1990

# Performance Study of a Prototype Reciprocating Piston Compressor with Special Attention to Valve Design and Gas Pulsation

J. Kim
*Thomas Industries*

W. Soedel
*Purdue University*

# PERFORMANCE STUDY OF A PROTOTYPE RECIPROCATING PISTON COMPRESSOR WITH SPECIAL ATTENTION TO VALVE DESIGN AND GAS PULSATION

by

J. Kim, Thomas Industries, Sheboygan, WI 53082

and

W. Soedel, Ray W. Herrick Laboratory,
School of Mechanical Engineering
Purdue University
West Lafayette, IN 47907

## ABSTRACT

A theoretical case study is presented where various valve, discharge and suction cavity parameters of a prototype reciprocating compressor were varied to study the resulting efficiency changes of the compressor. The paper presents and illustrates one way simulation can be used for design improvements. For this particular prototype, it turned out that changes in effective valve flow areas had the most significant influence on performance efficiency. Other effects such as gas pulsations, valve mass, etc., are also discussed.

## PERFORMANCE EFFICIENCY

In the following, the performance efficiency will be defined as the ratio of actual specific work to ideal specific work

$$\eta_c = \left(\frac{W_{ci}}{m_{ci}}\right) / \left(\frac{W_{ca}}{m_{ca}}\right)$$

where $W_{ci}$ and $W_{ca}$ are the ideal and actual piston work per revolution and $m_{ci}$ and $m_{ca}$ are the ideal and actual mass delivered per cycle. This relationship may also be written as

$$\eta_c = \eta_{pv}\ \eta_m$$

where $\eta_{pv}$ is the indicator efficiency equal to the ratio of the actual to the ideal area of the indicator diagram, and where $\eta_m$ is the mass flow efficiency. In terms of coefficients of performance (COP) which exclude electrical and friction losses, we may interpret $\eta_c$ as

$$\eta_c = \frac{(COP)_a}{(COP)_i}$$

In air compressors, where cooling is considered to be free, the ideal piston work is isothermal, while in refrigeration the ideal piston work is often taken as isentropic. However, any kind of indicator diagram can be taken as ideal, in principle. In this study, where the focus is on valve design and gas pulsation losses, it was decided to take the actual measured indicator diagram minus valve and gas pulsation losses as ideal, on the argument that deviations from isentropic compression and expansion, and leakage effects cannot be influenced by changing the valve and manifold designs alone.

## PARAMETER STUDY

The parameter study is based on a computer simulation of a fractional horsepower refrigeration compressor [1,2,3]. Certain parameters such as valve damping, cylinder heat transfer, and so on were identified by comparing the results of the simulation to prototype measurements.

Having achieved satisfactory agreement between simulation and prototype measurements at various operating conditions, it was argued that the simulation could be used to study the sensitivity of certain valve and manifold design parameters on the performance efficiency. It was decided to study only single parameter variations. One design variable was changed at a time, with intentionally large variations to see its effect more clearly. Each variation was explored in two directions. For example, starting with the prototype suction valve porting, the porting was made larger (favorable direction of parameter change) and also smaller (adverse direction). Table 1 summarizes a typical study, for a suction line pressure of 10 psig, a suction cavity temperature of 120°F, a compression ratio of 18, and a polytropic coefficient of 1.13, for refrigerant R12. In series 1, the suction and discharge cavities where reduced to a quarter of the original volumes, the effective suction and discharge flow areas were reduced to 70% of the original areas, and the suction and discharge valve response was made more sluggish by increasing the modal mass such that the natural frequency was reduced by 50%. In series 2, the suction and discharge cavity volumes were made four times larger than the original volumes, effective flow areas were increased by 30% and the modal masses of the valves were decreased such that the natural frequencies increased by 50%. Table 2 gives the results of this study. Figs. 1-3 show these results in graphical form. It is interesting to note that case 3 resulted in reversal of the original intention, namely to make series 1 represent negative change and series 2 positive change. As it turns out, the performance efficiency was actually improved by making the discharge cavity smaller as compared to making it larger, even while the indicated efficiency followed the original concept. The reason is that the dynamic interaction between the discharge valve and the discharge system produced a larger back flow from the larger cavity which resulted in a reduced mass flow efficiency. This is a very good example of the value of simulation, since engineering intuition is often mislead when judging complex system interactions.

It is also instructive to examine the indicator diagrams for some of the cases . The influence of the reduced and enlarged discharge cavity (case 3) is shown in Fig. 4. Because the effective discharge valve flow area restriction dominates the indicator diagram, it is not very much affected by the very large increase in cavity volume, even though the gas pulsation is effectively eliminated. Fig. 5 shows the effect of changes in effective suction valve flow area (case 4). When this flow area is restricted, considerable backflow occurs resulting in a large drop in $\eta_m$ and as a consequence $\eta_c$. The effect of changes in discharge valve flow area are shown in Fig. 6 (case 5), which shows clearly that the prototype compressor responds better to effective flow area enlargements than reductions in gas pulsations.

The individual losses for all cases are summerized for the series 1 changes in Fig. 7 and for the series 2 changes in Fig. 8.

For all other cases, the performance efficiencies turned out as intuitively predicted. For the prototype compressor under study, it was found that the effective suction and discharge valve flow areas were the most sensitive parameters.

## CONCLUSIONS

This paper demonstrates one possible way to do a performance identification and parameter sensitivity study for reciprocating compressors. It identifies the parameters which are the best candidates for positive (or negative) change. In the compressor under study, they were the effective suction and discharge valve flow areas.

Note that the numerical results of this study cannot easily be generalized since (a) the compressor was a research prototype, not a production model, and (b) depending on the base line design, other parameters, such as suction and discharge cavity volumes governing back pressures caused by gas pulsations may be dominant.

## ACKNOWLEDGEMENT

Support by Necchi S.p.A., for related work is greatly acknowledged.

## REFERENCES

1. W. Soedel, "Introduction to Computer Simulation of Positive Displacement Type Compressors", Short Course Text, Purdue University, 1974.
2. W. Soedel, "Gas Pulsations on Compressor and Engine Manifolds", Short Course Text, Purdue University, 1978.
3. J. Kim and W. Soedel, "Convergence of Gas Pulsation Simulations when Combining Time and Frequency Domains Iteratively", Proceedings of the 1990 International Compressor Engineering Conference, July 1990, Purdue University, West Lafayette, USA.


| case | variations | series 1 | series | major concern |
|---|---|---|---|---|
| 1 | none | original design | | Reference |
| 2 | suction cavity volume | 0.25(*1) | 4 | effect of gas pulsations |
| 3 | discharge cavity volume | 0.25 | 4 | |
| 4 | suction port area | 0.7 | 1.3 | effect of valve restrictions |
| 5 | discharge port area | 0.7 | 1.3 | |
| 6 | suction valve response (*2) | 0.5 | 1.5 | effect of valve response speed |
| 7 | discharge valve response | 0.5 | 1.5 | |

(*1) Means 0.25 times of the original value.

(*2) Natural frequency of the valve was changed such that a change of modal mass resulted.

Table 1 Parameter Changes Under Study

| case | $\eta_{pv}$ | | $\eta_m$ | | $\eta_c$ | |
|---|---|---|---|---|---|---|
| | series 1 | series 2 | series 1 | series 2 | series 1 | series 2 |
| 1 | 79.58 | | 98.51 | | 78.39 | |
| 2 | 79.01 | 80.04 | 97.10 | 76.72 | 78.31 | |
| 3 | 78.96 | 79.90 | 100.52 | 97.43 | 79.37 | 77.85 |
| 4 | 77.27 | 80.96 | 86.62 | 103.34 | 66.93 | 83.66 |
| 5 | 76.52 | 84.77 | 97.57 | 98.17 | 74.66 | 83.22 |
| 6 | 78.75 | 79.79 | 95.29 | 98.91 | 75.04 | 78.92 |
| 7 | 78.83 | 79.78 | 97.23 | 98.37 | 76.65 | 78.48 |

Table 2 Results of Changes Listed in Table 1.

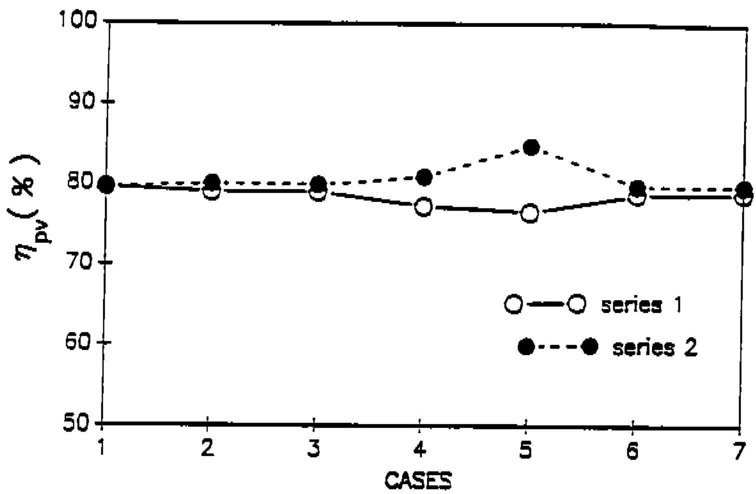


Fig. 1. Indicator Efficiencies $\eta_{pv}$.

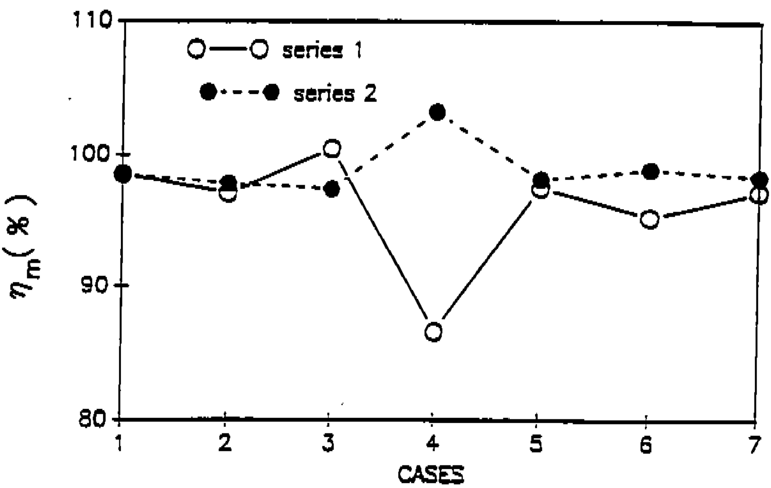


Fig. 2. Mass Flow Efficiencies $\eta_m$.

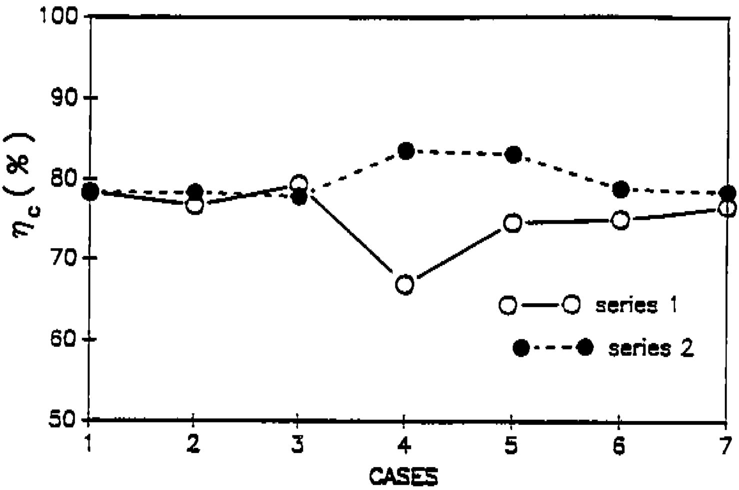


Fig. 3. Performance Efficiencies $\eta_c$.

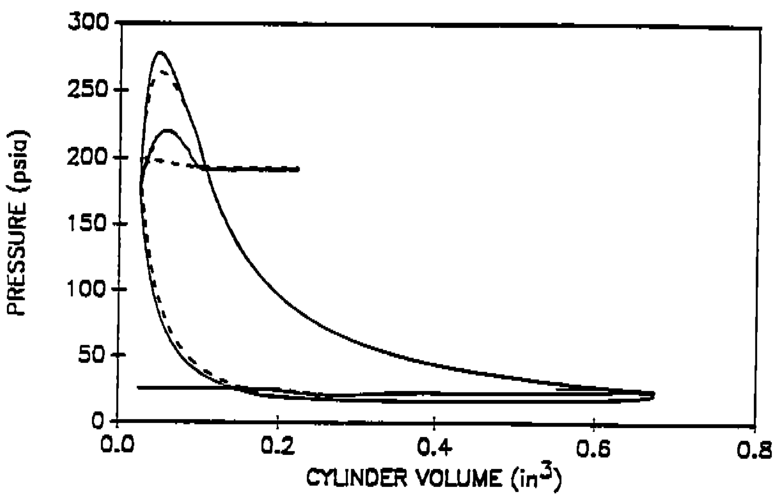


Fig. 4. Indicator Diagram for Case 3, ——, Series 1, - - - -, Series 2.

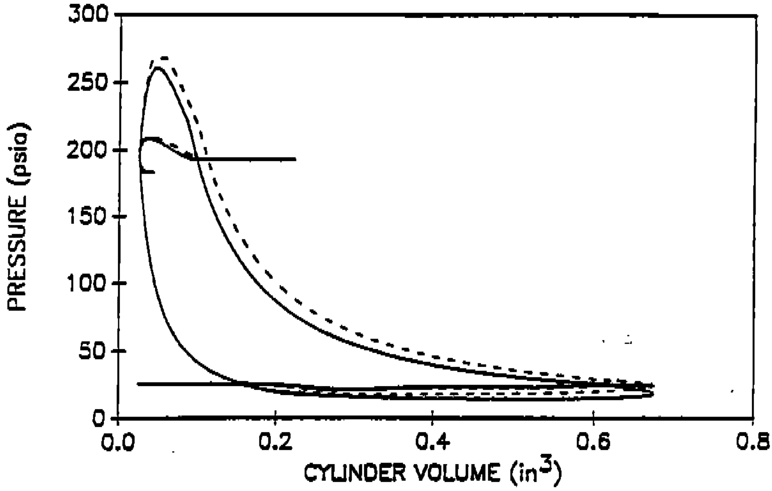


Fig. 5. Indicator Diagram for Case 4, ——, Series 1, – – – –, Series 2.

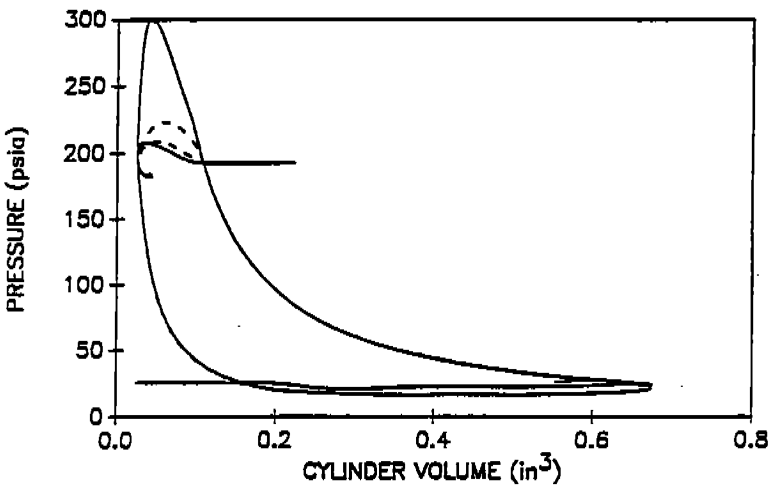


Fig. 6. Indicator Diagram for Case 5, ——, Series 1, – – – –, Series 2.

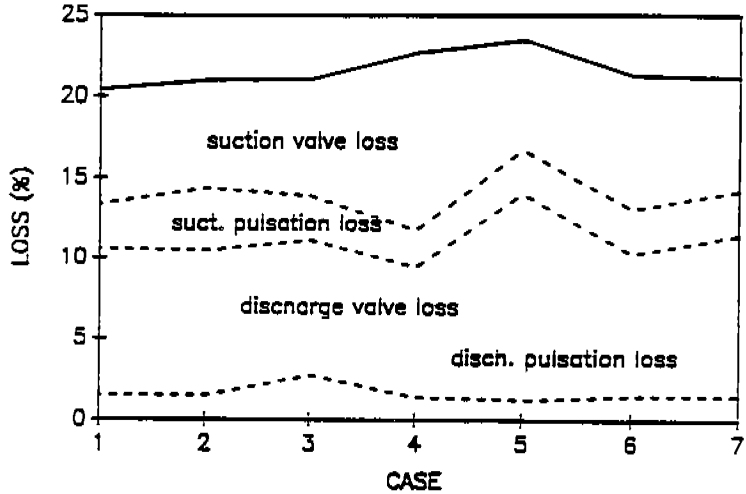


Fig. 7. Summary of Component Losses for Series 1 Changes.

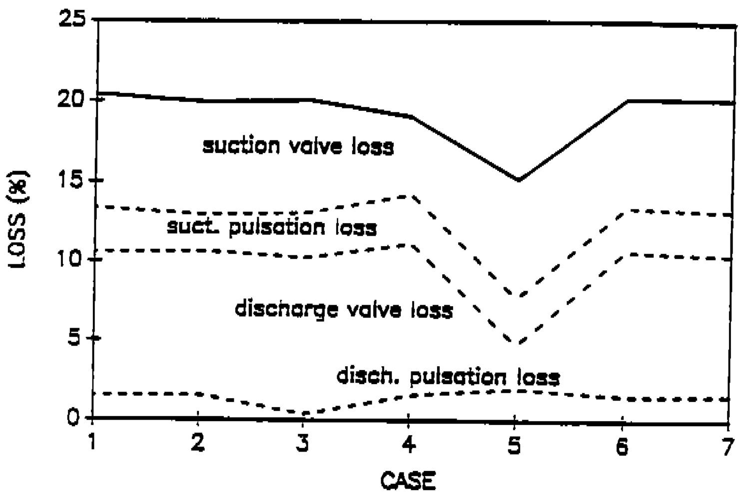


Fig. 8. Summary of Component Losses for Series 2 Changes.