FORCED RESPONSE OF CYLINDER MANIFOLD
FOR RECIPROCATING COMPRESSOR APPLICATIONS

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ABSTRACT

The system consisting of the reciprocating compressor and associated bottles, known as the “Cylinder manifold” may potentially be the source and location of high vibration problems. Consequently special attention must be paid to the complete simulation of the system to assure smooth and safe operation.

Applicable standards specify the items to be included in the study (crosshead guides, distance pieces, cylinder flanges, joints, supports, etc.). However only a model built using manufacturing drawings and validated by site measurements can provide a sufficient accurate description of the characteristics of these critical components and therefore realistic results.

Knowledge of the frequencies and amplitudes of pulsation induced shaking forces defined by acoustical simulation, internal gas forces in the cylinder, and unbalanced mechanical forces and moments allows a proper forced response analysis of the cylinder manifold system to be performed. These forces are applied to the finite element model to calculate the relevant vibrations and stress amplitudes by performing a harmonic analysis. When the dynamic stresses are out of the limits it is necessary to go back to the cylinder manifold system analysis or to the acoustical study to find a solution using different supports, with lower shaking forces, or by modifying the volume bottle design. This enables an iterative analysis of the system until all requirements have been satisfied.

Additional results of a forced response analysis are the reaction forces on the cylinder and discharge volume bottle supports.
When the application requires a large and heavy acoustic damping system with consequently a low mechanical natural frequency, or the compressor speed is significantly high, the possibility of mechanical resonance in the first design is very high. Therefore the execution of these studies at a very early stage of the project is fundamental.

The proper solution can be found only by close cooperation between the compressor manufacturer, end user, engineering contractor and vibration specialist.

**NOMENCLATURE**

- CAD  Computer Aided Design
- CAE  Computer Aided Engineering
- CMS  Cylinder Manifold System
- FEM  Finite Element Method
- RPM  Revolutions per Minute
- RMS  Root Mean Square

**INTRODUCTION**

Since reciprocating compressors have the advantage of high efficiency and flexibility of service, they are often used in chemical and petrochemical applications. However their variable flow generates pulsations and vibrations, which may produce fatigue failure of the system, loss of capacity, and increased maintenance costs. Therefore there is constant attention to increasing the safety, performance and reliability of these machines, which requires that designers make extensive use of the most advanced simulation methods to avoid vibration of the compressor manifold, which represents the heart of the system.

The cylinder manifold analysis must be performed at an early stage of the project and each component must be adequately modelled to accurately predict the natural frequency and relevant dynamic response in terms of vibration and cyclic stress level.

This is achieved by taking into account existing codes and applying modern simulation methods, and through feedback from the site operations group.

This paper covers the existing procedures, application experience and recommendations for properly considering the applied loads.

**SYSTEM DESIGN**

**Pulsation effects**

Reciprocating compressors generate a pulsating gas flow whose harmonic components may interact with plant piping and equipment [5], inducing resonance effects including high vibrations, poor performance, noise, and high risk of fatigue failures. An accurate pressure pulsation and mechanical analysis is the means to protect plant operations by limiting their effect through proper damping and/or support stiffening.

To perform this study a digital program developed and tuned using substantial experience and extensive field measurements is necessary. In this type of analysis, the transfer matrix describes each plant section and component. Following this principle, an overall transfer matrix can be developed for the complete plant and then solved for any exciting harmonic frequency present in the periodic gas flow generated by the compressor. Next, each plant configuration is analysed in all possible operating conditions, including speed variation, capacity control range, compressors running alone or in parallel, etc. All components are included in the study: piping, cylinders, valves, orifices, dampeners, coolers, separators, etc.

Pressure pulsations and shaking forces acting on the components are calculated at all significant points so that they can be used as input for the mechanical studies.

**Initial damper selection**

The first step is the selection of the type (empty volume or filter) and size (volume) of the pulsation device.

A digital sizing program is able to investigate three different configurations, as follows:

- With choke tube (filter with one chamber)
- With choke tube and baffle (filter two chambers)
- Without internals (volume bottle)

The program calculates the volume, the size and length of choke and the relevant pressure drop, optimizing the results for all operating conditions.

The pulsation damper is sized such that the API618 STD [1] allowable pulsations are met with a sufficient margin (e.g., 70-80% of allowable), based on experience. This margin in general is sufficient to allow satisfactory control of the resonance conditions found during the final pulsation study simulating the complete plant.

**Preliminary study of compressor layout**

Reciprocating compressors require the installation of pressure vessel pulsation dampeners close to the cylinders to limit the pressure pulsations in the plant. The dampers are part of the machine itself. In general, suction dampeners are mounted on the top of the cylinders and discharge dampeners at the bottom.

During this phase attention must be given to the interface of the dampeners with the machine from the lay-out standpoint and their design must take into account the available supporting points provided on the compressor, as well as maintenance needs.

The shaking forces are generated by the action of the pressure pulsations (e.g., damper ends) on geometric discontinuities. They are responsible for the mechanical vibrations of the system.

A best practice for the design of dampers is to place the cylinder connections at the center of the bottle. In fact, the
cylinder nozzle exit in the bottle (or entrance for the suction side) is the origin of the pressure pulsations in the bottle itself. If this origin is placed symmetrically with respect to the bottle ends, the pressure pulsations will hit the two opposite sides with the same phase, and therefore the associated instantaneous forces (pressure times cross section) will be identical in amplitude and absolute phase, but opposite in direction, with a zero resultant.

However sometimes it is not possible to provide nozzles at the center of the bottle. In that case it is advisable to add internals to reach the center of the damper so as to limit the action of the shaking forces to the internals themselves.

However it must be understood that although the shaking forces can be minimized, only in specific cases can they be completely avoided.

Therefore the compressor manufacturer should indicate the allowable shaking force values for suction and discharge dampers based upon the compressor type, damper configuration and field experience.

The table should indicate limits for suction and discharge volume bottles depending upon the compressor type and volume bottle configuration.

In order to avoid modifications identified in final acoustic and mechanical studies, a design of the main dimensions, supporting damper saddles and other sensitive elements is performed.

Once the size of the dampers is defined, through the introduction of the main parameters (e.g., diameter, length, nozzle thickness, number of supports, etc.), it is possible to estimate the 1st mechanical natural frequency of the manifold system (evaluated using a regression law based on a historical database and adjusted for the specific case) and its potential margin of error.

The above result is then compared with the highest significant exciting harmonic of the compressor (generally the 2nd) applying a safety margin of 20%.

This preliminary design allows the evaluation of alternative compressor layouts to validate the size, shape, and supports of pulsation dampers and other parameters prior to the complete definition of the compressor layout in order to be as close as possible to the final design. After that it is also possible to issue the first manufacturing drawings for the dampers with limited risk of later changes in the primary dimensions.

This very accurate calculation is called the “cylinder manifold analysis”.

The first step concerns the building of the mechanical model and is very important for the accuracy and confidence of the system results.

It is necessary to carry out an accurate structural evaluation of the system giving special attention to components that affect the dynamic behaviour and at the same time applying suitable simplifications to the model to achieve an appropriate balance between accuracy and rapid calculations. The performance of modern computers gives reasonable execution time even with the use of complex, accurate models.

Another important aspect that must be taken into account during the construction of the mechanical model is the integration (i.e., model interchange) between 3D CAD and CAE tools.

In the past, a serial engineering design process (designing, FE model, structural analysis) was conducted separately without close cooperation between the other disciplines that contributed to the whole design.

Today, new advanced modelling tools support an integrated approach called “Concurrent engineering design” that yields greater precision, and reduces time and cost.

A short description of FEM model types that have been developed is reported below.

Shell elements model

For many years, Shell FE compressor models represented the best approach in terms of accuracy of results, modelling effort and calculation time. This method does a good job of simulating components with small thickness compared to other geometric parameters (e.g., parts made from sheet metal) such as damper models, which are the most important components from a dynamic point of view.

The 4node shell element has both bending and membrane capabilities. Both in-plane and normal loads are permitted. The element has six degrees of freedom at each node: translations in the nodal x, y, and z directions and rotations about the nodal x, y, and z-axes.

The thickness is assumed to vary smoothly over the area of the element, with the thickness input at the four nodes. The reciprocating compressor cylinder is a very stiff part compared to the rest of the cylinder manifold system.

However the use of shell elements to model components such as cylinders, crossheads, and crankcases which have a structural behaviour more similar to that of a solid body introduces a significant approximation.

Considering that in the mechanical chain of the cylinder manifold system these components are much more rigid than distance pieces and volume bottles, it is advisable to model them as simplified geometrical shapes having “rigid body” behaviour.

Since shell elements do not provide the exact physical morphology of the component, it is important to verify the
agreement between the mass resulting from the model and the actual value indicated in the manufacturing drawing.

If these two values are different, an appropriate density should be assigned to the component in order to obtain the same value.

Shell models with their simplified geometric shapes (Fig. 1) provide a more intuitive means of understanding the general system behaviour and identifying the components that mainly affect the dynamics of the system.

Fig. 1 Shell Compressor and piping model

**Solid elements model**

This approach better simulates the actual geometric shape and physical morphology of components. The main difficulties encountered are:

- Accurate solid FE models need many nodes and consequently the number of equations to be solved is very large.
- Building a solid model using an FE pre-processor tool without a 3D CAD input file (i.e., from a 2D manufacturing drawing) requires a great effort.

The performance of modern computers and the continuous improvement of FE tools and 3D CAD software, coupled with the simplification and availability of these programs, have significantly reduced the time necessary to prepare the complete model.

To speed the process, the manufacturing drawings of each item should be built with 3D CAD. All the individual parts are then assembled in a global 3D General Arrangement inclusive of its compressor skid and connecting piping.

Today’s 3D CAD models for the design process can be used for FE modelling without spending a lot of time in adjusting them.

However CAD models used for manufacturing drawings still contain a lot of details and equipment which are not significant from a structural point of view so that collaboration between the designer and the structural analyst is necessary to insure that the model is properly simplified for the FE modelling.

This CAD translation step can use specific tools which provide the correct interface between CAD exported files (e.g., *.iges, *.step) and finite element pre-processor programs.

![3D CAD compressor model](image)

**Fig. 2 3D CAD compressor model**

Mechanical models used for standard compressor vibration analysis include cylinders, pulsation dampers, crosshead guides, distance pieces and supports, while pistons, rods, bearings, bolts, valves, etc. are taken into account as masses.

In the model, compressor foundations with constrained points can be simulated as follow:

- Concrete foundation: considered as a rigid body; all the connecting points between the compressor and foundation are constrained;
- Skid on concrete foundation: skid included in FE model; connecting points between skid and foundation constrained;
- Skid on a non-rigid structure (platform, shipment, permafrost, etc.): Flexible structure is introduced in the model with its characteristic stiffness matrix (Sub structuring techniques).

**Frequency analysis**

This is performed to calculate the mechanical natural frequencies and the mode shapes of the cylinder manifold system and is concluded when the minimum mechanical natural frequency of the system is at least 20-50% higher than the highest significant harmonic frequency of the exciting shaking forces, determined during the acoustical study and cylinder gas load calculation. If the exciting forces and natural frequencies are too close, a forced response study is mandatory to calculate the vibration amplitudes and relevant cyclic stresses.

This approach requires experience in shaking forces and gas load evaluation. It is appropriate to perform only a frequency analysis if the exciting forces have low amplitude.
(absolute value) and the natural mechanical frequencies are far from significant harmonic components. For instance, relatively low shaking force amplitude having a high frequency component, which may appear negligible during a preliminary evaluation, can in reality be very dangerous. In fact when its frequency is an exact match with a mode shape in which the deformation and the exciting forces are in the same direction, mechanical resonance will occur. Therefore, special care must be taken in the above evaluation and in case of doubt, a force response analysis is strongly recommended.

**DAMPER DESIGN VALIDATION EFFICIENCY**

Once the damper and general layout drawings are available, an analysis of the damper is performed. The damper simulation includes the complete geometry, and the endless line is replaced by a hypothetical plant including resonating lengths to evaluate the consequent shaking forces and assure appropriate damping.

These approximate calculations drastically reduce the impact of problems detected during the final acoustic study and the simple insertion of orifices can insure maintaining the design pulsation limits avoiding further damper analysis that could impact the damper fabrication schedule. In addition this approach allows meeting the contract delivery schedule without precise data on the client’s piping at a very early stage in the project when they are not available or are very preliminary.

It is also possible to determine with sufficient accuracy the amplitude and spectrum of damper shaking forces for the forced cylinder manifold study.

Gas loads acting on cylinder ends are obtained from compressor design calculations.

**CYLINDER MANIFOLD FORCED RESPONSE**

**Procedure**

The forced response of the cylinder manifold system is performed using as input data the harmonic components of the shaking forces defined by the pulsation study and cylinder gas loads. It allows the calculation of vibration amplitudes and stresses for each mode, along with the dynamic reaction forces on the cylinder and on the discharge bottle supports.

The forced harmonic response is calculated for each possible resonance condition caused by mechanical natural frequencies and exciting force harmonic components.

The Campbell diagram (Fig.3) correlates the mechanical natural frequencies and the exciting harmonic frequencies based on the compressor rotating speed (990 RPM) in order to highlight where resonance may occur.

The analysis is carried out for each harmonic, with +10% around the nominal frequency (RPM).
\[ \gamma_n \] is the generalized modal force  
\[ \gamma_n = \sum_j F_j D_{jn} / \sum_j m_j D_{jn}^2 \]

As stated by Standards, piping and compressor systems typically have damping ratios from 1% - 5%, and hence amplification factors of 10 to 50.

According to our experience and field measurements, a damping ratio of 0.02 associated with the most significant natural frequency is a good approximation.

**Exciting Forces**

The frequencies and the amplitudes of the pulsation induced shaking forces are obtained from the acoustical simulation and applied to the compressor FE model.

Exciting forces are generated by
- Pressure pulsation applied on suction and discharge volume bottles
- Cylinder pressure applied on the cylinder heads and balanced by reaction applied on the compressor frame.

Fig. 4 shows the model of the compressor complete with the crankcase and the applied loads shown in red.

**Analysis of results**

Typical results of forced analysis are as follow:
- Cyclic stress to be compared with the allowable value, (API cyclic stress limit 179 N/mm² (26000 psi) reduced to consider the stress concentration factors and an adequate safety factor).
- Vibration amplitude to be compared with allowable manufacturer limits based on experience and field measurements (Fig. 13).
- Other available results may be reactions due to vibrations transmitted by the compressor system to the foundation that are important to consider in the case of a skid mounted on a non-rigid structure such as a fixed or floating platform (Fig. 5).
FINAL ACOUSTIC SIMULATION

When the entire plant system design under consideration has been completed and final detailed data are available, the final acoustic and mechanical analysis is made. Generally when the described procedure has been fully applied, these final analyses only identify minor adjustments (e.g., orifice plates).

FIELD VERIFICATION

Field measurements on the compressor are carried out for the following reasons:
- Final tuning of FE model settings for new types of compressors or if it is necessary to introduce significant modifications to an existing standard model;
- Specifically required by the contract
- Correcting a problem identified during operation.

Measurements of a low speed compressor

Other exciting forces that can cause vibration problems in a compressor are the cylinder gas loads (Fig. 7).

Usually for gas loads the first harmonic component is the most significant and the others are quite low in comparison and in most cases not investigated.

Field measurements indicated a mechanical resonance of the bottle that had an exciting effect on the compressor system at the 3rd harmonic (i.e., 16.35 Hz).

As the acoustic shaking forces were quite low, the only 3rd harmonic component that was not negligible, even though low, was from gas load (Fig. 7).

A forced harmonic response analysis including the gas load components greater than the 2nd harmonic (e.g., for harmonic components of 3rd, 5th, and 7th order), showed that (Fig. 8 and Fig. 9) for the 3rd and 7th harmonic excitation responses, vibration amplitude and stresses were greater than allowable.

![Fig. 8 Comparison of vibration amplitude results and allowable limits](image8)

![Fig. 9 Cyclic stresses](image9)

Based on the above results it was suggested that the suction bottle support (Fig. 10) be modified to increase its stiffness and drastically reduce the vibration amplitude and associated stress to within allowable limits (Fig. 11). The modifications consist of a new longitudinal beam, and reinforcing plates and braces.
Compressor vibration measurements

The capacity control steps during the test were 25-50-75-100%, and were recorded with the significant operating conditions (pressures, temperatures, etc.).

Vibrations were measured at the cylinders, body slides, and suction and discharge volume bottle and were made along the three principal axes. The phase is referenced to an accelerometer positioned on the cylinder head in the X direction.

Vibration was measured in velocity (mm/s 0-Peak).

The acquisition frequency range was set at 5 – 300 Hz.

Before and during the measurements the chain was checked with a calibrator with a setting of 10 µm RMS @ 159.2 Hz.

The comparison between the forced response analysis (Fig. 12) of the compressor and the field measurements is reported.

The results of field measurements are given in terms of natural frequency of the compressor, vibration measurements [0-peak] in points indicated for each load condition (Fig. 14) and the spectrum analysis of vibration relative to each point (Fig. 15).
The dynamic behaviour of the FE system simulation was similar to that measured with the RPM 10% lower than the nominal value. Therefore this FE analysis calculation point was considered as the match with the field measurements.

OPTIMIZED PROCEDURE

Short delivery time for compressor packages requires specialized tools for each step of the design procedure. Once the compressor frame and cylinder arrangement are selected it is necessary to design the damper volume size, shape and supports.

To minimize changes or modifications due to vibration and acoustic problems in the original design of the manifold components, the following procedure is recommended (Fig.16):

- Initial volume bottle selection and damper validation (size and type);
- Preliminary study of compressor layout based on a sensitivity analysis, using statistical laws based on a database of case histories (shape and geometry of the damper must meet specific rules);
- Definition of final plant lay-out and damper manufacturing drawings;
- FE model of the cylinder manifold system and frequency analysis using shell/solid elements that yield a model of sufficient accuracy and reasonable calculation time;
- Damper verification with acoustic simulation and determination of exciting shaking forces;
- Determination of cylinder gas loads;
- Forced mechanical response of the cylinder manifold system performed applying acoustic shaking forces and cylinder gas loads

Fig. 15 Spectral analysis in Y direction for point 1

Fig. 16 Optimized design procedure flowchart

The flowchart summarizes the procedure for compressor design optimization starting from preliminary studies of damping systems through the detailed acoustic and mechanical analysis so as to fully simulate the system and apply the relevant exciting forces.

The procedure satisfies the requirements of API Approach 3 [1] and allows contract delivery requirements to be met taking into account the following:

- Allowable Limits (pressure pulsation, shaking forces, cyclic stresses, etc.)
- Plant data for the final acoustic analysis not available in the early stage of the project
• Difficulty in meeting the compressor delivery schedule without using preliminary plant data;
• Practical experience indicates that after pulsation devices are fabricated, the only design optimizations available are the installation of orifices, piping modifications and stiffening of the piping system.

CONCLUSIONS

The design of the compressor manifold system is fundamental for the safety, reliability and operation of reciprocating compressors.

The pulsation dampers are both part of the machine itself and of the plant piping system since they are the most effective elements to prevent piping vibrations. Specific analyses have to be performed to properly design these components from the beginning of the project in order to satisfy API Approach 3 requirements and to avoid major modifications that could compromise the contract delivery date.

Taking into account that, in most cases, the plant data needed for damper fabrication validation is not available on a time frame that matches the compressor design delivery schedule, it is absolutely necessary to follow an optimised procedure to minimize potential risks. This procedure must be based on practical experience taking into account the design and manufacturing aspects of the project.

In this connection it is fundamental that all the disciplines involved in the process work cooperatively to meet the contract delivery schedule.

When the sizing procedure that has been described is followed, the final acoustic and mechanical study will only require minor modifications that can be applied without project delivery impact.

The CAD programs now available allow the FE model to be built directly from the manufacturing drawings, thus avoiding possible translation errors.

Finally models can be improved by validation using field measurements.

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