World Academy of Science, Engineering and Technology
International Journal of Economics and Management Engineering
Vol:3, No:4, 2009

Reciprocating Compressor Optimum Design and Manufacturing with respect to Performance, Reliability and Cost

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***Abstract*—**Reciprocating compressors are flexible to handle wide capacity and condition swings, offer a very efficient method of compressing almost any gas mixture in wide range of pressure, can generate high head independent of density, and have numerous applications and wide power ratings. These make them vital component in various units of industrial plants. In this paper optimum reciprocating compressor configuration regarding inter-stage pressures, low suction pressure, non-lubricated cylinder, speed of machine, capacity control system, compressor valve, lubrication system, piston rod coating, cylinder liner material, barring device, pressure drops, rod load, pin reversal, discharge temperature, cylinder coolant system, performance, flow, coupling, special tools, condition monitoring (including vibration, thermal and rod drop monitoring), commercial points, delivery and acoustic conditions are presented.

***Keywords*—**Design, Optimum, Reciprocating compressor.

I. INTRODUCTION

RECIPROCATING compressors are the most common type of compressors found in industrial applications [1]-[4]. **Worldwide installed reciprocating compressor horsepower is approximately three times that of centrifugal compressors** and maintenance costs of reciprocating compressors are approximately three and half times greater than those for centrifugal compressors [3]. Design and manufacturing of a reciprocating compressor shall be involved optimization process with respect to full array of compressor data. Otherwise, it may end up with equipment that may not be suitable over the full operating cases, reliability level, commercial terms or power requirement for which the manufacturer and client expect.

II. MACHINE DESIGN

A. Inter-stage Design Pressures

Discharge pressure of each stage is normally protected by pressure relief valves, high pressure discharge switches are seldom seen [2]. Optimum inter-stage pressures can be obtained by formulation and optimization of performance and investment for compressor and inter-stage facilities. **Inter-stage pressures based on just compressor without respect to inter-stage facilities, are not justified**. Inter-stage pressures are going to increase during part load operation and high 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 compromise to fix PRV-Pressure Relief Valve set points (and inter-stage facility design pressure accordingly) based on worst case of part load operation and process variations (such as high suction pressure). For common design cases (50% and 75% part load steps and around 5%-7% suction pressure variation), PRV pressure set points will be around 10-15% higher due to part load operation and suction pressure variations.

B. Low Suction Pressure with Full Flow

Sometimes, due to process requirements, reciprocating compressors shall be capable to operate at lowest suction pressure and full design flow at normal discharge pressure. It can have strong effect on electric motor sizing as well as compressor sizing and machine frame rating (for example more than 35% electric motor power increase for 20% suction below normal). It is absolutely necessary to respect this condition for motor and compressor sizing. It is optimum to respect them in basic design and design stage to avoid costly changes.

C. Minimum Speed Lubricated Cylinder

Reliable machine is involved low speed (around 350 RPM) and lubricated cylinder. Optimum piston speed is 3-4.4 m/s. Optimum configuration: horizontal cylinder(s), discharge nozzle on the bottom side. For small compressor selection, it may be necessary to deviate lubricated cylinder or low speed. **Optimum option is lubricated cylinder with available lowest speed machines.** Probably less than 20% of all reciprocating compressors are designed for non-lubricated operation just because of process demands (oxygen, high pressure air, down stream facilities sensitive to oil such as some catalysts, etc) [2].

D. Unloader and Capacity Control System

Step-less capacity control system uses finger type unloader, is pneumatically actuated, and unloads the suction valve for only a portion of compression cycle to achieve adjusted

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capacity [2]. Finger type unloaders have potential for damaging the valve sealing elements and require more care for maintenance [2]. Valves and unloaders cause around 44% of unscheduled reciprocating shut down [5], [6] and this selection has a strong effect on reliability [5]-[8]. **For small machine, 100% spill back is optimum solution, because power is low. For big machine optimum configuration is selection of part load steps based on plug/port unloader and clearance pocket.**

E. Condition Monitoring

Condition monitoring [9]-[13] shall be particularly cost effective and include necessary items to identify malfunctions at an early stage (lower maintenance costs and risk of accidents). Optimum vibration monitoring: 1- Vibration – continuous monitoring (Shut down). **Velocity transducers are preferred over accelerometers** due to better signal to noise ratio [9]. The optimum configuration: each end of the crankcase about halfway up from the base plate in line with a main bearing [9]. 2- Each cross head accelerometer (Alarm). 3- Electric motor vibration (Shut down). Optimum temperature monitoring: 1- High gas discharge temperature - each cylinder (Alarm and Shutdown). 2- Pressure packing case - piston rod temperature (Alarm). 3- High cross head pin temperature (Alarm). 4- High main and motor bearing temperature (Alarm). 5- Valve temperature (Monitoring). 6- Oil temperature out of frame (Alarm). 7- High jacket water temperature - each cylinder (Alarm). Optimum implementation is properly set trip levels that are just high enough over the normal operating levels to reach to mechanical failures, but not so high as to miss the failure prior to catastrophic release [9]. Proximity probes are typically located under the piston rods [9] and used to measure the rod position and determine wear of the piston and rider bands, malfunction e.g. cracked piston rod attachment, a broken crosshead shoe, or even a liquid carryover to a cylinder. Optimum figure: just for alarm and not for shut down. Optimum cold run outs and normal conditions operating run outs are about 50 micrometers (2 mils) and on the order of 50 to 150 micrometers (2 to 6 mils) peak to peak, respectively [9].

F. Valve Selection

Cylinder valves are the most critical components of reciprocating compressors and strongly influence the reliability and efficiency [5]-[8], [10]. Valve defects are obviously responsible for most of the unscheduled maintenance events [5]-[8], [10]. Three main valve types: ring type, ported plate and poppet. **For big machines (generally low speed and high pressure ratios) and small machines (relatively higher speeds) ring type valves and plate type valves are optimum choice respectively.** Optimum valve size shall be obtained with respect to efficiency, reliability and performance requirements including minimum clearance volume. Lift is the distance travelled by the valve moving elements. The higher the lift, the higher the valve flow area, lower the valve pressure drop, less consumed power, higher moving elements impact velocities and lower valve durability. Acceptable compromise should be found. Optimum valve spring stiffness is also important. Too stiff spring can lead to valve flutter (more compressor power and considerable wear rate) or early closing of valve (reduce capacity). Too light spring cause valve late closing and the reverse flow (higher velocity, less reliability and reducing capacity). Nonlinear partial differential equations describing the valve differential pressure and the valve element motion (such as [14]) can be used in optimization process to estimate optimum valve lift, spring stiffness and gas velocity for each machine and application.

G. Piston Rod Coating

Piston rod seal is second important area for reliability of reciprocating compressor and most likely path for potentially hazardous process gas leakage [8]. Packing life could be improved three times by adding the proper tungsten carbide piston rod coating [10]. It is optimum selection.

H. Cylinder Liner Material

Cylinder liner is used to provide a renewable surface to the wearing. The liners made by Ni-Resist cast iron (high Nickel content) are not recommended due to problems such as permanent distortions. **Optimum selection is grey cast iron** [8] except very high pressure or extremely high corrosive applications.

I. Passive Vibration Reduction System

Sometimes odd number of cylinders is not avoidable. In this case dummy crosshead shall be used to reduce vibration. Also spring-mass-spring system shall be studied for passive force counter balance and more reduction in vibration, where dummy crosshead is, on the one hand, flexibly attached to a movable piston assembly and on the other hand, to the stationary compressor casing using auxiliary mechanical springs.

J. Future Expansion

Future expansion planning can save money and time if process changes (capacity increase, molecular weight increase due to catalyst change, etc) are foreseen [12]. **Optimum selection is sizing cylinders for economical operation at the present rate. The frame can be sized for future applications**. When the future conditions become a reality, the cylinders can be changed while keeping the same frame. Generally it is optimum to over size the journal diameter include margins for future development, thus ensuring that crankshaft size would never become the first important limitation of the design [15].

III. COMMERCIAL OPTIMUM CONDITIONS

A. Commercial Conditions

It is absolutely necessary to receive at least three proposals and have minimum two technically accepted offers for main components and equipment. It is completely justified to extend proposal dead time, clarification time, accept optimum configuration, reasonable deviations and attend extensive clarification meetings to have at least two clarified and technically accepted proposals.

B. Delivery

Small and medium machines shall be delivery fully fabricated as one skid mounted package. **For very big machine, optimum figure is to deliver machine prefabricated (including crankcase, distance pieces, etc) while cylinders are dismantled. Assembled cylinders are delivered to site separately and installed.** Sometimes it is required to offer all site supervision work for cylinder installation as closed price.

IV. Auxiliaries and Accessories

A. Lubrication System

API 614 is typically applied only to reciprocating compressor trains involving a large turbine driver and gear unit [11]. Optimum oil system shall include two oil pump, both sized at least 20% over (Two motor driven identical with run down tank, or well known crankshaft driven main oil pump, supplying UPS power for one pump is not acceptable alternative), dual removable bundle shell and tube oil coolers (TEMA C) and double oil filters with removable element and stainless steel piping.

B. Coupling and Torsional Analysis

Usually the potential exits for torsional resonance and torsional fatigue failure [2]. Coupling is best available option for modification to tune the system. Coupling option: 1- **High torsional stiffness coupling (it is optimum if allowed by torsional analysis)**. 2- Flexible coupling (more elasticity and damping and more maintenance). 3- Direct forged flanged rigid connection (no coupling), with single bearing motor. Coupling for special purpose machines shall be as per API 671.

Typically, for reciprocating compressor, lateral natural frequencies will be positioned well above significant torsional natural frequencies, so **lateral critical studies are not required**. A stress analysis shall be performed if the torsional excitation falls close to the torsional natural frequency to ensure that the resonance will not be harmful for the system. The torsional vibration analysis report shall include data used in mass elastic system, display of force vs. speed, torsional critical speeds, deflection (mode shape diagram), worst case design, upset condition results (such as start up, short circuit, electrical network faults, etc) and how the input data variance will affect the results (sensitivity analysis). Continuous operation at torsional resonance shall be avoided. Changing the load sequence could help reduce torsional vibration. Synchronous motor or system started on a frequency basis need more care (definitely need a transient torsional start up analysis). As a rue of thumb, **electric motor shaft diameter to be equal to or greater than the compressor crankshaft diameter.**

Generally coupling shall be capable to allow continuous operation with twice the estimated cold-to-hot thermal growth. The coupling to shaft connection shall usually be rated for a minimum of 125% of the maximum driver power. Numerous torsional vibration problems continue to occur in compressor trains. Main reasons are lack of comprehensive torsional vibration analysis and study, improper application and maintenance of coupling (mainly flexible couplings) and lack of monitoring.

Shaft materials should be high strength steel. If welds are required on shaft, a weld-able shaft material should be used. Proper weld procedures and material compatibility must be considered. Fabrication details such as the electric machine pole bolt torque, etc should consider loads due to torsional vibration. Avoid full load shutdowns for compressor especially in train with torque sensitive equipment. Some designers offer to use stiff system with short flanged connection (no coupling) and single bearing electric machine. These designs have much lower damping however higher natural frequencies (more rugged, less elasticity and damping). These designs may be acceptable after careful review of torsional analysis including all possible operating steady state and transient torsional situations.

It is important to measure and verify torsional vibration during performance test. Based on site observations following transient events are critical and shall be respected in details: start up, short circuit, machine possible malfunctions (such as valve failure in reciprocating compressors) and loaded shut down. A train which passes through a torsional natural frequency during start up may produce significant transient shaft stresses. If the system is started on a frequency basis, a start up analysis should be performed to determine if low cycle fatigue is a potential problem. Coupling torque is usually chosen on the basis of mean requirements for full load. It must have a sufficient service factor to handle and likely overload (such as electrical faults). Minimum recommended service factor for special purpose units is"1.75".

C. 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 valve moving elements (slugging) and reduce the lubrication effectiveness. For any application, a good sized suction drum with a drain provision shall be included [12]. It may be a part of pulsation control, if properly done. Cylinder cooling system must be monitored and controlled. **Coolant inlet temperature between 6 ºC and 16 ºC above inlet gas temperature** [12]. For exotic gases or operations near critical areas, much care needs to be taken also make sure the thermodynamic model is proper.

D. Pulsation Control

Optimum technique trends to dissipate less energy than reliance on special solutions such as orifices to control pulsation levels [16]. Acoustic reviews shall be performed for design and guarantee points as well as all other operating cases and combinations of pressures, speeds and load steps [13]. Pulsation can also alter the timing of the valve motion and decrease efficiency and reliability [5]. Preliminary optimum value for maximum allowable peak to peak is recommended as (1), in SI units, where "P_1" and "P_L" are percentage of average absolute pressure (%) and line pressure (Bara), respectively.

$$P_1(\%) = \frac{6}{(P_L)^{1/3}} \tag{1}$$

E. Pulsation Shaking Forces

Reduction of pressure pulsation can be accompanied by an increase in shaking forces (or unbalanced forces) [17]. It illustrates that shaking forces shall be determined and controlled and piping and vessels properly supported. The margin of separation between the mechanical natural frequency (MNF) of system (including piping and bottles) and excitation frequency is 20% and MNF shall be greater than 2.4 times maximum run speed [11], [18]. If not meet limits, the force response (including stress analysis) is required. The cylinder gas forces (also called frame stretch or cylinder stretch force) can be significant source of excitation (can cause high frequency vibration on the bottles and piping close to the compressor) and lead to excessive pulsation bottle vibration even if the pulsation shaking forces meet limits. Flow induced pulsation is rarely seen [17]. API 618 Design Approach 3 and less rigorous analysis, to control pulsation and shaking (unbalanced) force levels and avoiding mechanical resonance can result in an optimized design [17]. Pulsation and vibration analysis report shall include Time Domain (TD) and Frequency Domain (FD) simulations, Time Domain (TD) plots of key forces and pressure pulsation, dynamic pressure drop, models including mounting details (mounting plate, bolts, localized skid, etc) and shell flexibility (nozzle connection flexibility), calculated cylinder stretch forces, mode shape of bottles and piping and compressor stiffness assumption (compressor frame modeled as flexible support) [18].

F. Coolers

Inter stage cooler and after cooler shall be sized carefully. Undersized cooler can cause excessive pressure drop and power loss. Pulsation and shaking force studies are necessary to avoid vibration problem in cooler. Increased cooler cross section area to decrease pressure drop can cause significant increase of shaking force and cooler vibration. Secondary volumes may be studied to reduce this vibration however in some cases it can not reduce it and modification of recycle line are required to significantly lower shaking [17].

G. Dynamic Package Analysis

The dynamic package analysis shall include modeling and simulation of the foundation at the same time. The accuracy of this analysis is strongly influenced by the design of the foundation especially for pile installation [18]. It is even more important for packages mounted on offshore platforms, FPSO (Floating Production Storage Offloading Vessels), modules mounted on steel structure, new and unproven skid design and where the local soil conditions are suspect. **Skid lifting study (including lifting lug details and calculations review) is necessary.** Transit study and environmental loading analysis are also recommended [18].

H. Barring Device

When compressor stopped for an extended time, turn it around a quarter turn every week by barring device. Manual barring device is for small compressor. Pneumatic barring device is for compressor rated over 750 KW, without area classification problem or intermittent power availability and preferred technically [11].

I. Special Tools

Optimum check list for special tools [11] for big machines: 1- bearing extractor 2- piston extractor 3- valve extractor 4- piston fit up tool 5- hydraulic tightening system 6- crosshead assembling tool, 7- special lifting tools 8- partition plate assembling tools 9- mandrels for wear bands. For special tools, tool boxes required and they shall be delivered with main machines, in separate and tagged boxes.

J. Package General Arrangement

Layout is often complex and compromise must be made between the support requirements, process requirements, vibration/pulsation conditions, access and maintenance. **Optimum configuration is to install local panel near compressor package (around 200 mm away from package) but on separate skid (frame) which is installed on foundation, to avoid vibration damage.** For maintenance, consider spool removal and avoid support removal. **Access required in front of cylinder for cylinder piston dismantling and non-drive end of compressor.** Pay attention to minimum elevation requirement of pulsation bottle suction flange to keep suction line "No Pocket".

K. Piping Analysis

The thermal piping design often requires that flexibility be added to the system which is counter to requirement for more support and increase stiffness to meet vibration design. These analyses shall be conducted by same party to optimize design iteration and result in an overall optimized system. Piping thermal analysis is necessary [18] especially when the coolers are off-skid, multiple compressors on a common header, extremely cold ambient temperature, or operation over a very wide range of conditions.

L. Cylinder Lubrication System

Using the proper type of lubricant as well as establishing the proper lubricant rates to the cylinder and packing can be most important for machine reliability [10]. The life of the compressor valves, piston rings, rider bands, and pressure packing can all be significantly affected by the type and quality of lubrication used. Too much or wrong type of lubrication can increase the effects of valve stiction (viscous adhesion) and reduction in reliability. Reliable lubrication system with optimum type and rate of lubrication shall be selected [10].

V. Performance

A. Expected Pressure Drops

Preliminary optimum pressure drop values: pulsation dampeners and suppression devices: totally less than 1% pressure, intercooler: 0.70 bar. The use of orifice plates, especially on high-speed single-act compressor, can contribute to significant pressure drops [13].

B. Rod Load and Pin Reversal Details

Maximum Rod Load is recommended to be less than 80% of allowable rod load. Duration and peak magnitude load of rod reversal shall not less than 15º of crank angle and 3% of the actual combined load [11] in the opposite direction, respectively and shall be checked for all possible operating cases [13], [19] (especially low suction and part load steps).

C. Highest Expected Discharge Temperature

High discharge temperatures cause problems with lubrication cooking and valve deterioration [12] and shall be reviewed at least for average and maximum suction temperatures [13]. The maximum predicted discharge temperature [1], [2], [4], [11] shall not exceed 150ºC and not exceed 135ºC for hydrogen rich service (MW of 12 or less). **Gas discharge temperatures less than 118ºC tended to extend life of wearing parts** [10]. It is optimum figure.

D. Performance Curves

Required Performance Curves [13]: 1- Suction Pressure vs. Load. 2- Suction Pressure vs. Flow. 3- Discharge Pressure vs. Load. 4- Discharge Pressure vs. Flow. 5- Suction Pressure vs. Discharge Pressure, per load step. Performance curves are used to safely control the unit across its defined operating range [13].

E. Flow Details

Flow details are flow curves from unit’s minimum achievable flow rate to its maximum achievable, in specified increments [13]. Alternative is flow versus discharge pressure plots of specific suction pressures (more compact and common when suction pressure variation is limited).

F. Load Step Curvature

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 pressure can have significant changes in load and flow. **Often, steep load curves may indicate improper sizing of cylinders** and units with steep load step curves can also prove difficult to automate and tune [13].

G. Machine Shop Run Test

Shop mechanical run test [11] is the first test after manufacturing of machine and last test before delivery. Shop test results, including vibration, may seem to have limited usefulness because supporting structure and operating condition of the machine are different with final site installation. However this test is a unique opportunity to find defects in design and manufacturing phases while machine is still in fabrication shop. Generally the lager the power per compressor throw leads to higher the dynamic forces and 370 KW per throw or more is a high risk machine [20]. It is justified and optimum to record and analyze shop run test measurements (Kalman Filter can be used to optimally evaluate machine dynamic characteristics based on measured data).

VI. Conclusion

Optimum configuration of reciprocating compressors regarding component design and manufacturing, commercial points, auxiliary and accessories, performance and reliability are addressed in this paper.

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