation

For the interpretation of the findings one has to keep in mind that measured and calculated data have been compared. Discrepancies between the measured and calculated values on the one hand might be led back to uncertainties of assumptions made within the calculation method. On the other hand discrepancies might be explained by uncertainties allocated to the measurement techniques or to disturbances within the measurement environment. Further investigation is needed to clarify the dependencies. In particular one has to analyse which elements of the calculation method might offer potential for uncertainties.

The comparison of the measured and calculated sound levels shows that there are discrepancies in the lower and higher frequency ranges while in the mid frequency range a good correlation can be observed. The mid frequency range contributes significantly to the

overall sound power level, since within this range the single Terz-band centre frequencies provide the highest levels. If the calculation method as described in 4.3.2. is used to calculate backwards to the internal sound pressure level one will receive reliable solutions within the mid frequency range.

The elevated sound power level L_{wAa} (Intensity) compared to L_{wAa} (Pulsation) within the lower frequency range might be explained by disturbances within the measurement environment which could affect the signals of the intensity probes more than the signals of the acceleration sensors. This discrepancy will lead to calculated internal sound pressure levels L_{pi} higher than they actually are. That means that the calculation within the low frequency range is a little bit conservative and provides some safety margin. If the calculated sound spectrum is used as input data for downstream components like separators they will be planned more properly than necessary which can be accepted anyway. Furthermore the low frequency range contributes only little to the A-weighted overall sound power level so that a discrepancy might be tolerable. Within the low frequency range the calculated sound power levels L_{wAa} (Acceleration) and L_{wAa} (Pulsation) are almost equal. In case measured data of an acceleration sensor is available one can deploy the calculation method backwards and will receive reliable solutions for the internal sound pressure level L_{pi} .

If it comes to the high frequency range the measured sound power level L_{wAa} (Intensity) is below the calculated sound power level L_{wAa} (Pulsation). If the calculation method is deployed backwards one will compute an internal sound pressure level L_{pi} which is lower than it actually is. This case might not be acceptable in terms of the design criterion since the calculation will lead to a notable failure and a design based on these values might be undersized. So this case needs further action.

The described discrepancies can be found at all operation points in similar characteristic which leads to the conclusion that this might be a frequency-dependent effect. If this discrepancy between measured and calculated levels shall be further investigated it is worthwhile to have a deeper look at the single elements of the calculation method. The pipe sound deduction index R_R consists of several terms. [5]

$$R_R = C + 10 \lg \left(\frac{c_R \cdot m''_R}{\rho_F \cdot c_F \cdot d_i} \right) + A(f)$$

c_R – Sound velocity of the longitudinal waves in the pipe wall

c_F – Sound velocity in the fluid

m''_R – Weight per unit area of the pipe

ρ_F – Density of the fluid

d_i – Pipe inside diameter

The constant C is frequency-independent. The following term considers certain material characteristics and geometrical characteristics and can be considered constant for one operation point, whereas the last term $A(f)$ is frequency-dependent.

Within the high frequency range both sound power levels, the calculated L_{wAa} (Acceleration) and the L_{wAa} (Intensity), are below the calculated L_{wAa} (Pulsation). Thus the damping characteristics of the real pipe seem to be higher than considered in the VDI 3733 calculation method within this frequency range. The proportions of the installed pipe have to be considered, since the wall thickness is relatively thick compared to the diameter of the pipe which may cause higher damping characteristics. To verify this assumption and to analyse the discrepancies within the low frequency range further investigation will be carried out during prospective testing.

An approach to explaining the discrepancies between the measured and calculated levels are the already mentioned uncertainties in estimating the pipe sound deduction index R_R especially within the low frequency range [5]. Theoretically one can develop a frequency-dependent correction factor which compensates the discrepancies. Developing such a correction factor means to investigate the dependencies first and to interpret them physically and technically.

It is not the aim of the authors to modify the calculation method or the way how to estimate the pipe sound deduction index R_R . Rather a practical and feasible approach shall be developed which allows estimating the internal sound pressure level using the presented calculation method despite possible uncertainties. In addition the findings of the analysis of the gathered data will also be taken in consideration.

Figure 8 shows the characteristic of the three discussed sound power levels in an abstracted manner.

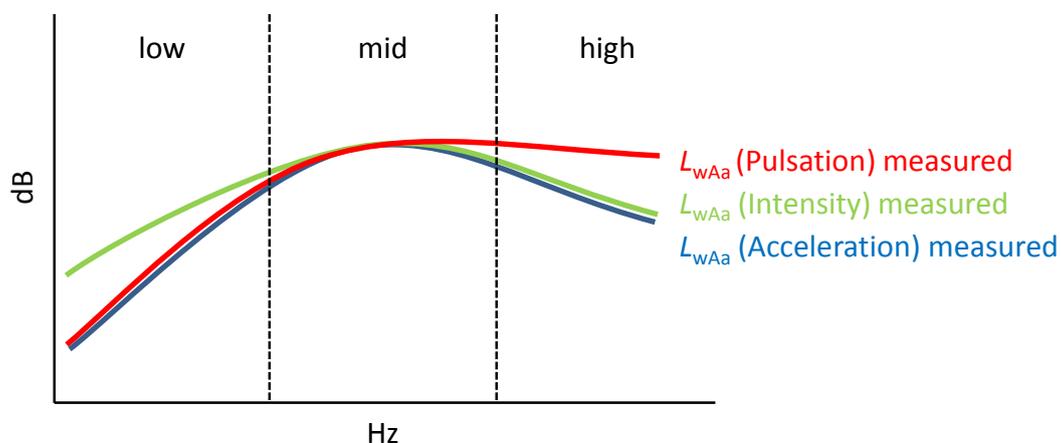


Figure 8: Abstracted manner of the sound power level [source: MAN Diesel & Turbo SE]

Assuming only the signal of an intensity probe is available, that means that the external radiated sound power level L_{wAa} (Intensity) can be used to estimate the internal sound pressure level L_{pi} . One can calculate conservatively within the low frequency range that will lead to a certain safety margin which can be accepted as described above. Within the mid frequency range the backwards deployment of the calculation method can reliably be deployed. Difficulties will occur within the high frequency range since the backwards calculation will lead to an unacceptable failure. The calculated internal sound pressure level will always be lower than it actually is. With regard to a practical approach one can use the findings of the acoustical analysis. The sound power level L_{wAa} (Pulsation) which is calculated based on the internal sound pressure level has almost the same constant level within the mid frequency range and the high frequency range. If only the sound power level L_{wAa} (Intensity) is available one can substitute the measured level within the high frequency range by the measured level of the mid frequency range. If the backwards calculation method is deployed using these substituted sound power levels the solution will be closer to the actual internal sound pressure level than without substitution.

5. Conclusions

An acoustic analysis for the newly developed oil-injected process gas screw compressor has been performed during the prototype testing. This paper presents the calculation method according to the VDI guideline 3733 [5] which allows to compute the external radiated sound power level of a pipe based on the measured internal sound pressure level. It has been shown that this method can be inverted which, based on the measured external radiated sound power level, enables one to conclude the internal sound pressure level which is introduced into the discharge pipe by the compressor. Depending on the frequency the inverse function leads to results close to the actual internal sound pressure level. Within the low and high frequency range the calculated levels do not match the actual levels. Thus a practical approach has been introduced which allows to receive results which are yet close to the actual internal sound pressure level. The mid frequency range contributes significantly to the overall sound power level. Within this range the calculation method can be reliably utilised both ways without expecting considerable deviations.

Regarding the pulsation as a design criterion this approach helps to provide sufficient input data which can be considered for the design of components of the entire system.

Future prototype testings will take place under different and extended operation conditions. Additional sound analyses will be performed to verify the recent findings. In case of similarities one could reliably utilise the introduced approach to estimate the internal sound pressure level

inside the pipe. Simple measurement techniques could be applied rather than complex ones, which reduces the effort of testing.

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