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ADVANCED TECHNOLOGIES FOR RECIPROCATING COMPRESSORS WITH RESPECT TO PERFORMANCE AND RELIABILITY

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Abstract:

Reciprocating compressors play a key role in the today industry. They are vital machines in various units of industrial plants from air compression units to refineries, gas processing plants, petrochemical facilities, refrigeration systems, LNG units and so on. They are the most efficient and the most common type of compressor. Advantages of new technologies compared to old-fashion designs for various components and sub-systems of these machines are described. The focus of article is on modern developments and latest achievements for reciprocating compressors to enhance the performance and reliability, while keeping cost within a certain acceptable limit. Case studies are also presented.

Keywords: Reciprocating Compressor, Performance, Reliability.

1. Introduction

The key principles that have guided and developed the modern machine design have always been:
1- Fitness for purpose, cost and economics. Currently minimum life cost is very important that leads to selection of high efficiency machine options (such as reciprocating compressors, as the most efficient compressor type).
2- High reliability, high availability, safety, health and environmental protection.
Above mentioned principles lead many compression units to use reciprocating compressors. Latest technologies in reciprocating compressor design, component selection, performance, monitoring and machine management are discussed. Reciprocating compressors (known also as “piston-type” compressors) are the most common type of compressors [1-4]. They can generate high head independent of the gas density. They are currently the only available compressor for above 1000 bar pressures. Successful piston-type compressors are presented for above 4000 barg. They are the best machines for any service that the capacity is relatively low and pressure is relatively high (such as many natural gas or refrigeration applications).

The best configuration is the horizontal cylinder(s) with discharge nozzle on the bottom side. Vertical and inclined machines are still in use, but there are not popular and may only be accepted for non-critical applications and small sizes (usually below 350 kW). Non-horizontal machines are employed to save space (less footprint) or for better commercial conditions. Figure 1 shows an example of a medium size reciprocating compressor for a process application.

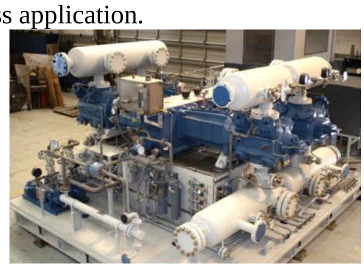

Figure 1 An example of a medium size horizontal-type reciprocating compressor for a process application. Suction pulsation bottles are installed above the cylinders. Discharge pulsation bottles are installed under the cylinders.

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2. Performance And Process
2.1. Performance and Integration

The most important performance aspect is the compressor integration with the process. Proper integration of the reciprocating compressor (including performance, control and capacity control) and the process facilities can only be confirmed after extensive studies of both systems. Review of all compressor performance curves, matching machine curves with process plant requirements, studies of mutual interactions and simulations of all operating situations play crucial role in this regard.

A modern recommendation is to avoid a steep load curve. A review of the steepness of the load curves can quickly identify which load steps (and where) are quite steep in nature, and thus small changes in the pressure can have significant changes in the load and the flow (the capacity). Steep load curves may indicate improper sizing of the cylinders. Compressors with steep load step curves can also offer difficulties in the control and the automation [1].

2.2. Rod Load and Pin Reversal Details

The maximum rod load should be less than 80% of the allowable rod load. Duration and peak magnitude of rod load reversal should not be less than 20º of crank angle and 5% of the actual combined load (in the opposite direction), respectively, to guarantee sufficient lubrication of the piston rod-crosshead pin for long-term trouble-free operation of this critical mechanism. Above mentioned limits are traditional requirements. They should be considered as hard limits, applicable for very special operating situations. With respect to modern simulation methods and latest mechanism optimization procedures, a modern expectation is that for commonly used operating cases, much better values are targeted. All possible operating cases (such as very low suction pressures and all part-load steps) should be carefully studied, and even a specific operating situation could be banned if the load reversal is not sufficient for long-term reliability. In modern reciprocating compressors, the design is optimized in a way that the rod load reversal duration and magnitude exceed above mentioned limits. For example, rod reversal duration above 70º of crank angle and rod reversal load more than 25% of the combined load (in the opposite direction) are commonly achieved. Figure 2 shows an example of a large size piston and piston rod for a large reciprocating compressor.

Figure 2 An example of a large size piston and piston rod for a large reciprocating compressor.

2.3. Discharge Temperature

High discharge temperatures can cause problems with any sealing element (any non-metallic component in contact with compressed gas), lubrication cooking, and cylinder valve deterioration. Discharge temperatures should be reviewed for all possible operation situations at least for the average and the maximum suction temperatures. The maximum predicted discharge temperature **[1-4]** should not exceed 150ºC and not exceed 135ºC for hydrogen rich services (molecular weight of 12 or less). Latest trend is toward limiting gas discharge temperatures below 120ºC to extend life of wearing parts.

3. Compressor Design
3.1. Modern Inter-stage Pressure Arrangement

Discharge pressure of each compressor stage is normally protected by pressure relief valves (PRVs). High pressure discharge switches are often used, but they cannot replace PRVs **[2]**. Inter-stage pressures are going to increase during part-load operation and high suction pressure (suction pressure more than normal suction pressure). If not tolerate-able, additional clearance pocket on first stage cylinder and part-load operation inhabitation by the controlling logic can be studied. Generally, it is the best solution to accept the inter-stage pressure increases and fix the inter-stage facility design pressures (and PRV set points) based on the increased values at part-load and high suction pressure. These inter-stage pressures are usually around 10-20% higher than normal values. In other words, for commonly used applications, 15-20% safety margin on inter-stage pressures at the basic design stage is usually more than enough for unpredicted issues like this.

3.2. Low Suction Pressure

Sometimes, because of process requirements, reciprocating compressors should be capable to operate at a relatively low suction pressure and a full design flow (full rated capacity) for a normal discharge pressure. For example, suction pressure could drop considerably because of an extreme pressure drop in a gas supply system. Some processes (such as some gas processing modules or refinery units) are usually supplied gas to a reciprocating compressor at very low pressures, sometimes far below predicated values. It is common to receive gas at an unpredicted low suction, even lower than theoretical minimum pressures. This consideration can have strong effect on the compressor sizing especially the machine frame rating and the driver power. In a case study, based on experiences of a special refinery unit, the reciprocating compressor is sized for 20% suction pressure below the normal suction pressure that results in an around 35% increase in the compressor frame rated power and driver power. Latest experiences strongly recommend respecting this issue in the compressor biding stage to avoid costly future changes.

3.3. Cylinder Valve Selection

Cylinder valves are the most critical components of reciprocating compressors and strongly influence the reliability, operation, performance and efficiency of these machines **[5-8]**. Cylinder valve defects are obviously responsible for most of the unscheduled maintenance events **[9-13]**. Valve components should be operated (opened and closed) several billions of times during their operating life without being affected by fatigue or other degradation mechanisms.

Correct material selection and proper component design are keys to achieve a successful valve operation. Reciprocating compressor valves should be supplied from a reputable valve manufacturer with proper references and long-term successful production history. Some compressor manufacturers are also active in cylinder valve business. Usually, the cylinder valves manufactured by compressor vendors should be dealt with a great care. Cylinder valve design and manufacturing should be considered as very delicate tasks. Only a few professional valve manufacturers can supply high quality cylinder valves. Sometimes, for special applications, there are only three qualified cylinder valve suppliers with satisfactory references.

Advanced polymers have excellent mechanical properties and are capable of working at hostile cylinder valve conditions that also include extreme mechanical stresses and relatively high temperatures. In most cases, modern, sophisticated, low mass polymer sealing elements can vastly increase valve life as well as reduce energy consumption and maintenance costs. The main advantages of modern polymers over old-fashion metallic valve parts are: improved heat resistance, high fatigue life, high tolerance to dirt and corrosive traces (liquids or chemicals) particularly those in dirty gases, reduced wear and improved sealing capabilities.

Main valve types are:

1- Ring type valve.
2- Plate type valve.
3- Poppet valve.

For large machines (generally low speeds and high pressure ratios) and small machines (relatively high speeds) ring type valve and plate type valve are the best choices, respectively. The best valve size should be obtained with respect to efficiency, reliability and performance requirements considering many operational and machine design factors such as the minimum clearance volume. Lift is the distance travelled by the valve moving elements. The higher the lift, the higher the valve flow area, the lower the valve pressure drop, the less consumed power, the higher moving element impact velocities and the lower valve durability. An optimum valve size and an acceptable valve lift should be found for each application. Optimum valve spring stiffness is also important. Springs that are too stiff, can lead to valve flutter (more compressor power and considerable wear rate) or early closing of valve (reduce capacity). Springs which are too light, can cause valve late closing and the reverse flow (higher velocity, less reliability and reduced capacity). Nonlinear partial differential equations describing the valve differential pressure and the valve element motion (such as [14]) can be used in an optimization process to estimate optimum valve lift, spring stiffness and gas velocity for each machine and application.

Traditionally, poppet valves were popular many years ago. Previous generations of poppet valves have left service because of poor performance, low reliability and operational problems. High valve lifts were a famous characteristic of the previous generations of poppet valve. Today some compressor manufacturers offer new versions of specially-designed poppet valves with great promises. Lifts of these new valves are within modern ranges. Performance and long-term references of these valves should be considered carefully. Sophisticated evaluations are required to highlighting clearly advantages and disadvantages compared to commonly used valves (such as ring type valves) for each application. Figure 3 shows an example of a ring type cylinder valve for a reciprocating compressor.

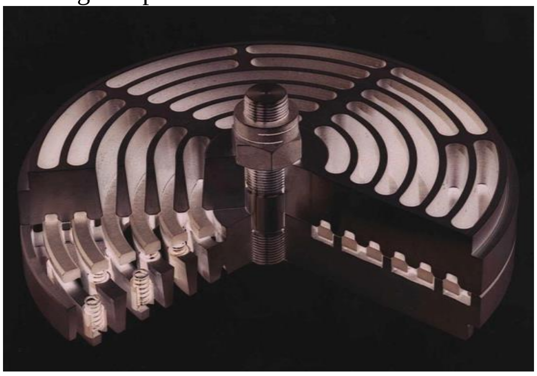

Figure 3 An example of a cylinder valve for a reciprocating compressor. This is a ring type valve, suitable for medium and large reciprocating compressors.

3.4. Latest Capacity Control Systems

One of the new technologies presented for reciprocating compressors is the step-less capacity control system. A step-less capacity control system uses finger type unloader and unloads the suction valve for only a portion of each compression cycle to achieve an adjusted capacity [2]. It is a hydraulically actuated system with a very complex mechanism, and a very sophisticated control which is offered by very limited manufacturers. The principle works by reversing part of the total gas that has been taken into the cylinder by conveying it back to the suction chamber by holding the suction valve open for a controlled and variable proportion of the compression stroke. Therefore the discharge capacity and power reduction are virtually proportional. In this way, the capacity of the compressor can be regulated steplessly, and the energy consumption is reduced proportionally with respect to the reduction in the capacity. The selected step-less capacity control device should only be used with a valve from the same manufacturer.

Finger type unloaders (whether in a step-less capacity control system or as a standalone finger type unloader) have potential for damaging the valve sealing elements and require more care for maintenance [2]. Valves and unloaders cause around 44% of unscheduled reciprocating shutdown [5,6] and this selection has an important effect on machine reliability and maintenance [5-8]. In addition, unloader selection has a strong effect on the performance, operational flexibility and the start-up and shut-down of the machine. Plug type or port type unloaders can offer better reliability and performance compared to finger type unloaders. However, they are not available for small sizes.

Great care should be exercised for a small machine unloader selection. Finger type unloaders are only available option for some small reciprocating compressors. For some tiny machines (say below 100 kW), even a 100% spillback (recycle loop) could be an acceptable capacity control solution, because the wasted power is low. However, operational flexibility often is considered more important than a 10-20% added reliability and finger type unloaders are provided (usually in addition of 100% spillback loop) for small machines in critical applications such as refinery, petrochemical or gas processing plants.

For medium size machines (from 300kW to 2 MW), the best capacity control configuration is the selection of part-load steps based on plug/port unloader, and if necessary the clearance pocket. Clearance pockets should also be dealt with care since they could offer some reliability and operational issues.

New technologies (such as step-less capacity control devices) are not suitable for all applications. Step-less capacity control devices are only recommended for large machines (say above 2 MW) with great durations of part-load operation. Figure 4 shows an example of a step-less capacity control for a large reciprocating compressor.

Figure 4 An example of a step-less capacity control for a reciprocating compressor. This complex system is used on a ring type valve in this example. It is recommended for large reciprocating compressors with extended operating times at part-loads.

Step-less system is a fast-acting, accurately controlled arrangement for the energy-saving operation and rapid control of reciprocating compressors. Step-less control system allows the operator to compress only the required amount of gas in a very dynamic fashion. However, this system uses special instruments and actuators and nearly always brings a long list of deviations (to compressor manufacturer and project specifications) and special requirements in design, installation and operation. Of course proper manufacturer guarantees can be offered by manufacturer and satisfactory assistances in all levels could be received. Overall, it is a modern and special-purpose system that should only be employed when really necessary. Only for large machines with long durations of part-loads and requirements for fast follow-up, this special system is recommended.

Theoretically, when using step-less capacity control system, bypass loop could be eliminated. However, it is just theory and the best recommendation is to provide a 100% recycle loop (bypass loop) for operational flexibility and continued operation if the automatic step-less capacity control system shows a problem. Modern step-less system is expected in critical applications and a 100% recycle system could be justified for such a critical service. Modern step-less capacity control systems are relatively reliable systems and their record of reliability is not worse than conventional unloader systems. On the other hand, these complex systems include various mechanical, hydraulic, electrical and control sub-systems and their reliability could be higher than a certain limit. An important issue could be the system cannot hold its position in case of a problem. In other words, the system should be set to 100% load or 100% unload the machine in case of a failure or problem (such as hardware/software issue, actuator problem, hydraulic system issue, instrument failure, or another operational problem). A full load option plus a 100% bypass (recycle loop) can offer a good operational flexibility up to the shutdown of the machine in the first possible opportunity.

3.5. Advanced Condition Monitoring

Condition monitoring systems [9-15] should be particularly cost effective, at the same time should include all necessary items to identify malfunctions at an early stage. End result of an optimum condition monitoring system should be a relatively low maintenance cost and the lowest risk. An advanced vibration monitoring system includes:

- Continuous vibration monitoring of the compressor and the driver. Velocity transducers are preferred over accelerometers because of a better signal to noise ratio **[9]**. For large machine (>1MW), both should be employed. An advanced configuration is vibration monitoring at each end of the crankcase about halfway up from the base-plate in line with main bearings.
- Accelerometer at each cross-head.

An advanced temperature monitoring should include:

1- Gas discharge temperature for each cylinder.
2- Pressure packing case - piston rod temperature.
3- Crosshead pin temperature.
4- Driver strategic temperatures, particularly driver bearing temperature.
5- Valve temperature.
6- Oil temperature out of compressor frame.
7- High jacket water temperature for each cylinder.

Proximity probes should be located under the piston rods and used to measure the rod position (rod run-out) and determine malfunctions such as wear of the piston, rider band problems, cracked piston rod (or crack in any piston rod attachment), a broken crosshead shoe, or even a liquid carryover to a cylinder. Latest recommendation is to use rod run-out measurements just for monitoring and alarm (definitely not for trip). Recommend limits for cold run-out and normal operation (hot) run-out are 60 microns and 170 microns (peak to peak), respectively.

All shutdown functions should be 2 out of 3 voting to avoid unnecessary trip. Generally minimum numbers of shutdowns should be assigned for a reciprocating compressor in a critical service such as a hydrogen unit in a refinery, a gas processing crucial role, important refrigeration module and so on.

Low pressure trip of the lubrication oil system is considered an essential trip case. Operators always encourage very high vibration level for a shutdown (even sometimes 8-10 times the normal vibration level). There are always discussions about high discharge temperature shutdowns. Many experienced operators argue that they prefer to tolerate relatively high discharge temperature (and a high temperature at the cylinder valves, which could result in valve and all wearing part life reduction), compared to an unscheduled trip of a machine that can result in a critical refinery unit shutdown with production losses of an around 0.5 million dollar per day. Of course safety risks should always be assessed in these situations. Author's recommendation is to consider high discharge temperature trip (since it is a mandatory API618 requirement and constantly insisted by safety teams). However, trip level should be set properly high (based on accurate simulations and realistic thermal/safety evaluations) to avoid unnecessary shutdown.

3.6. Rod Packing and Piston Rod Coating

Piston rod seal is the second important area for the reliability of a reciprocating compressor and most likely path for potentially hazardous process gas leakage [8]. Packing life could be improved 2-3 times by adding an advanced coating, particularly a proper tungsten carbide piston rod coating [10]. Finding a suitable packing for a process reciprocating compressor is usually a difficult task. There are a few reputable manufacturers that can offer good products. Some compressor manufacturers are also actively participated in this market with relatively good reliability records. Of course clever compressor manufacturers left this business (along with cylinder valves) to dedicated manufacturers. Reference check is a key. As an indication, based on the author experiences, it is difficult to get a rod packing offer with a guarantee more than one year.

3.7. Optimum Speed

Traditionally, a reliable reciprocating compressor is involved low speed (around 330 rpm) and lubricated-cylinders. Since 20 years ago, for small and medium reciprocating compressors, many vendors have intended to deviate lubricated-cylinder technology and low-speed requirements. The speed limit is still a critical issue. High speed ranges could result in relatively small machines with commercial, weight and space benefits. However, machine reliability decreases particular for contacting and sliding surfaces. Wears and frictions as result of a high speed operation could be detrimental. The speed should be selected based on a proper optimization. The optimum piston speed is usually obtained in 3 - 4.5 m/s range, depending on each application. For large and medium machines, the speed should be around 300-350 rpm. For small machine (say below 250 kW) speed up to 600 rpm may be used. Speed limits for oil-free compressors are usually lower than lubricated ones.

3.8. Oil-Free Compressors

Ten years ago, probably, less than 25% of all reciprocating compressors were designed for non-lubricated operation. Main driver for oil-free technology has been the process demand. Oil-free requirement is important for special applications such as oxygen, high pressure air, when oil could be problematic at downstream (for example, poisoning the reactor catalyst), and similar applications **[2]**. Nowadays oil-free reciprocating compressor manufacturing is considered an advanced and successful business. Currently oil-free compressors are used in a vast range of processes to offer high performance operation and high quality products (free of oil). The oil-free technology uses special materials (such as special polymer materials) for piston rings, rod packing and other sealing systems in contact with compressed gas to avoid any oil carryover by the gas. Lubrication oil system is still required for bearings and crankshaft mechanism. However, lubrication oil is isolated from the compressed gas using proper dry sealing systems. Theoretically, the cost of oil-free machines may be slightly higher compared to oil-lubricated machines. Every year, this small cost difference is further decreased. Expected life of dry seal components may be lower, since dry oil-free components could be weaker (compared to lubricated sliding components) against the friction and the wear at sliding and contacting surfaces. However, overall reliability offered for an entire process unit by an oil-free operation could be more than an oil-lubricated machine for commonly-used services, if suitable oil-free components (particular sealing materials) are employed and proper references are available. Oil-free technology is an important innovation of the modern compressor industry.

Recently the author has received several offers from two different reciprocating compressor manufacturers to change purchased lubricated cylinder machines (at early stage of deisgn) to non-lubricated machines, without any commercial increase (the same cost and the same delivery time). In all cases, the process team and plant licensors have encouraged oil-free machines. However, operation team and maintenance representatives have raised some concerns about possibly shorter life of non-lubricated wearing components. Except one case for a relatively high pressure machine that comparable successful references could not be offered by vendor, all other cases have been transformed to oil-free compressors.

There are limited suppliers for other oil-free positive displacement compression technologies for small and medium size compressors. For example, there are only four reputable manufacturers worldwide for oil-free dry-type screw compressors for medium capacity ranges at some process applications, which can offer successful references. Centrifugal compressors (even small units or integrally-gear compact machines) cannot be used below a certain level of the power and the capacity. Considering all above mentioned facts, the oil-free reciprocating compressor is an extremely strategic piece of equipment. Today oil-free (non-lubricated) piston-type compressors constitute around 50% of reciprocating compressor business for small and medium size applications in commonly used pressures and services. There are some limits in oil-free technologies (such as pressure, speed, successful references, some size limits, and others). For large, high-speed or high-pressure machines, lubricated cylinder technology could still be the first choice. An optimum option for large (>1.5 MW) or high pressure (say above 50 barg), if there is no process requirement for zero oil carryover, is still the lubricated cylinder with low speed machines (around 300-350 rpm).

3.9. Advanced Passive Vibration Control

Usually, the preferred design of reciprocating compressor for small and medium sizes is a two-cylinder machine. For large machine, four-cylinder and six-cylinders are commonly used. Sometimes, odd number of cylinders is unavoidable. In these cases, dummy crosshead should be used to reduce the operating vibration. State of art spring-mass-spring systems can be studied for passive vibration control (more reduction in the vibration). It is a new technology. In this innovative system, the dummy crosshead is on the one hand attached to a movable piston assembly by a flexible member and on the other hand to the stationary compressor casing using auxiliary mechanical springs. Masses, dimensions and stiffness are optimally calculated to offer minimum operating vibration.

4. Compressor Packaging

4.1. Modern Suction Drum and Latest Coolant System

Liquids should never form inside the cylinder [10-12]. Liquid contributes to poor reliability, can cause high impact velocities, can lead to stressing of cylinder valve moving elements (slugging) and reduce the sealing effectiveness. For any application, a sufficiently sized suction drum with a drain provision should be provided [12]. Sometimes this suction drum is provided as a part of pulsation control system. This arrangement is not recommended. Generally, a separate vertical-type suction drum (separate from the horizontal-type suction pulsation vessel), near the compressor package is the best option.

Cylinder cooling system operation and temperatures should be carefully monitored and controlled. As an indication, coolant inlet temperature should be between 7ºC and 15ºC above the inlet gas temperature. For exotic gases or operations near critical areas, more care should be taken for the coolant temperature control. Thermodynamic model should be carefully reviewed to make sure the liquid cannot be formed and the cooling system performance is properly set for the cylinder(s) and other critical points (such as rod packing). The first purpose of the cooling system is avoiding the hot spots within the cylinder and the rod packing. Cooling system function is even more critical in modern oil-free compressors. Vulnerable dry sealing elements rely on cooling system for proper operation and their battle against frictional heat, sliding wears and operational hot spots. Ample margins for cooling water flows are always encouraged. However, a coolant relatively low temperature (for example, lower than the inlet gas temperature plus a 5ºC margin) could be problematic. Coolant temperature should be closely controlled within the minimum and maximum limits.

4.2. Modern Pulsation Control

Modern pulsation control techniques trend to dissipate less energy than reliance on special solutions such as orifices to damp pulsation levels [16]. Acoustic reviews should be performed for the rated and the guarantee points as well as all other expected operating cases and combinations of pressures, speeds and load steps. Pulsation can also alter the timing of the cylinder valve motion and decrease efficiency and reliability of the machine [5]. Based on the latest optimization processes, at the compressor bidding stage, the pulsation limit is recommended around 95%-85% of API 618 (Approach 3) limits to have some margin (around 5-15%) to mitigate risk during detail design of the compressor package (particularly pulsation bottles), the construction and the installation periods as well as unpredicted deviations and problems.

Vertical-type suction pulsation vessels have been used in large reciprocating compressors for years. They have offered many pulsation, vibration and operational problems. Specifically, the relatively long (say more than 1 m, sometimes even 2 m or more) connecting line between the compressor and the vertical pulsation vessel (which should stand near machine, but there is always some limits) could be a source of severe pulsation and a good candidate for detrimental resonances. Considering this fact, the only acceptable pulsation vessel arrangement is horizontal vessels for both the suction and the discharge. Even mammoth type large installations for large machine (large horizontal suction pulsation vessels with massive supports on top of the compressor cylinder) is much better than deadly pulsation resonances offered as result of vertical suction vessel with relatively long connecting pipe spool. Some vendors have been insisted on vertical-vessel solution with orifices for pulsation resonance control. This solution may be attractive commercially (in terms of initial price) since it eliminates large horizontal vessels with required expensive, strong supports. However, it can offer many operational and reliability issues, and it results in extremely high combined cost (initial plus operation costs). In general, relying on damping for proper operation is a dangerous and risky strategy. Smart design is avoiding resonance, rather creating resonance and use damping to suppress it. In case of any minor deviation from the theoretical basis (or any future change), the orifices inside the vessels (the best place for damping the pulsation harmonics is at the nozzle end inside the vessel) should be replaced with new ones, which can result in cutting and re-welding of thick, critical pulsation vessels at the site. To avoid all above mentioned risks, the horizontal vessels, strategically installed near the cylinders are offered as the best solution.

Combined pulsation vessel (one long vessel connected to two or three cylinders) has been an old-fashion design, proposed by some medium-size compressor manufacturers (usually by manufacturers with relatively small pulsation control team). Historically, this classic solution was popular many years ago, before modern methods and rules of pulsation control. Latest theoretical works [16-18] and new comprehensive simulations have proved their ineffectiveness and their disadvantages. The best recommendation is a suction horizontal pulsation vessel and a discharge horizontal pulsation vessel for each cylinder.

The distance between the horizontal pulsation vessel nozzle and the cylinder nozzle should be selected properly. The optimum distance should guarantee trouble free access to valves, unloaders and other cylinder accessories without removal of the pulsation vessels. At the same time, the distance should not be so long to cause any resonance with generated pulsation (and their harmonics) since unsuppressed pulsation harmonics existed at immediate discharge of the machine (between the machine and the pulsation vessel) could be detrimental. As an indication, for medium size machines, a distance around 0.4-0.7 m could be a good selection considering access, valve maintenance and pulsation.

4.3. Pulsation Shaking Forces

The pulsation vessel volumes and dimensions (and connecting piping) should be designed and optimized with great care. The pulsation suppression and shaking force reduction should be done as a combined action. Sometimes, the reduction of pressure pulsation (if not done properly, particularly if performed as a single task, without checking the shaking forces) can be accompanied by an increase in the shaking forces (or the unbalanced forces) [17]. Shaking forces should be determined and controlled properly. Piping and vessels should be properly supported. The margin of separation between the mechanical natural frequency (MNF) of system (including piping and pulsation bottles) and excitation frequency should be at least 15-20%. Mechanical natural frequency (MNF) should preferably be greater than 2.3 times the maximum run speed [11,18]. If not meet limits, the force response evaluations (including extensive stress analysis and comprehensive fatigue studies) are required. The cylinder gas forces (also called “frame stretch” or “cylinder stretch force”) can be a significant source of excitation. These forces can cause high frequency vibration on the pulsation bottles and the connecting piping close to the compressor. It could lead to excessive vibration (such as pulsation bottle vibration) even if the pulsation shaking forces meet limits.

4.4. Integrated Dynamic Analysis

The dynamic analysis of the compressor package should include modelling and simulation of the foundation and the complete compressor package at the same time. The accuracy of this analysis is strongly influenced by the modelling of the foundation especially for pile foundations [18-20]. Generally a massive foundation is required for a reciprocating compressor. As an indication, the foundation mass should be around (or more than) 5 times of the combined compressor package mass. Foundation bolts should be designed with special care. Foundation bolts from a suitably, high strength steel (say above 700 Mpa yields) are recommended with relatively high initial tension to keep the foundation and the machine as a single unit during operation under sever dynamic forces. Foundation bolt tensions are usually set between 45-55% of the bolt material yield to offer excellent clamping forces and good reliability.

5. Case Studies

The first case study is a four-throw (four cylinders) process reciprocating compressors for a hydrocracker unit in a large refinery. The train speed is 330 rpm and the driver motor nameplate is around 8 MW. The expected operation pattern is suggested that compressor will work at the normal flow more than 90% of the total operation time. On this basis, step-less capacity control is not selected. Selected cylinder valve is ring type valve and selected capacity control method is the plug type unloder. The unloader system can offer operation at 50% and 75% of the full capacity. Operation at 25% capacity is prohibited because of rod load reversal issues and operational problems. A 50% recycle valve (bypass) is provided for operational flexibility. Pressure could reach more than 150 barg and lubricated cylinder is used. Lubricated cylinder and packing are only available option for this large and high pressure machine. Figure 5 shows rod load curves (the combined gas and inertia load) of this heavy-duty special-purpose reciprocating compressor.

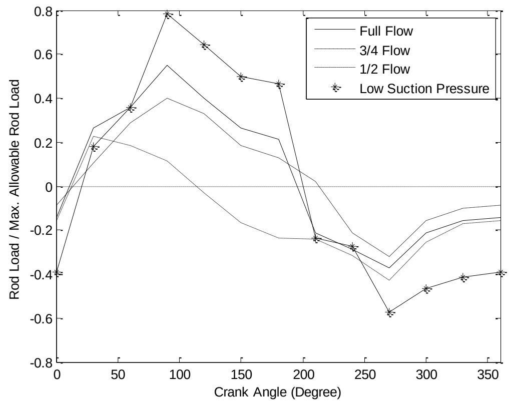

Figure 5 Rod load curves for various operating conditions of the reciprocating compressor of the first case study. Rod load (actual rod load/allowable rod load) vs. crank angle are shown.

The low suction case is identified as 8% suction pressure reduction compared to the normal operation (the lowest suction pressure could be 92% of the normal suction pressure). The low suction pressure causes the highest rod load. The lowest rod reversal duration belongs to the operation case at 50% capacity. This rod reversal duration is still more than 80º of crank angle. This plot shows that all possible process variations particularly suction pressure variations and any part-load operation should be respected in the rod load calculation.

The second case study is a two-throw (two cylinders) process reciprocating compressor for a refinery service. It is a medium size (around 500 kW) make-up hydrogen compressor for a large hydro-treating unit. Machine speed is around 356 rpm. Flange to flange connection (rigid connection) for direct coupling the electric motor and the compressor is selected because of better life and lower maintenance. For this machine, ring type valve and plug type unloader are specified. A 100% recycle valve is provided for an operational flexibility. It is an oil-free compressor (non-lubricated cylinder and packing).

The third case study is a two-throw (two cylinders) hydrogen reciprocating compressor for a petrochemical unit. It is a small-size machine, using non-lubricated technology with rated power around 90 kW. This hydrogen compressor operated at 490 rpm speed, which is an optimum speed for this tiny machine. Plate type valves with finger type unloaders, and a 100% recycle valve are provided. High-torsional-stiffness coupling (metallic coupling) is selected to obtain easier alignment with reasonable maintenance (and good operation).

6. Discussions And Final Notes

Large reciprocating compressors (>2MW) are important equipment used in a wide range of process industries [21]. Most of these compressors have huge power requirements and their capacity often needs to be regulated. Advanced capacity control technologies (such as step-less capacity control systems) are effective approaches for saving large amounts of energy in process industries.

An analysis of the premature failures [22] in reciprocating compressor particularly crankshaft failures, valve problems and unloader issues shows the importance of torsional studies and pulsation analysis. Special care should be taken for valves, unloaders and piston rod packing as the main reasons for unscheduled shutdown. Excessive vibration levels are usually observed in the reciprocating compressors [23]. It is necessary to perform a reliable finite element dynamic study to identify dynamic behaviour of the compressor. The rotor-crankshaft assembly should be considered as a flexible body, during this dynamic analysis. The rotor-crankshaft dynamic model should be updated by using experimental modal analysis. The forces of the pressure and the slider-crank mechanism should also be included in the dynamic model.

Pulsation fluid models for a reciprocating compressor and associated equipment and piping have become more sophisticated as the power and speed ranges for the compressor have increased [24]. This is because of the need to more reliably predict system responses for control of high pulsation and vibrations. Using more advanced computational models, designers are able to more accurately determine expected gas pulsation levels and corresponding response frequencies and mode shapes, resulting in more efficient and reliable pulsation bottle designs with lower power consumption. Pulsation models of reciprocating compressor systems commonly utilize a one-dimensional (1D) representation with acoustic length modifications to represent the three-dimensional (3D) system, given the simplicity and cost-effectiveness of this approach. One-dimensional transient fluid models are generally accurate for pulsating flow piping systems where the dominant physical length is in the flow direction. For high-speed reciprocating compressors (say above 500 rpm), in the areas near the compressor cylinder, or zones very close to the fluid force excitation from the piston, the 1D assumptions breakdown since many of the high-frequency energy components have not diminished. In addition, for new reciprocating compressor cylinder designs (any machine speed), certain 1D representations of the gas passage may not be valid and could lead to incorrect predictions. Uncontrolled responses associated with the cylinder gas passage system are primarily evident as higher frequency vibrations and high cycle fatigue failures at the compressor valves, cylinder body, cylinder nozzles and other components. Inaccurate designs will lead to use of cylinder nozzle orifice plates, poor valve life, and low compressor efficiency due to high dynamic pressure drop.

The 3D response analysis should be performed using modern finite element models to analyze the fluid space of the cylinder gas passage, nozzle, and pulsation bottle system. Comparing to the field data for the reciprocating compressor system, the 3D acoustic response model predicted responses in the gas passage system which matched the field data very closely [24]. The 3D acoustic modal analysis tool is an effective means of determining the natural responses of solid and fluid combined system (reciprocating compressor system and compressed gas). True 3D model provides a means of visualizing the acoustic response, which can be beneficial in determining means of attenuating, altering, or shifting the acoustic response. The utilization of 3D acoustic modal response modeling resulted in a better means of determining source modes and causes of high pulsation for the reciprocating compressor system. Used in combination for 1D pulsation filter designs, the 3D response modeling can help to avoid the use of orifice plates in the cylinder nozzles and other problematic cylinder resonances that can lead to high vibrations and valve failures.

7. Conclusion

Reciprocating compressors can fit to nearly all processes and plant sizes, using a proper concept and approach. Benefits of modern reciprocating compressors over old fashion units are discussed. The life cycle benefits are dominated by four factors: capital cost, plant availability, project schedule and plant productions. The expectation is that the capital cost is slightly higher for new technologies explained in this article. However, considerable improvements (efficiency, availability, reliability, and schedule) can well mitigate the impact of a slight increase in the capital cost.

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