World Academy of Science, Engineering and Technology
International Journal of Mechanical and Mechatronics Engineering
Vol:3, No:7, 2009

Reciprocating Equipment Piston Rod Dynamic Elastic-Plastic Deformation Analysis

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Abstract—Analysis of reciprocating equipment piston rod leads to nonlinear elastic-plastic deformation analysis of rod with initial imperfection under axial dynamic load. In this paper a new and effective model and analytical formulations are presented to evaluate dynamic deformation and elastic-plastic stresses of reciprocating machine piston rod. This new method has capability to account for geometric nonlinearity, elastic-plastic deformation and dynamic effects. Proposed method can be used for evaluation of piston rod performance for various reciprocating machines under different operation situations. Rod load curves and maximum allowable rod load are calculated with presented method for a refinery type reciprocating compressor. Useful recommendations and guidelines for rod load, rod load reversal and rod drop monitoring are also addressed.

Keywords—Deformation, Reciprocating Equipment, Rod.

I. Introduction

RECIPROCATING machines are most efficient available machines [1]–[4]. Reciprocating equipment (reciprocating compressors, reciprocating pumps, gas and diesel engines, reciprocating expanders, etc) are vital components in industrial plants. They are flexible to handle wide capacity and condition swings. In many cases the reliability of plant is dictated by the reliability of the reciprocating machines. Important topics related to these machines are rod load and piston rod reliability [5]. Reliability assessment of reciprocating machine piston rod leads to nonlinear deformation analysis of rod with initial imperfection under axial dynamic load. It presents a real challenge due to complex nonlinear deformation, elastic-plastic behavior and dynamic effects. Previous proposed Finite Element Methods (F.E.M.) are restricted in various limits and uncertainties and relatively accurate results need countless meshes, continual re-meshing and other special considerations. The reciprocating machine piston rod is subjected to a combination of fluid (gas in compressors and engines and liquid in pumps) and inertia (of reciprocating and rotating components) loads. Rod load can be influenced by changes in operating pressures, capacity changes [6]–[8], unloading, valve leaks and other causes. Rod loading must be kept within the limit set because overloading can cause excess run out of the piston rod resulting in premature packing wear, leakage, reduced efficiency, increased maintenance expense and potential of catastrophic failure including risk of personnel injury [8], [9]. Also with the pollution laws becoming more stringent, leakage control takes on a much greater significant.

This paper presents model, formulation and simulation routine for predicting piston rod deformation and rod load limit. This formulation and routine can be used to generate machine loading tables and rod run-out values that permit operators to load machine confidentially at rated capacity for best efficiency and maximum throughput. Without these data, low confidence placement may lead to low load factor and low capacity or serious effects of overloading.

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II. Definition of Problem and Model

Experience with high pressure reciprocating machines revealed that piston rod problems are important sources of machine failure and machine unexpected shut down for repair. These failures consisted of severe scuffing and wear between sealing element (piston and packing rings) and the counter surface (liner and piston rods) causing erratic seal life, sometimes only a few hours duration, together with badly damage cylinder liners and piston rods. The rod load varies from compression to tension and vice-versa along the piston stroke. Load carrying limit referred as rod load or frame load is maximum continuous operating force that machine can safely and reliability withstands. It is load that can be borne by the crankshaft, connecting rod, frame, piston rod, bolting and bearing. Knowledge of rod load has great effect on reciprocating machine safety and reliability. There were many reports about insufficient accuracy in the machine manufacturer's rod load data. Discrepancies more than 20% were reported between machine manufacturer load curves and reciprocating machine site performance results [10]. For old machines reported differences and discrepancies are considerably higher. For safe and reliable operation, operators should be furnished with accurate and reliable rod load data because operating conditions and pressures can be varied due to many reasons. Operator shall be able to keep machine running confidentially and safely using proper data and load tables and curves. It is necessary to develop a more precise method for prediction of reciprocating machine piston rod loads. The piston rod load depends on the cylinder fluid pressure and reciprocating machine component inertia forces. One of the most important issues in this regard is piston rod deformation and piston rod load bearing capacity which will be addressed in this paper. In many cases, piston rod stress or deformation can be the limiting factor or the "Weakest Link in the Chain".

Axial position of the piston depends upon crank angular position and connecting rod-crank ratio. The piston velocity is zero at beginning of the stroke and decrease zero at the end of stroke. Piston acceleration is maximum at each end of the stroke where piston changes direction. Acceleration is zero near mid-stroke where the velocity is maximum. Forces acting on reciprocating machine moving parts are: 1- Cylinder fluid pressure forces. 2- Inertia and dynamic forces of reciprocating parts. 3- Centrifugal forces of rotating parts. The cylinder fluid pressure forces are main loads. Inertia, dynamic and centrifugal forces are usually considerable for big machines (with heavy moving masses) or high speed machines [11]. The total load of cylinder fluid is found by a simple multiplication of the pressure times the piston area. If the piston is double-acting, the sum of loads on head end and crank end of piston must be taken into account. The inertia rod load is proportional to the square of machine speed. Inertia loads become important for large and heavy pistons, high speeds or little pressure differences. In most cases combined loads (fluid plus inertia) are lower than the fluid loads. This may not be true for low pressure ratio (high volumetric efficiency) applications. As an approximate rule, if the discharge Volumetric Efficiency (VE) is less than 50% (which is very common), the fluid load will reach a maximum after the 90 degrees, so it is opposite in sign to the inertia load. In this paper main concern is rod deformation of process reciprocating machines which have normally slow speed and high pressure differences. Inertia loads are small compare with fluid loads for these machines. However both cylinder fluid load and inertia load are considered in presented model and simulation. It is necessary to note that in-cylinder pressures are different with nominal pressures due to several effects such as valve and cylinder passage losses (typically 2-10% difference), pulsation control devices (around 1% difference), cylinder valves (typically < 7% difference) and valve dynamics (inertia, sticktion, flutter, etc). These effects shall be addressed in accurate rod load calculations.

Fig. 1 shows schematic diagram of mechanism of special purpose heavy duty reciprocating machine.

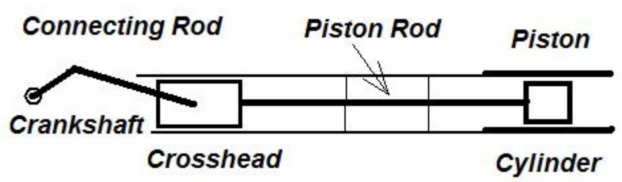

Fig. 1 Schematic diagram of mechanism of special purpose heavy duty reciprocating machine.

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Only axial loading is considered and only transverse inertial are taken into account. Fig. 2 presents model for piston rod elastic-plastic nonlinear deformation.

The elastic-plastic deformation in form of elastic-plastic hinge takes into account the coupling effects between bending and compression. Large numbers of elements in the elastic-plastic hinge (more than 30) are considered to obtain accurate results.

Fig. 3 shows model for elastic-plastic hinge assumed in middle of piston rod. This model can simulate the progression of thin plastic zones in both sides of rod cross section. This elastic-plastic nonlinear deformation can model small values of run out resulted from overloading. Piston rod is modeled by an initially imperfect rod with initial length ($2L_o$) with reference to Fig. 2. As noted there is an assumed elastic-plastic hinge in the middle of rod. Initial imperfection is shown by (y_o). Mass (m_R) is assumed concentrated mass at elastic-plastic hinge in the middle of rod. Parameter (α) and (a) are deflection angle of half of rod and initial half height of middle of the elastic-plastic hinge respectively.

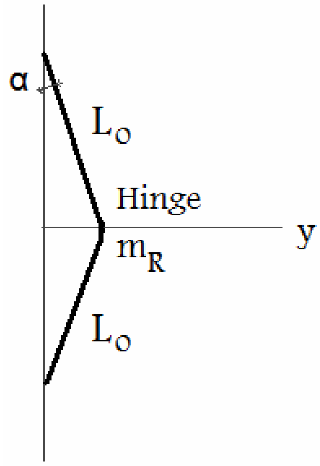

Fig. 2 Model for piston rod elastic-plastic nonlinear dynamic deformation.

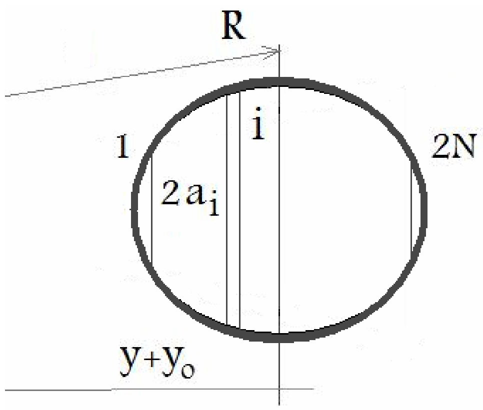

Fig. 3 Model for elastic-plastic hinge in middle of piston rod.

III. Formulation

Using "Principle of Virtual Work" [12], [13], rod equations of motion can be expressed as (1) and (2). In these equations parameters "F", "y", "M" and "N" are load transmitted by the rod, transverse deflection of rod due loading, bending moment in the elastic-plastic hinge and force in the elastic-plastic hinge, respectively. As noted parameter (α) is deflection angle of half of rod and mass (m_R) is assumed concentrated mass at elastic-plastic hinge in the middle of rod.

$$2F(t)\cos(\alpha(t)) + m_R \frac{d^2y}{dt^2}\sin(\alpha(t)) + 2N(t) = 0 \quad (1)$$

$$2M(t) - 2F(t)L_o \sin(\alpha(t)) + m_R \frac{d^2y}{dt^2} L_o \cos(\alpha(t)) = 0 \quad (2)$$

Functions "M" and "N" are time dependant moment and force in the elastic-plastic hinge and expressed as (3) and (4) respectively.

$$N = \cos(\alpha) \sum_{i=1}^{2N} \sigma_i \ \Delta A_i \quad (3)$$

$$M = \sum_{i=1}^{2N} \sigma_i \ z_i \ \Delta A_i \quad (4)$$

Parameters "ΔA_i", "z_i" and" σ_i" are area of i-th slice of elastic-plastic hinge, abscissa of the middle of i-th slice of elastic-plastic hinge and stress in the i-th slice of elastic-plastic hinge. With respect to presented model, (5) to (7) can be written.

$$a_i(1+\varepsilon_i) = (R + z_i)\sin(\alpha) \quad (5)$$

$$L_o \sin(\alpha) = y + y_o \quad (6)$$

$$\sigma_i = f(\varepsilon_i) \quad (7)$$

Parameters "a_i", "ε_i" and "R" are half height of middle of the i-th slice of elastic-plastic hinge, strain in the i-th slice of elastic-plastic hinge and elastic-plastic hinge curvature radius. Function "f" in (7) is rod material constitutive strain-stress relation. In the initial state, (8) is valid. As noted "y_o" is initial imperfection of rod and parameter "α_o" is initial deflection angle of half of rod.

$$\sin(\alpha_o) = {}^{y_o}\!/_{L_o} \quad (8)$$

With respect to (1) to (7), rod dynamic deformation, details of stresses and possible permanent deflection can be calculated with knowledge of the force-time relation "$F=F(t)$" which comes from cylinder fluid and inertial forces of reciprocating machine. End results of these calculations can be maximum allowable rod load for machine using recursive procedure. Material strain rate sensitivity is also considered in formulation (in (7)). To model the strain rate effects, empirical "Cowper-Symond" equation is used.

IV. Results

Fig. 4 shows rod load curves (combined gas and inertia load) of heavy duty special purpose reciprocating compressor in refinery service. This rod load and allowable rod load are calculated based on presented method.

To calculate allowable rod load, operating run-out is limited to 150 micrometer (6 mils) peak to peak. As indicated, acceptable rod load peak in design condition shall be less than 80% of allowable rod load. However in operating conditions, rod load can be slightly increased to improve machine performance. For this purpose accurate model (such as present model) shall be used and all safety and reliability steps and regulations shall be followed and assured.

As shown in Fig. 4, variation in suction pressure (in this case low suction, around 7% suction pressure reduction) causes higher rod load. It reminds that all possible process variations especially suction pressure (lower or higher than normal operation) shall be respected in rod load calculation. Working in relief condition (discharge pressure equal by relief valve set pressure) has same effect and can cause higher rod load. Fig. 4 shows also that unloading steps can change rod load behavior and curves. For this compressor, 3/4 flow and 1/2 flow are obtained by clearance pocket and valve unloder respectively. As shown in Fig. 4, clearance pocket (3/4 flow) results in slightly lower rod load with same pattern as rod load curve of full flow. However valve unloder (unloading one side of double acting cylinder as 1/2 flow) results in completely different rod load behavior. Actually case of 1/2 flow (valve unloder) shows rod load generated by one side of cylinder.

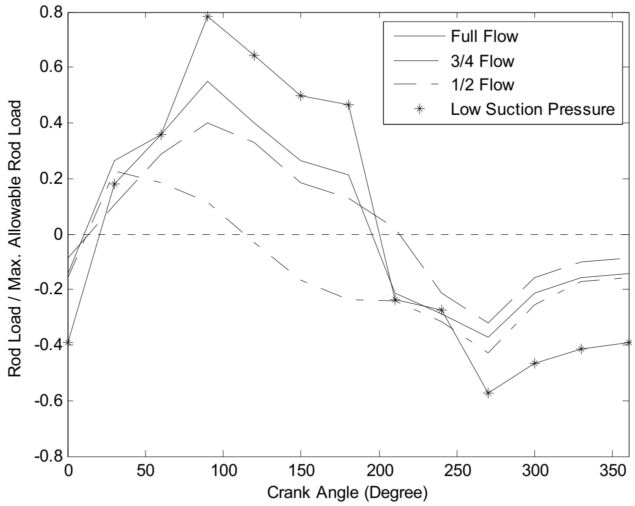

Fig. 4 Rod load curves for various operating conditions of reciprocating compressor.

Based on operation experiences and extensive simulations, reciprocating machine capacity can be increased from less than 90% to more than 96% of rated capacity (rated flow) using presented analysis method. For a 20 MW reciprocating machine station, it yields to more than 300,000 USD annual benefit and fuel saving in addition of maintenance savings. Presented method can also avoid overloading and unnecessary shut down of machines. For a typical heavy duty reciprocating machine in refinery service (for example in the middle size make up hydrogen reciprocating compressors) each day of shut down means more than 200,000 USD loss of profit.

V. Rod Load Reversal

To assure adequate lubrication of the crosshead pin, the load vector should change director during the piston stroke. The combined dynamic piston rod load changes direction from compression to tension. This change in direction allows clearance between the cross head bushing and the cross head pin in the area of the point of lubrication. Similarly, in single acting crank end loading, there is a change in direction of loading along the cylinder axis. The lack of change of direction of the dynamic piston rod load effects on cross head bushing lubrication and heat is generated between the cross head bushing and cross head pin. The heat causes the seizing of the pin with the crosshead bushing. It can result in catastrophic failure.

The lack of direction change is also referred to as "Lack of Reversal". It is necessary to check this issue for all possible operating cases including suction pressure variations, unloading steps, etc. Each reciprocating machine model has own minimum requirement for this reversal. Duration and peak magnitude load of rod reversal shall not be less than 15º of crank angle and 3% of the actual combined load in the opposite direction, respectively. These minimum requirements shall be checked for all possible operating cases (especially low suction and part load steps) and also proper safety margins shall be respected.

VI. Practical Recommendations

Emphasis on load monitoring has reduced the instances of overloading while simultaneously raised the average load. In reciprocating machines proximity probes are typically located under the piston rods and are used to measure the rod position (piston rod drop being measured continuously by a proximity sensor). It is also to determine wear of the piston and rider bands [7], [8], [14].

In the event of a malfunction such as overloading, permanent plastic deformation of rod, a cracked piston rod attachment, a broken crosshead shoe, or even liquid carryover to a cylinder, the operating rod run-out will increase significantly. Alarm will show excessive run-out. By this monitoring method, malfunction can be found in early stage before major damage. Cold run-outs are usually held to about 50 micrometer (2 mils) peak to peak and operating run-outs under normal conditions are typically in the order of 50 to 150 micrometer (2 to 6 mils) peak to peak. Although the theory is simple, getting and interpretation actual online monitoring presents challenge. Shut down is not recommended for rod drop monitoring to avoid unnecessary shut down of machine for example in transient cases or monitoring errors. Alarm set point shall also be selected very carefully with respect to all operating conditions, machine current situations, detailed analysis as presented in this paper and experience.

Surveys identified that piston rod and packing life on high pressure reciprocating applications could be improved two or three times by adding the proper tungsten carbide piston rod coating. It is strongly recommended even for small reciprocating machines.

Reciprocating machine can be tested several times annually using electronic analyzer equipment. This practice is cost effective and can reduce maintenance expense through early detection of worn or damage parts. It also verifies cylinder pressures for actual rod load calculation. Pressure velocity analysis is a technique that has proven to be very effective in assessing the condition of reciprocating machinery. Dynamic pressure transducers are used to measure the pressure inside the cylinder over the course of the stroke. This allows the analyst to evaluate the condition of the rod, packing rings, valves, etc, while at the same time measuring the dynamic piston fluid forces for piston rod load calculation.

The electronic analyzer simulates the indicated mean effective pressure and applies appropriate multipliers for speed and cylinder swept volume to model the piston rod load and power. One type of analyzer simulates piston velocity, and thus volume, with voltage generators. Another type of analyzer uses a digital computing device to calculate a table of volume increments as a function of crank rotation. It is beyond the scope of this paper to provide an assessment of electronic analyzer detail and accuracy. Simply stated the analyzer monitors the pressure and relates it to volume by tracking the crankshaft motion. This monitoring technique is strongly recommended.

VII. Conclusion

In many cases the reliability of plant is dictated by the reliability of the reciprocating machines. Important topics related to these machines are piston rod load and piston rod reliability. Safety, reliability and service life of reciprocating machines depend to a large extent on the operating of the piston rod mechanism and related packing. In this paper an effective analytical model and simulation method are presented to evaluate reciprocating machine rod deformation. Presented method is able to account for geometric nonlinearity, elastic-plastic deformation, dynamic effects and different loading profile of various reciprocating machines. Analytical simulation results are discussed. Useful recommendations and guidelines for rod load, load reversal and rod drop monitoring are also presented.

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