Vibration analysis in reciprocating compressors

V Kacani
Leobersdorfer Maschinenfabrik GmbH, Vienna, Austria
Vasillaq.Kacani@LMF.at

Abstract. This paper presents the influence of modelling on the mechanical natural frequencies, the effect of inertia loads on the structure vibration, the impact of the crank gear damping on speed fluctuation to ensure a safe operation and increasing the reliability of reciprocating compressors. In this paper it is shown, that conventional way of modelling is not sufficient. For best results it is required to include the whole system (bare block, frame, coupling, main driver, vessels, pipe work, etc.) in the model (see results in Table 1).

1. Introduction
The main dynamic loads on reciprocating compressors are: the shaking forces due to pressure pulsation, inertia forces and gas forces, and additional forces due to torsion vibration of the drive train. The shaking forces occur typically when there is a change in pipe direction or area, at cylinder passages, on the ends of the pulsation dampers, elbows, reducers, orifices, tee pieces and on other pipe components. The inertia forces are caused by oscillating of reciprocating masses and inertia forces act through the cross head on the structures and on the main bearing of the crankshaft. The cylinder gas forces are generated within the cylinder during the working cycle: suction, compression, discharge and expansion. The gas forces are generally the highest one and act on cylinder covers and through the crank mechanism. They also act on crosshead guide and on crankshaft main bearing. These loads lead to vibration of the compressor components. Modeling has a direct impact on the modal analysis and the calculation of the forced vibration. The modeling criteria as well as the pre-stressed effects of the structures due to internal pressure, temperature and the misalignments during the assembly of the components, are of importance for the calculation of natural mechanical frequencies and the system’s response under dynamic loads. Moreover, modelling of the structure, modelling of the crank gear, the damping factors, and the damping ratio of the mechanical structure are decisive for the dynamic behavior of the drive train and also for the design of other mechanical components.

The influence of inertia and cylinder gas forces, in addition to the pulsation shaking forces are included in the forced-mechanical-response analysis of the compressor mechanical model. This requires an adequate finite element model of reciprocating compressors, specifically of the crankshaft, of the crankcase, of the distance pieces and of the pulsation suppression devices. For these components 2D and 3D-modeling is used. This model also enables an undertaking of the lateral dynamic analysis of the crankshaft and the stress evaluation under torsion and bending [3], [11]. The calculation of the torsional/lateral vibration mode shapes and the mechanical natural frequencies of crankshaft with flywheel will be performed with a separate 3D-model.
2. Vibration analysis of the complete compressor unit

The compressor unit consists of all in-skid components such as compressor manifold, piping system, e-motor / engine, pipe, base frame, pipe/vessel supports, anti-vibration mounts etc. The skid is connected on its termination points to the off-skid piping system. In different regulations and standards [2], [3], [11] are included recommendations and proposal regarding the scope of supply, limits of the system as well as the exciting loads for the vibration analysis of the compressor mechanical model. For many reciprocating compressor units, such as off-shore applications or compressors mounted on a skid, the recommended limits of the system should be extended, at least, up to base frame - foundation interface. If available, the anti-vibration mounts should be included in the analysis. Not only the cylinder gas forces but also the inertia loads must be taken into consideration while performing the mechanical response analysis of the reciprocating compressor systems. The goal of vibration analysis is the determination of the dynamic load on all components of compressor unit and a comparison with the maximum allowable values [2],[11]. This requires an accurate modeling of all parts and adequate definition of the connecting zones (interfaces) between all components involved.

2.1. Modeling

The modeling of reciprocating compressors requires much experience and represents a balancing act. On the one hand, the model should be as simple as possible to reduce calculation time and on the other hand as fine as necessary to get the correct technical solutions. The preprocessing is the most time consuming step while undertaking vibration analysis. In this paper an automatic parametric modeling of the components of the reciprocating compressor is shown. The script language APDL [5] (ANSYS Parametric Design Language) is used for modelling of all components. Special interfaces are defined to get the dynamic force members on bolts or contact areas/zones for all connecting parts. Generally, the crankcase is rarely modelled. The modelling effort for the crankcase may be expensive but it is absolutely necessary for the vibration analysis. Using the parametric design language, the modeling effort can be considerably reduced. The correct influence of cylinder gas forces and crank gear mass forces on vibrations of the machine can only be achieved by an adequate modelling of the crankcase, cross head, distance piece, and pulsation suppression devices. The use of 1D pipe or beam elements is only for some components such as pipes, tee-pieces support beams [6] admissible. Crankcase, distance piece, base frame and pulsation suppression devices should be modeled at least with 2D shell elements. The recommended system limits, specified in the API 618 [11] needs to be expanded with all components up to the base frame, including if available the anti-vibration mounts. The following figures show the finite element models for crankcase, crankshaft, flywheel, cylinder, cylinder supports, piping systems, pulsation suppression device, e-motor, base frame, piping and vessel supports. The bolts are modeled with pretension elements and are connected to the structure via spider with MPC-elements [5]. Figure 1 shows the finite element model of the base frame, including the pedestal for the reciprocating compressor and e-motor as well as the base plates for the vertical separators: a) on-shore application, the skid is connected with bolts to the concrete fundament b) off-shore application, the skid is isolated, supported on anti-vibration mounts (AVM). The bolts and AVM are part of base frame model. Figure 2 shows the crankcase manifold including the crankshaft, flywheel and the distance piece. The presented model is also suitable for the dynamic lateral analysis of the crankshaft. For the crankshaft main bearings, spring elements with appropriate stiffness and damping were used.
Figure 1. FE-Model of base frame and pedestal: a) onshore-application b) offshore- application.

Figure 2. Crankcase, flywheel, crankshaft, main bearing, distance piece

Based on the parametric technique described above, Figure 3 shows the finite element model of the two-stage offshore reciprocating compressor, horizontal arrangement, type B154, with a power of 1MW, 720 rpm, 45ton weight. The finite element model includes all components of the compressor unit: crankcase, crankshaft, flywheel, double compartment distance pieces, cylinders, cylinder supports, e-motor, pulsation suppression devices, coolers, separators, piping system, base frame including pedestal, anti-vibrating mounts and all corresponding supports.

Figure 3 a) FEM of compressor unit, b) photograph of the offshore application with anti-vibrating mounts.
2.2 Modal analysis.

After modeling the compressor, modal analysis was performed. The effect of pre-load conditions – temperature [7], [10], internal pressure, misalignment - on the Natural Mechanical Frequencies (NMF’s) must be considered. Under pre-stressed conditions the mechanical system is stiffer - hence the NMF’s are higher. In this paper the influence of the internal pressure on the NMF’s is investigated. Figure 4b) shows the circumferential mode of vibration and the corresponding natural frequency of a vertical vessel – length L=3000mm, Diameter OD=609mm, shell thickness t=15mm – depending on the internal pressure. The curve in figure 4 c) represents the ration of the frequency f(p) under internal pressure p to the frequency f0 at pressure zero. The difference between the two natural frequencies f(p) and f0 is up to 25%, and depends on internal pressure.

Further the system modelling limits have a major influence on the Natural Mechanical Frequencies (NMF’s) of compressor units. In Figure 5 three different models of a compressor unit are shown: a) pulsation suppression devices and piping system, b) crankcase manifold, piping system, cylinder and cylinder support, and c) complete compressor unit including e-motor, base-frame with anchor bolts on concrete foundation. Figures 5a) and 5b) represent a simplified model that is commonly used to carry out pulsation studies in reciprocating compressors. The third model is closer to the real system. This model must be used for simulation calculation.
The following table 1 contains the first 10 MNF’s, calculated for the three above models: a), b) and c). The differences between the natural frequencies are more than 100%. The first model is practically unsuitable for conducting vibration analysis. The model is very simple and is far away from the real system, photograph d). Only the third model can provide useful results. With this model both the dynamic response analysis and the thermal study can be performed immediately one after another. So, quickly and easy an optimal solution can be found, as the pipe thermal stress analysis is in conflict with the goals of the mechanical dynamic design.

### Table 1

| Number | Model a | Model b | Model c |
|--------|---------|---------|---------|
| 1      | 28.5    | 27.7    | 16.2    |
| 2      | 29.6    | 28.7    | 17.9    |
| 3      | 40.4    | 39.4    | 22.1    |
| 4      | 59.6    | 40.4    | 25.5    |
| 5      | 63.1    | 41.9    | 26.3    |
| 6      | 67.0    | 43.1    | 31.0    |
| 7      | 74.8    | 60.8    | 34.5    |
| 8      | 84.7    | 61.1    | 35.2    |
| 9      | 86.7    | 63.0    | 38.0    |
| 10     | 92.7    | 63.9    | 39.0    |

The difference between the NMF’s ~75%

2.3. **Forced mechanical response analysis**

Next, the forced response analysis shall be performed. The exciter forces on the structure are the inertia (mass) forces, gas (stretching) forces in cylinders as well as the shaking forces on valve passages, cylinder passage, heads of the pressure pulsation devices, separators, coolers, pipe work components such as elbows, tee's, reducers, safety valves, etc. Figure 6 shows harmonic amplitudes of the cylinder gas force $F_{\text{Gas,axial}}$ acting in cylinder in axial direction (cylinder axis), gas and mass force $F_{\text{crosshead,vertical}}$ acting on the crosshead guide in vertical direction (vertical to cylinder axis), gas and mass force $F_{\text{Crank pin,vertical}}$, $F_{\text{Crank pin,axial}}$ acting on the crank pin in normal and axial direction. Thus there are four different dynamic exciter forces for each axis of the reciprocating compressor. Further, additional forces, resulting from the torsion should also be considered. The exciter forces must be determined for all operating conditions of the reciprocating compressor. For unloaded start-up and shut-down, only the mass forces of oscillated and rotated parts are acting on the compressor components.
Figure 6. Exciting forces of reciprocating compressor: $F_{\text{Gas,axial}}$, $F_{\text{crosshead,vertical}}$, $F_{\text{Crank_pin,vertical}}$, $F_{\text{Crank_pin,axial}}$

Figure 7 shows the results of the simulations during the unloaded start-up only under inertia forces. The results are for the offshore application shown in Figure 3. Figure 7 a) shows the dynamic loads on the Anti-Vibrating Mount (AVM) in three directions: axial $F_x$, vertical $F_y$ and horizontal $F_z$. Figure 7 b) contains the vibration velocity at inlet pipe, at cylinder 1, and at cylinder 2. Figure 8 shows the results for the onshore application: a) crankcase bolt forces and b) vibration velocities. The investigation can be done for all specified operating condition of reciprocating compressor. The last step is the thermal stress analysis which can be performed with the same model. The simulation steps - modeling, modal analysis, dynamic analysis, thermal analysis - can be repeated until the required specified criteria are met. During the simulation process also different standards, regulations and specifications [1], [2], [3], [4], [11], [12] must be considered.

Figure 7. a) Dynamic loads on AVM b) vibration velocities at inlet pipe, cylinder 1, and cylinder 2 for the offshore application in Figure 3.
3. Torsional vibration

The reciprocating compressors have an extensive drive train. There are very complex components such as, crankshaft, flywheel, electric machine, turbine, engine, fan, screw compressor, blower, viscous damper, coupling, gear unit, V-belt etc. The correct determination of all necessary data and parameters for the Torsional Vibration Analysis (TVA) is very important. This has very high impact on the quality of the calculation, on the reliability and the availability of reciprocating machine train. The main parameters of the components are the torsional stiffness, the damping factors and the excitations. The calculation of stiffness of the crankshaft of reciprocating compressors is described in detail in [8], [9]. Figure 9 shows the Finite Element (FE) model of the crankshaft and flywheel of a four cylinder 3MW horizontal compressor. The calculated 1st and 2nd Natural Torsional Frequencies (NTF) are \( f_1 = 114 \) and \( f_2 = 323 \)Hz. The mass moment of inertia \( J \) of the crankshaft is \( \sim 25 \text{kgm}^2 \). The stiffness \( C_1 \) and \( C_2 \) for one or two mass spring torsional system for the crank shaft – Figure 9 - can be determined from previously calculated NTF’s \( (f_1 = 114 \text{ and } f_2 = 323 \text{Hz}) \) and \( J \’s \) based on the corresponding frequency equations (Eq. 1), (Eq. 2) and (Eq. 3) below [13]. For two mass spring model - with \( J_1 = J_2 = J/2 = 12.5 \text{kgm}^2 \) - the calculated torsional stiffness are (from Eq. 2 and Eq. 3) \( C_1 = 21138976 \text{Nm/rad}, C_2 = 15619613 \text{Nm/rad} \).

\[
\text{Single mass – spring system :} \quad 4\pi^2 \cdot f_1^2 = \frac{C_1}{J_1} \quad (\text{Eq. 1})
\]

\[
\text{Two mass – spring System :} \quad 4\pi^2 \cdot f_1^2 = \frac{1}{2} \left[ \frac{C_1 + C_2}{J_2} \cdot \frac{C_1}{J_1} + \sqrt{\frac{C_1 + C_2}{J_2} \cdot \frac{C_1}{J_1} + \frac{C_1 + C_2}{J_2} \cdot \frac{C_1}{J_1} - \frac{4 \cdot C_1 \cdot C_2}{J_1 \cdot J_2}} \right] \quad (\text{Eq. 2})
\]

\[
4\pi^2 \cdot f_2^2 = \frac{1}{2} \left[ \frac{C_1 + C_2}{J_2} \cdot \frac{C_1}{J_1} - \sqrt{\frac{C_1 + C_2}{J_2} \cdot \frac{C_1}{J_1} + \frac{C_1 + C_2}{J_2} \cdot \frac{C_1}{J_1} - \frac{4 \cdot C_1 \cdot C_2}{J_1 \cdot J_2}} \right] \quad (\text{Eq. 3})
\]

Figure 9 Crankshaft torsional model

The relative shaft damping coefficient \( k_i \) (various couplings or shaft damping) depends on the Lehr’s damping factor and the corresponding natural torsional frequency [14]. For the absolute damping coefficient \( k_a \), there are data available only for diesel engines [13], [14]. The absolute damping factors depends on piston area \( A \), crank radius \( r \), and the type of engine represented through coefficient \( \mu \) which are included in the following equation (4):

\[
\ldots
\]

Figure 8. a) Dynamic loads on crankcase bolt b) vibration velocities at inlet pipe, cylinder 1, and cylinder 2 for the onshore application in Figure 5.
The mathematical model of each throw [9] of the crankshaft of reciprocating compressor is (Eq. 5):

\[
\dot{\phi}_i = \omega \cdot r \cdot \mu \quad (Eq. 4)
\]

\[
J(\phi) \cdot \dot{\phi}_i^2 + \frac{1}{2} \cdot \frac{dJ(\phi)}{d\phi} \cdot \dot{\phi}_i^2 + k_{a,i} \cdot \dot{\phi}_i + k_{r,i} \cdot (\phi'_i - \phi'_{i-1}) + C_{r,i} \cdot (\phi_i - \phi_{i-1})
\]

\[
- k_j \cdot (\phi'_{i-1} - \phi'_{i}) + C_j \cdot (\phi_{i-1} - \phi_i) + M_j(\phi_i, \phi'_i) = 0 \quad (Eq. 5)
\]

**Figure 10.** Crank gear equation of motion for the throw i: throw angle \(\phi_i\), angular velocity \(\dot{\phi}_i\).

For the simulation under resonance conditions the equation of motions in [8] and [9] should be extended with the equation for connecting rod and oscillating parts (crosshead, piston rod and piston). Therefore the mass, the mass moment of inertia and the center of gravity (CG) location of the connecting rod are required. The equations of motion can be derived from the kinetic energy of the system. The required torques on the shafts of driven train and the angular velocities of each mass can be obtained from the numerical solution of differential equations [15, Page 360]. The torsional vibration analysis should be performed for all operation cases of the compressor unit including: continuous operation, part-loads, start-up, shut-down under loaded and unloaded condition, emergency shut-down etc. Generally the emergency shut-down represents a critical case. Furthermore, the operation of the system under resonance condition must be taken into consideration. In this paper the calculation of speed fluctuation of the crankshaft throw, and dynamic torques under the resonance conditions – high frequency resonance with up to ~15\(^{th}\) harmonic - of the reciprocating compressor, is described. Near the resonance, high dynamic shaft torques and a high fluctuation of rated speed is expected. This will result in higher rod load and cross head normal force. Furthermore this can affect the cylinder PV-Curve and it can influence the power consumption, capacity, and the pressure pulsation. Since the absolute damping of crank gear has a strong influence on the speed fluctuation, their precise determination is of major importance. The following figures show the simulation results of torsional vibration analysis.
Figure 11. a), b) Torque amplitude, c) Speed fluctuation of crankshaft throw.

Figure 11 a) shows the torque amplitude over compressor speed for three different absolute damping factors $k_a$ of crank gear. Near the resonance the dynamic torque at the crankshaft, as expected, is very high. To show the influence of the damping the calculation was performed for three different damping coefficients: $k_{a1}=0$, $k_{a2}=5\text{Nms/rad}$ and $k_{a3}=10\text{Nms/rad}$. Damping coefficient $k_{a2}$ corresponds approximately to the above Equation (4) for diesel engine. Depending on the calculated torque loads, generally a separation margin from the critical speed is required.

Figure 11 b) and 11 c) shows the dynamic torque amplitude and the speed fluctuation versus crankshaft angle, at resonance condition for different damping coefficients.

4. Summary

- The finite element model of compressor units should include at least the following components: crankcase, crankshaft, distance piece, e-motor, pipe work, pressure pulsation devices and base-frame. A combination of 1D, 2D and 3D elements is required to realize a proper model. The crankcase, the distance piece, the pressure pulsation devices and the base-frame must be modeled with 2D elements. For some local zones 3D elements are required. The modeling methods and the scope of the system should be carefully selected. Depending on the scope of the model, the natural frequencies differ from each other up to 100% and sometimes more.
- The modal analysis as well as the forced mechanical response analysis, must be carried out at pre-stressed condition of the compressor unit. In the pre-stressed state, the system is stiffer, and consequently the natural frequencies are higher. For vessel under internal pressure the NMF’s are up to 25% higher compared to the natural mechanical frequencies for the unloaded conditions!
- The forced vibration analysis must be performed, not only under the shaking-forces and cylinder- gas-forces as specified in API 618, but also under inertia-forces.
- With the presented FE model - Figure 5c) - can be performed successively, the forced response mechanical analysis and the thermal analysis (flexibility analysis) of the reciprocating compressor systems. So quickly and easy an optimal design of the system can be found ([11], Page 61, 62).
- The presented model allows a lateral dynamic analysis of the crankshaft, under inertia forces, gas forces as well as forces due to torsion.
• The torsional vibration analysis should be performed for all operation cases of the compressor unit including: continuous operation, part-loads, start-up, shut-down under loaded and unload-ed condition, emergency shut-down. The calculation must be carried out also for the expected torsional critical speeds (resonance condition). At the critical speed or near the critical speed the torque amplitude and the speed fluctuation are very high. This causes higher rod load and cross head normal force. This can also affect the cylinder- PV-Curve and so the power consumption, capacity, and the pressure pulsations of the machine.

References
[1] VDI 2230 Systematic calculation of high duty bolted joints.
[2] VDI 3842 Vibration in piping systems
[3] VDI 2049 Torsional vibration of driveline - Calculation, measurement, reduction
[4] ISO 10816-8:2014, Mechanical vibration - Evaluation of machine vibration by measurements on non-rotating parts - Part 8: Reciprocating compressor systems
[5] ANSYS Computer-aided engineering software.
[6] Kacani V., Huttar E., Stibi H. Torsionsschwingungen in Kolbenkompressoranlagen. Der Kolbenkompressor- eine zeitgemäße Arbeitsmaschine 4.-5. November, 1999, Dresden
[7] Kacani V., Huttar E., Economical pipe stress analysis using parametric modelling and design standards. 4th Conference of the EFRC, 2005, Antwerpen
[8] Kacani V. Huttar E., Heumesser T. Simulation of Reciprocating Compressor Start-Up. 6th Conference of the EFRC 2008, Düsseldorf
[9] Kacani V. Simulation of Reciprocating Compressor Start-Up and Shut down under Loaded and Unloaded Conditions. International Compressor Engineering Conference at Purdue, July 16-19, 2012
[10] Kacani V., Huttar H., Ognar, G. State of the Art Design and calculation of Vessels and Piping Systems in Reciprocating Compressors. ICCR, Xi’an, China.
[11] API Standard 618, Fifth Edition, 2007, Reciprocating Compressors for Petroleum, Chemical, and Gas Industry Services.
[12] FKM Rechnerischer Festigkeitsnachweis für Maschinenbauteile. VDMA Verlag
[13] Holzweißig F., Dresig H. Lehrbuch der Maschinendynamik. Springer Verlag 1979
[14] Andreas Laschet. Simulation von Antriebssystemen. Springer Verlag. Springer Verlag 1988
[15] Julien C. Sprott. Numerical Recipes. Cambridge University Press 1991