Valve Stress Analysis- For Fatigue Problems

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INTRODUCTION

In recent years, the compressor industry reduced the size of their compressors without an attendant loss in capacity by increasing compressor speed. However, increased speed brought on a whole host of dynamics problems. The dynamics of self-operating valves, for example, became very complicated and presents problems even today.

The compressor in a refrigeration system can be likened to the heart in a human. The similarity is even greater when valves are considered in the two systems. Just as with the valves in a heart, failure of the valves in a compressor can cause loss of life. Poor operation (that is, improper timing when they open and close) strongly influences performance. In a compressor system, pulsating flow delivered by the compressor strongly determines the amount of noise from the system. Pulsating flow is strongly influenced by the valves.

Thus the valves greatly affect compressor life, performance and noise. This paper is concerned with only one life that is, valve failures.

Valves systems can fail in a number of different ways. I believe they can be categorized as:

1. Plastic flow or erosion of the valve, valve seat, or supporting members so that back flow occurs from improper seating. See, for example, reference [6].

2. Impact failure due to high velocity striking of the valve on the valve seat. See, for example, references [2,3,4,14,23]

3. Fatigue failure of the valve.

All three modes of failure are important and are difficult to predict by analytical or experimental means. This paper will be concerned primarily with fatigue failures.

BASIC FATIGUE ENDURANCE CONCEPTS

For a given set of environmental conditions the fatigue capacity of a part is governed by the strength of the material and the maximum tensile stress experienced (usually developed at the discontinuities). Material discontinuities may be macroscopic in size due to design geometry, or they may be microscopic in size due to surface scratches, dislocations, etc. The peak stress developed at the discontinuity is a function of the size, location, and orientation and configuration of the discontinuity.

A designer confronted with a fatigue problem should select the material with the best inherent fatigue resistant properties. Further he should select the material processing such that this strength level is maximized and stress raisers are minimized. A particular case history, from the metallurgical and materials processing point of view, is reported in reference [18]. See also [20,21].

ACCELERATED LIFE TESTS

Until some ten years ago, valve designers had to be content with these basic metallurgical principles, and static design principles (see for example [15,19,25]). Their designs were backed up by "accelerated life tests". These accelerated life tests were necessary since, in many cases, compressors were guaranteed to operate for as much as five years of continuous operation at 60 cycles per second.

* Numbers in brackets refer to the selected but not exhaustive list of references.

** They should more properly be called accelerated failure tests.

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Accelerated life tests can be done either in situ; that is, inside an operating compressor or in special valve fixtures. When done in situ, a choice must be made between operating at normal or higher speeds. At normal speeds, accelerated life is achieved by increasing valve stress by changing the load on the compressor. As shown in Ukrainetz [22], it is extremely important to know the relationship between load on the compressor and valve stresses. More than one compressor designer has been known (years ago) to believe that load conditions which accelerated the life of the electric motor also accelerated the life of the valves. The advent of the strain gage showed how wrong this can be. Strain gages made it possible to have better correlation between results of this type of accelerated life test with field experience. However, Gluck [9] showed just how difficult it is, even with good strain gage measurements, to actually predict field life.

The use of higher running speeds to accelerate life is very difficult. First, compressor components are not designed for higher than normal speeds. Therefore failure of other components in the compressor can be expected along with or prior to valve failure. For example, severe motor problems would occur if the speed were increased significantly, but even if a different drive were devised to bypass that or other component failure problems, the effect of changing the fundamental frequency of the valve opening and closing and its effect on valve stresses would be questionable. The answer to that question is not known. For these reasons, higher speed in situ accelerated life tests are not used.

Special valve fixtures have also been commonly used for determining valve acceptance. A useful impulsive loading system is shown in Figure 1. In this system the valve is mounted on a valve plate in the normal manner. The valve plate is located close to a rotating disc which has a series of holes close to the circumference of the disc. An air supply behind the disc supplies high pressure air so that as the holes in the disc line up with the valve port(s), a pulse of air is delivered to the valve. Control of the speed and number of holes in the disc determines the frequency and time duration of the pulses. The pressure determines the magnitude of the pulse on the valve. It is important that the components touching the valve simulate the geometry inside the compressor as much as possible.

With this test, as with the in situ tests, most manufacturers have been reasonably successful determining good and bad valves and designs. Again, however, prediction of actual life in the field based on this test is apparently not possible.

The use of electrodynamic vibrators to shake valve systems for accelerated life tests of valve systems has been used little if at all in the compressor field. I believe they hold much promise to obtain data which can be used for life prediction provided that the frequency content is monitored and related properly to real life conditions. The aerospace industry has been able to use such equipment to good advantage. For example, they determine a spectral density envelope for strain as a function of frequency for the particular part being tested. Figure 2 shows a power spectrum of strain as a function of frequency. The various curves are representative for various operating conditions. The envelope is a "safe" condition for the test. The assumption of a particular fatigue law then allows the development of a relationship between the time for the part to be on the electrodynamic shaker as a function of spectral density level. This enables testing for a particular life in the field. Now that the strain gage is available to us, we should be able to do something similar. Implementation, of course, would require a good research program.

MEASUREMENT OF "REAL-LIFE" STRAIN

The use of foil or wire resistive strain gages on compressor valves has already been mentioned. First to report the use of strain gages on small hermetic compressors was Lowery [12], followed quickly by Ukrainetz and Gluck [10]. Others have improved upon the techniques that they developed. Today most of the problems in using strain gages on valves in compressors have been solved. However, an engineer is always faced with the question of where is the proper location for the gage (i.e. the point of maximum strain). The second problem is that of upsetting the environment of the valve with the strain gage installation. Although Lowery showed that miniature strain gages do not themselves affect the dynamics of the valve significantly, they and their wiring do require space. Provision may affect the fluid flow and therefore the dynamic forces on the valve. For these reasons, each different compressor requires the development of its own strain gage installation. Strain gages are now widely used by compressor manufacturers.

The Japanese have developed a copper plating method for measurement of stress distribution in valves in situ. See, for example, Kawahira's paper [11]. I am unaware of any American manufacturer using this procedure. This procedure involves the inspection of the copper plating on the valve after it has been used in the
compressor. Density of spots on the plat­
ing is then related to valve stress.

Payne [16] records the use of photostress
technique for determining the best location
for strain gages. A thin coating of bire-
fringent plastic is placed on the valve for
viewing with a polariscope for strain pat-
terns. His work was done with a steady
air blowing on the valve through the ports.
Therefore, the valve was held in a static
condition. When locations of the
maximum static strain are the same as the
locations of the maximum dynamic strain
this method works very well to locate
points of maximum strain. Some manufac-
turers use this procedure or variations of
it.

Typical strain versus time measurements for
a cantilever type suction leaf valve are
shown in Figures 3 and 4. Figure 3 shows
strain very close to the support. Figure 4
shows strain very close to the port or
free end. Three points are indicated in
these pictures. The first shows when the
valve first opens. The second shows when
the valve hits the stop and the third when
the valve closes. Figures 5 and 6 are
typical strain time histories of a flexing
suction ring valve in the circumferential
direction. The two figures show strain at
different circumferential locations.

INTERPRETATION OF STRAIN MEASUREMENTS

Figures 3 through 6 indicate how complex
the strain history is on valves. A signifi-
cant question is how to interpret these
measurements. They cannot replace the
accelerated life test completely since
they only indicate the macroscopic strain.
However, they do that very well, making it
possible to relate macroscopic strain to
design geometries and compressor loads.
Currently designers use these measurements
in two ways. First, they compare peak
values with measurements made on compres-
sors for which field life experiences are
known. This comparison gives them a
"go-no go" test.

The second way that designers use strain
measurements is by recognizing features of
strain time records. This is especially
useful when coupled with measurements of
pressure versus time and valve displac-
ment versus time. Often a designer can
determine poor or good design by comparing
these three measurements with each other
and with similar measurements for other
compressors.

I suggest strain gages also be used on
valves in the impulsive accelerated life
test fixtures. Thus strain on this test
can be correlated with other measurements
and experience. Obviously strain gages
will also be needed with any use of
electrodynamic shakers.

But even if these experimental techniques
using strain gages and accelerated life
tests become well developed, strain mea-
surements are difficult to relate to de-
sign parameters such as port, valve and
valve support geometry. The measurements
only permit a comparison of designs with
each other or with an acceptable standard.
Thus, knowledge of strain magnitudes may
indicate a poor design, but do not in them-
selves indicate what a new design should be
for improved life. To do that we must
marry measurements to an analytical
approach.

ANALYTICAL APPROACHES

Davis [5] reports an approach based on (1)
field experience with many different
valves in mainly large compressors, and (2)
a simple theory which places special im-
portance on the free natural period of the
valve and the crank angles corresponding
to when the valve makes contact and leaves
contact with its stop or retainer. These
angles are indicated in Figure 7 as crank
degrees before end of piston stroke. \( \theta_1 \)
corresponds to the instant when the valve
would have had to leave the stop if it
were to close exactly at the end of stroke
in the absence of gas forces (i.e., one
quarter of its natural period). \( \theta_2 \) is de-
termined by the instant that the gas forces
on the valve exactly equal the valve spring
force when the valve is against its stop.
\( \theta_3 \) corresponds to when the valve becomes
fully open against its stop. The criterion
for proper design is reported as \( \frac{\theta_2}{\theta_1} > 2 \)
and \( \frac{\theta_2}{\theta_3} < 0.7 \). Simple equations are de-
veloped to predict \( \theta_1, \theta_2, \) and \( \theta_3 \).

Creswick [4], using similar equations esti-
mates impact velocities between the valve
and the valve plate, or valve stop. Un-
doubtedly, for those valves for which im-
pact conditions are critical, this approach
is very important. We find, however, that
the estimating equations are not accurate
for all compressors.

Wambsganss [24] and others following used
more complicated equations simulating the
compressor and valve dynamics as a func-
tion of time on modern computers. The
general procedure followed the well known
work of Castigliola, but equations using
more than one vibrational mode of the
valve were used. Wambsganss' simulation
program predicted valve motion, thero-
dynamic variables and compressor performance
reasonably well. His was a semi-analyti-
cal procedure in that certain parameters
of the compressor and the valve had to be
determined from experiment. For example, the modal damping for the valves is determined empirically and, for valves which are complex in shape, natural frequencies are also determined experimentally. A major problem that faced Wambsganss was that of calculating the force on the valve. This required experimental determination of orifice coefficients and valve drag (force) coefficients as a function of plenum and cylinder pressures and valve displacements. For a general discussion of the compressor simulation problem see Qvale, et al. [17].

Many improvements in mathematical model and simulation programs have been made by others. I am aware of three investigators who attempted to predict valve stresses using Wambsganss' mathematical model. Doige [8] first tried to calculate the force distribution on the valve as a function of time without the use of empirically determined force (drag) coefficients. He was not successful. Then Doige [7,8] and later Moaveni [13] considered bypassing the detailed, unavailable description of the force on the valve. They substituted estimates of the energy imparted to the valve from the time of opening until stop contact and position of maximum strain. Equations developed by them made it possible to predict stresses as a function of certain valve design parameters. Their procedure however requires that a prototype compressor be tested with strain gages on the valve. Measurements coupled with analysis then enable the estimation of the amount of energy imparted to several modes of vibration of the valve. Since the analytical expressions used are functions of certain design parameters of the valve assembly, prediction of strain for changes in these design parameters from the prototype valves is possible. For example, Figure 8 indicates how strain at contact with the stop and the peak value of strain can be predicted as a function of stop depth for a cantilever type suction valve on a small compressor.

Adams [1] extended the simulation program of Wambsganss (and others) to the prediction of strain in bending ring valves. His contribution was primarily in the use of strain modes as measured in a resonant test. This enabled getting around the inaccuracies of differentiation of displacement modes for strain. Like Wambsganss [24] and Payne [16] he used drag coefficients for gas pressure force estimations on the valve. Participation of modes is calculated in a similar manner as Payne. His results also showed the need to use higher modes of vibration to predict strain. Using only one mode prior to valve contact with the stop and then using two modes he was able to achieve the correlation with measurements shown in Figure 9.

Current research work, using finite differences and finite elements is rivaling the strain mode summation method. This seems to have some advantages for those cases where a stop with a complex shape is used.

As we develop better ways to determine the gas forcing function on the valve and the strain modes as a function of the design geometry, we shall do better predicting valve strains. However, the current need of drag coefficients appears likely to continue to require experimental determinations for adequate accuracy. A new approach to the forcing function is needed to drop the experiment for drag coefficients. Thus a purely analytical prediction of strain is not yet possible. In the meantime, we see that accurate strain predictions are still quite difficult (requiring more research) using the simulation technique of Wambsganss (or others).

Also in the meantime, the approach taken by Doige and Moaveni should be considered for use by designers in their search for optimum designs. This procedure does not attempt a time history prediction of strain and therefore bypasses some of the simulation problems. Instead, it utilizes measured strain time histories on a prototype compressor. Since this measurement has become almost universally done for most new compressors, the data is already available. The effect of small changes in certain valve assembly parameters on strain can then be predicted by the analytical procedure.

**SUMMARY AND CONCLUSIONS**

1. Fatigue failures of compressor valves may be due to microscopic material or machining stress raisers or due to large dynamic stresses caused by excessive load or poor geometric design.

2. It is not likely that the accelerated life test for valves will be replaced soon by either quick laboratory measurements of strain or analytical strain predictions.

3. The use of strain gages in accelerated life tests can make accelerated life tests much more quantitative and thus better able to correlate with field experience.

4. Research should be started to improve accelerated life tests (perhaps using electrodynamic shakers) to the point that statistical life in the field can be predicted.

5. Field measurements of strain (eg: copper plating and photostress) can be
utilized to help locate strain gages.

6. With more research it appears that the strain mode approach (or finite element analysis) coupled to a computer simulation program can be made to predict strain as a function of design parameters. However, the approach seems unlikely to become purely analytical in the foreseeable future.

7. The use of measured strain time histories for estimates of the energy imparted to the valve coupled to modal analysis shows promise as a predictor of strain for certain valve design changes.

REFERENCES


Fig. 5 Circumferential Strain, Flexing Ring-Type Valve, 45° Location [1]

Fig. 6 Circumferential Strain, Flexing Ring-Type Valve, 75° Location [1]

Figure 7. $\Delta p$ and Valve Motion Illustrating $\theta_1, \theta_2, \theta_3$: [5]

Figure 8. Theoretical and Measured Strain Vs Stop Depth. Cantilever Type Leaf Valve. [13]

Fig. 1 Tangential Strain for Flexing Ring-Type Valve, 75° Location, [1]