USING SIMULATION OF RECIPROCATING COMPRESSOR VALVE DYNAMICS TO IMPROVE ECONOMIC PERFORMANCE

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Reciprocating compressor performance can sometimes be improved by subtle changes in valve design. Modelling valve behavior can lead to optimum performance without resorting to a field trial and error approach. Three compressor valve studies presented here demonstrate the benefits of valve modelling.

In the first case, the capacity of a compressor is *increased* by reducing the valve lift. Plotting BHP/MMSCFD versus valve lift assists in optimizing the design. This case also presents a method of calculating the economic effect of improvements in valve performance.

Case two demonstrates the effect of inadequate flow area through the valve. Pressure in the clearance volume cannot decrease fast enough if flow areas are inadequate; the result is late valve closure, and therefore decreased valve life.

The importance of considering the design of the cylinder casting in addition to that of the valves is shown in the third case. Here, insufficient cylinder flow area restricted gas flow.

Using modelling, the analyst has the opportunity to evaluate the proposed solution over the entire range of operating conditions. This enables selecting a valve design that solves the immediate problem and that will perform adequately throughout the specified range of conditions.

BENEFITS OF VALVE SIMULATION

The main advantage of valve simulation is that it replaces trial and error as a means to solve valve problems, and thus saves a considerable amount of time and expense. Trial and error, although successfully used in some cases, is time-consuming and problematic. Some quantities such as valve displacements and impact velocities in an operating compressor are very difficult to measure. It is often impossible to measure results over the range of operating conditions, or to “measure” increases in valve life (in a timely manner) for different trials. The use of simulation as a valve design and trouble-shooting tool overcomes these problems.

Short valve life results in lost revenue as well as high valve repair costs and potential consequential damage. Poor efficiency is costly due to wasted energy consumption and in some circumstances can decrease compressor capacity. Good valve design involves a balance between efficiency and valve life.

This paper shows results of valve dynamics simulations for analyzing compressor and valve performance. The simulation was developed based on three previous models [4, 8, 10]. Some of the important ideas in the work came from manufacturers [3, 6, 9, 12].

Examples show how adjusting valve parameters can solve reliability and efficiency problems.

CASE 1 - VALVE LIFT AFFECTS COMPRESSOR PERFORMANCE

Problem

A reciprocating compressor, during its first month of operation, had the first stage valves fail "several times" (75% of the time on the discharge side and 25% on the suction side), most often with broken outer rings (spalled patches in area of contact with seat).

The Unit

The compressor is a six throw two stage unit with variable volume pockets, compressing 0.6 specific gravity natural gas from 38 psig to 370 psig. It has three 19.5" bore cylinders in the first stage and three 13.75" bore cylinders in the second stage, with three suction and three discharge valves per end. It is driven by a 3,000 HP motor running at 885 rpm. The original multi-ring valves in the first stage had 0.150" lift.

The Simulation

Valve simulation, shown in Figure 1, indicates that the first stage valves, both suction and discharge, close prematurely, and then rebound and finally close about 20 degrees after dead center. The reverse gas flow after dead centre produces a violent impact, which leads to premature failure and therefore should be avoided. The simulation showed that stiffer springs had little effect in correcting the late closing, and might result in valve fluttering.

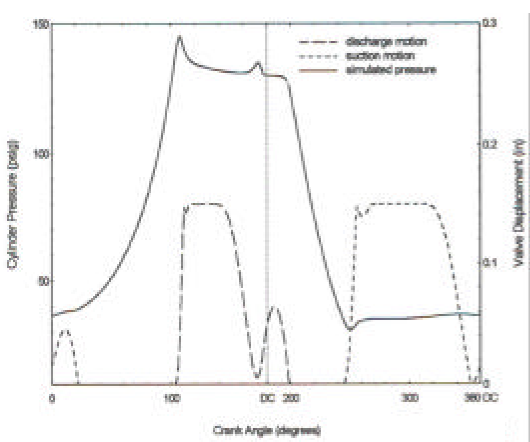


Fig. 1 Predicted Valve Motion and PT for 1C, STAGE 1

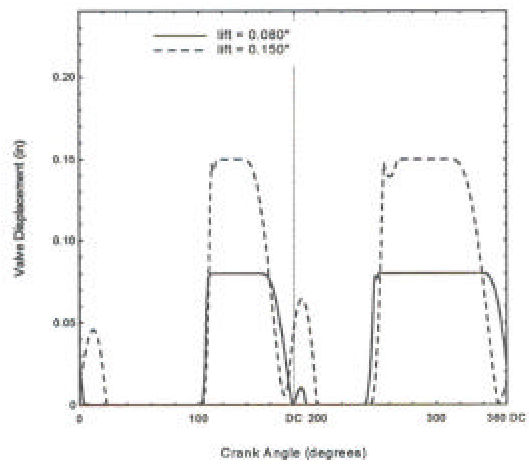


Fig. 2 Improved Valve Motions in Stage 1, Crank end

Reducing the lift, however, had a positive effect. Closure occurs around 10 degrees after dead center (DC) if the lift is reduced to 0.080".

Although reducing lift will cause higher valve loss, the simulation showed that compressor capacity would increase while efficiency (BHP/MMSCFD) stayed within an acceptable range. The capacity increased because back flow (gas reversing through the cylinder after dead centre) was much reduced.

Figure 3 shows that a lift of 0.080" is a good compromise between maximizing capacity and minimizing BHP/MMSCFD. Decreasing the lift produces only a small increase in BHP/MMSCFD but a significant increase in capacity. In this case the lift was reduced and no failures were reported in the following six months.

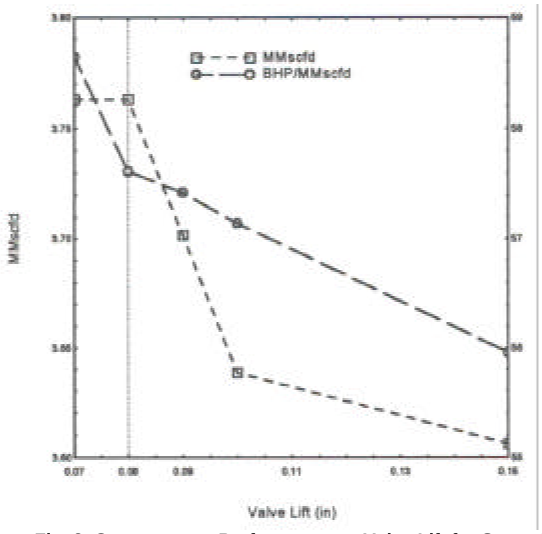


Fig. 3 Compressors Performance vs Valve Lift for Stage 1, Crank End

In addition to increasing compressor capacity, reducing lift will improve valve dynamics. Figures 4 and 5 indicate maximum plate impact velocities. For the discharge side, when lift was reduced from 0.150" to 0.080", impact velocity dropped by 52% on the valve seat and by 25% on the valve guard. Although the impact velocity on the seat is much less than on the guard, the seat contact area is also much smaller, so the stress is still significant.

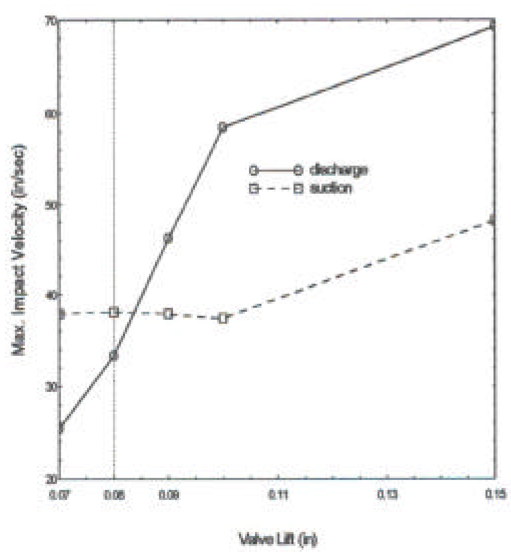


Fig. 4 Plate Max. Impact Velocity on Seat

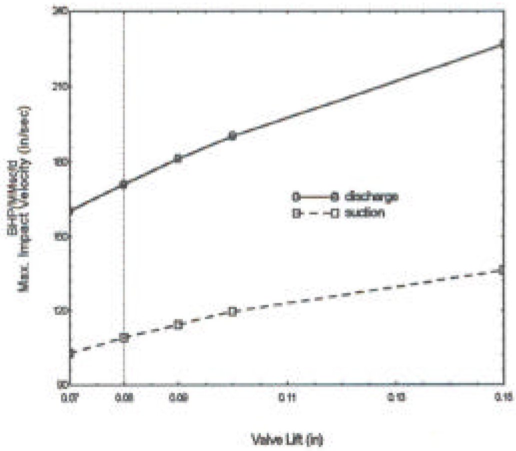


Fig. 5 Plate Max. Impact Velocity on Guard

The reduction in impact velocity means that reducing lift in this case will improve valve life. A compressor valve manufacturer [12] has reported that a valve should be reliable for about 10^9 cycles. That is, if a compressor runs at 885 RPM, its valves should last more than one year. Svenzon [11] showed that the number of cycles to failure will increase as the impact velocity decreases.

This case shows that wide variations in both valve life and performance are associated with variations in lift. Simulation helps identify an optimum balance among the various factors.

Economic Effect of Valve Improvement
This section shows the actual economic impact from increased production and reduced cost of parts and labour. Compressor horsepower and capacity are shown in Figure 6 versus first stage valve lift. It is assumed that the company can sell all the gas it can produce, that the cost of fuel gas equals the cost of sales gas, and that the driver has some excess horsepower. For simplicity, it is assumed that the lower heating value of the fuel is 1000 BTU/ft^3.

Ignoring factors which do not change with lift, such as wages, then

Revenue ($/yr) = [P*Q*(365 – t*f)]

and

expenses ($/yr) =
[24* P*C*Q*B*(365 – t*f)]/10^6 + Z*f

per end due to repairs and fuel

where
P = gas selling price ($)
Q = capacity (MMscfd)
t = days lost due to a valve change.
f = frequency of repairs per year
C = driver heat rate (BTU/[BHP-Hr])
Z = valve repair cost ($)
B = BHP/MMSCFD

For this example, lift is initially set at 0.0150”, and then is reduced to 0.080”. Assume
P = $1.30 per MSCF
C = 8333 BTU/BHP-Hr

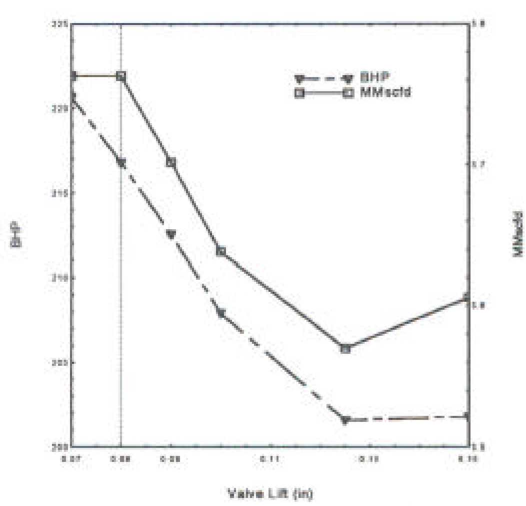


Fig. 6 Compressor Performance vs Valve Lift

Z = [$100(parts) + $100 (labour)]/each cylinder end
f = 12/year if lift is 0.150” and 1 if lift is 0.080”
t = 1/3 day
Q and B for the different lifts are shown in Table 1.

Table 1. Predicted Cylinder Performance

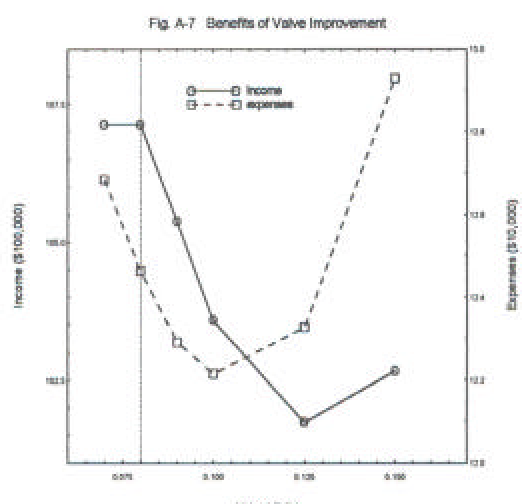
Lift	Q per end	B per end
0.150”	3.61	56.0
0.080”	3.76	57.6

Table 2. Valve Economics for 6 ends

Lift	Income	Expenses
0.150”	$10,165,038/ yr-stage	$128,248/ yr-stage
0.080”	$10,704,720/ yr-stage	$124,180/ yr-stage
Benefit	**$539,682**	**$4,067**

Table 2 and Figure 7 show revenue and expenses for two lifts, for six cylinder ends. Revenue is higher while operations and maintenance costs are slightly lower when lift is 0.080”. Under the circumstances, bottom line profit was increased by over $½ million per year through valve optimization – with a very low gas price assumption.

Fig. 7 Benefits of Valve Improvement



CASE 2 – VALVE LOSSES AFFECT VALVE LIFE

Problem

This study was undertaken to determine the cause of and solutions to premature valve failures in a compressor unit in which the valves had been failing approximately every 3 weeks for a period of 18 months. The owner wanted to evaluate the performance of valves with different designs, specifically a two port valve from one manufacturer and a four port valve from another.

The Unit

The compressor has four throws and three stages, and compresses gas from 170 psig to 920 psig. The third stage has one cylinder with 6.5" bore x 6.0" stroke running at 991 rpm and compressing natural gas of 0.7 specific gravity. Two 4.25" mass damping valves (four port type) were installed on the third stage head end suction and discharge during the measurement.

The Simulation

Figure 8 shows the measured and predicted head end pressure-volume (PV) diagrams for the third stage of the compressor. The measured curve is for the four port type; the predicted curves are for the four port type, the two port type, and a cylinder modified to hold extra valves.

Note that the measured and predicted curves for the four port type agree very well; minor differences are likely due to pulsation.

Figure 8 shows that the four port valve has less valve loss than the two port valve, and is therefore preferable. However, valve loss is excessive for both types.

The simulation found that the maximum pressure drop even across the four port valve was as high as 18% of absolute line pressure for suction, and 22% for discharge. Normal values are about 6%.

The conclusion: the valves failed due to the high valve losses and pressure drops, which cause very late valve closure and thus very high impacts. Reducing valve lift would not have solved this problem; it would have increased valve loss and therefore made valve performance worse.

Solutions

The recommendation was to increase the valve effective flow area. One approach is to install a different cylinder, designed to hold one or two extra suction and discharge valves for each end. Figure 8 shows that the average valve loss with two extra valves installed would be 7%, compared to the original average valve loss of 20% in the four port valve case.

Figure 9 compares the impact velocity for the different cases. Installing the extra valves, and therefore doubling the flow areas, makes a dramatic improvement.

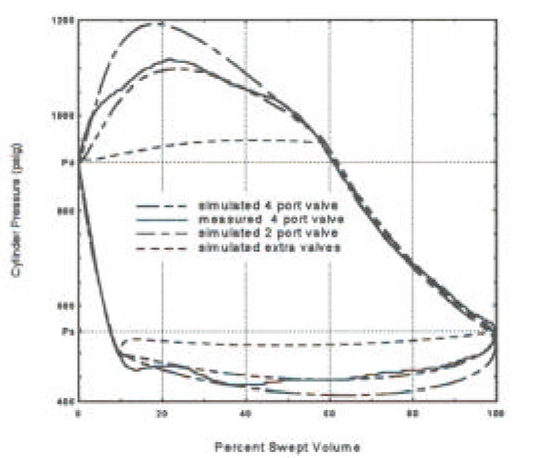


Fig. 8 PV diagrams in 3rd Stage, Head End

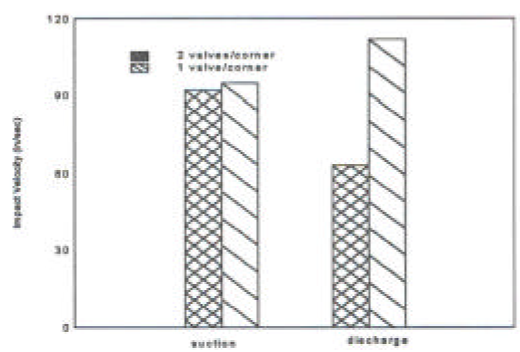


Fig. 9 Comparison of Impact on Valve Seat

CASE 3 – INSUFFICIENT CYLINDER FLOW AREA CONSTRICTS GAS FLOW

Problem

The compressor studied has experienced occasional valve failures in the past. Damaged valves were characterized by two or three broken poppets and sporadically broken poppet springs. The frequency of valve failures depended on flow conditions and compressor operating regimes.

The worst failures frequency was observed when certain valves were damaged almost monthly, requiring immediate repairs. Since the unit has 24 suction and 24 discharge double deck poppet valves, each with 30 plastic poppets, the potential for down-time was high. To prevent further failures, a valve study was launched.

The Unit

The compressor is a single stage, double-acting unit with variable volume pockets. It has four 17.5” cylinders and a 19” stroke. It is driven by a 6,000 HP integral engine running at 300 rpm and compressing 0.6 SG pipeline natural gas from 630 psig to 790 psig.

The Simulation

Figure 10 compares three PV diagrams in a head end: measured, simulated with valve pocket and simulated without pocket. The predicted PV curve is much closer to the measured curve if the simulation considers valve pocket loss (which is defined as the pressure drop between the valve and cylinder bore).

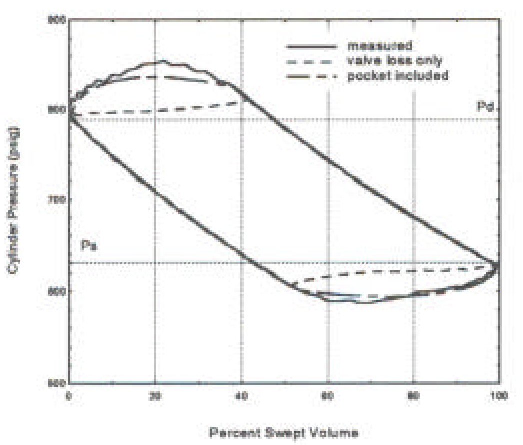


Fig. 10 Comparison of PV for 3H, Case 3

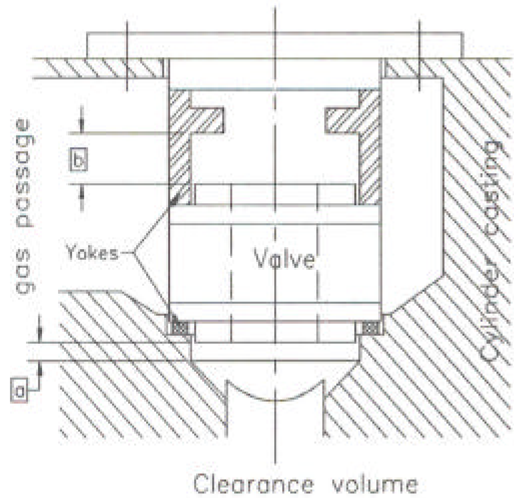


Fig. 11 Valve pocket detail

Cheema [2] has reported that in some cases the resistance offered to flow by the cylinder casting can be equal to the resistance offered by the valves. Constricted flow areas adjacent to a valve contribute a significant amount of pressure loss because they impose changes in velocity and direction.

Figure 11 indicates the installation of the valve in the compressor. Dimension a = 0.3” and b = 0.5”. These distances, which are influenced by the thickness of the valve yokes, are too small; the valve pocket loss factor was calculated to be between 1.9 and 2.4. Therefore, flow resistance and pressure loss are high [1].

The simulated valve motions shown in Figure 12 indicate that the valves flutter in the originally modelled operating condition. During a valve opening event, a good valve should be *fully* open more than 70% of its open period. If the forward gas force is not strong enough to push the plate against its guard, the valve will flutter. This is harmful to valve life because it increases the number of impacts. Valve flutter occurs when valve flow area is too large or springs are too stiff.

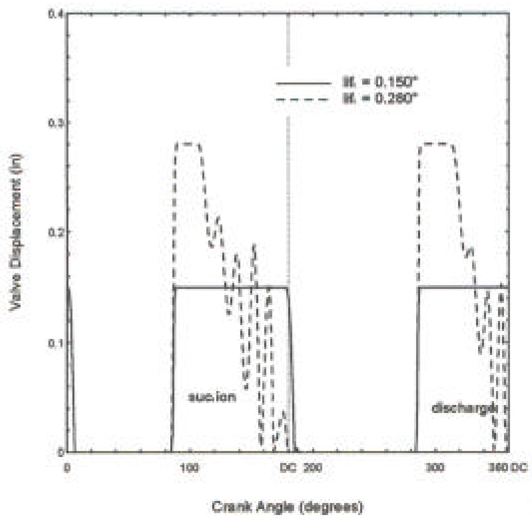


Fig. 12 Comparison of Valve Motions

The recommendations based on the above simulation results were:

1. Reduce valve lift from 0.28" to 0.15" to stop valve fluttering. This will lead to a reduced valve flow area, increased gas force on the poppet, and the poppet will be kept against its guard most of the time.
2. Increase yoke thickness so that the valve pocket flow area has less resistance. This will reduce the total horsepower consumed by the valve by about 18%
3. Pressure tap the valve caps so that future measurements will exclude the effect of pulsation.

Table 3 shows the total horsepower consumed by the original and modified valves.

Table 3 Comparison of Indicated HP–Cylinder 3

	Original HP	HP – Rec. 1 Reduced lift	HP – Rec. 2 Increased yoke
Head end	375	385	312
Crank End	478	496	387
Totals	**853**	**881**	**699**

Fuel cost can be calculated based on Table 3. The difference between cylinder horsepower with and without valve pocket loss is 154 hp for cylinder 3; for the four cylinder unit the total difference is 616 HP.

The cost of wasted fuel would easily exceed $100,000 per year, even at a low fuel cost of $2/MCF.

This case shows that cylinder flow area should be considered during valve modelling, especially if the model does not otherwise coincide with field measurements. Accurate valve modelling can be useful not only for valve trouble-shooting but also for saving fuel costs.

CONCLUSIONS

Efficient performance and reasonable valve life demand optimum valve design. Valves in a compressor can work *optimally* only in a small range of operating conditions (gas composition, temperature, suction and discharge pressure, etc.) If such conditions are changed, the valve flow area, springs and lift must be reconsidered in order to keep the compressor and valves working well.

A comparison method to choose valves is used by some manufacturers. They may assume that if a valve worked successfully in one case, then the same valve should work well in a similar case. This assumption is not necessarily accurate since valve performance is affected by so many variables.

This valve simulation accurately accounts for valve losses due to back flow, because it predicts real mass flows across valves (instead of assuming zero at dead centers). Therefore, compressor capacity, horsepower and rod load are more accurately predicted than traditional methods [5, 7].

Computerized valve performance analysis, in the hands of an experienced analyst, is a tool that enables precise specification of valves for any combination of conditions. A valve dynamics simulation can give valuable information about valve losses, valve opening and closing angles and impact velocities.

Ultimately, simulation can enable optimum designs for maximizing profit.

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