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Some Considerations on the Structural Reliability of Compressor Valves

J. S. Laub
Carrier Corporation

SOME CONSIDERATIONS ON THE STRUCTURAL RELIABILITY OF COMPRESSOR VALVES

James S. Laub, Senior Engineer
Carrier Corporation, Research Division
Syracuse, New York, USA

1 INTRODUCTION

Refrigeration compressor valve life and its description in terms of reliability are complex functions of material defects and machine operating characteristics. The following paper discusses some aspects of valve life that are determined by material properties and conditions. Detailed studies of several aspects of the material performance of compressor valves have led to description of structural performance by a variety of methods, most of which are in common use, but with some statistical refinements which allow better selection of stress levels and materials by the designer and compressor engineer.

Surface and volume material defects are considered as stress concentrations. These produce a distribution of defect strengths throughout the valve and are the primary controllers of valve structural reliability. Less notch-sensitive materials give better performance, and notch fatigue testing provides information on this material parameter.

Defect strengths determine certain failure distribution parameters. Conventional high carbon valve steels possess a fatigue limit which gives a cut-off point in the cumulative failure distribution when stress levels are near the fatigue strength. The failure distribution itself gives an indication of the defect strength range to be encountered in any given instance. The proportion of valves surviving any given test quantifies the probability of encountering a fatal defect.

Basic material, valve fabrication techniques, handling and assembly all influence the failure rate of valves. The following discussion is limited to considerations involving the basic material fatigue performance. By using the statistical distribution idea, many hitherto mysterious effects in valve reliability can be understood. The statistical distribution of defect strengths and their concurrent spatial distribution of defects through the material provide a basis for designing and evaluating valve characteristics by rational engineering principles.

2 VALVE DEFECTS - STRESS CONCENTRATION FACTORS

Surface Defects

For shallow surface notches such as are often encountered on compressor valves, the load configuration is one of either bending or a combination of bending and torsion. Usually the torsional loading component is small because of the location of the critically stressed area even on curved spoke valves, so only the bending case need be considered. From the derivation given by Neuber in [1], it is found that the depth of the notch, t, and the radius at its root, ρ, are the determining parameters; component thickness has no effect.

The stress concentration factor, K_t, for single surface scratches perpendicular to the principal tensile stress, $\sigma_{nominal}$, is:

$$K_t = \frac{\sigma_{max}}{\sigma_{nom}} = 3\sqrt{\frac{t}{2\rho} - 1} + \frac{4}{\sqrt{\frac{t}{2\rho} + 2}} \qquad (1)$$

Values of K_t for various t/ρ ratios are shown in Figure 1. These may be used for single scratches, dents, and pits such as occur on valves which have been damaged by handling. Typical values for surface scratches found in valves made up by tool room methods are as follows: $t = 0.2 \times 10^{-3}$ in., $\rho = 0.34 \times 10^{-3}$ in., $K_t = 2.2$. The notch geometry was obtained using 500X photography on a prototype valve.

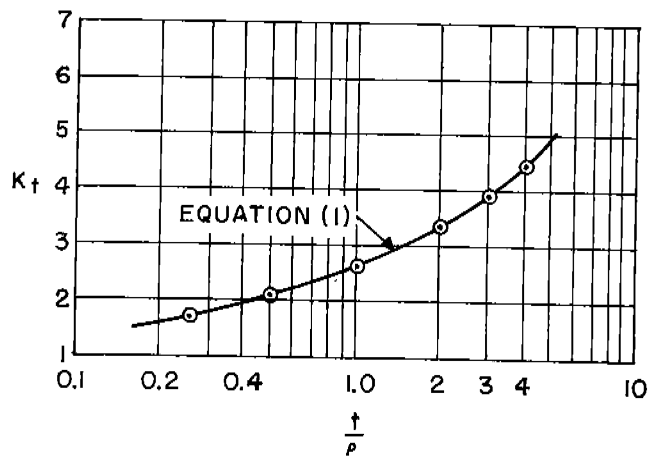

FIGURE 1

Elastic Stress Concentration Factor for Single Surface Scratches

It is readily apparent why this form of damage must be avoided anywhere near the critically stressed area of a valve. Note that ρ has a lower bound roughly equal to 1/2 the grain diameter, d. This is because of the noncontinuous nature of real technical materials. Notches cause slip (yield) across the entire grain into which they intrude so that at the single crystal yield load, the notch is immediately blunted to the configuration of the boundary of the grain. This is one of the reasons fine-grained, homogeneous materials tend to be more notch sensitive than coarse-grained materials.

Commercial valves have specially finished, sometimes peened surfaces that are covered with a random network of shallow, blunt-rooted notches which have the aggregate effect of relieving each other. The problem of interacting notches is also solved in [1], where a modified notch depth, t_w, is defined as a function of the spacing, b, between the repeating notches.

$$t_w = t \cdot \gamma \qquad (2)$$

γ is called the coefficient of load relief and is:

$$\gamma = \frac{b}{\pi t} \; \mathrm{Tanh} \, \frac{\pi t}{b}$$

The stress concentration factor is then:

$$K_t = 1 + 2\sqrt{\frac{t_w}{\rho}} \qquad (3)$$

Figure 2 represents graphs for both γ and the elastic stress concentration factor for repeated notches. The spacing is the distance between nearly parallel scratches or notches aligned normal to the direction of principal tensile stress.

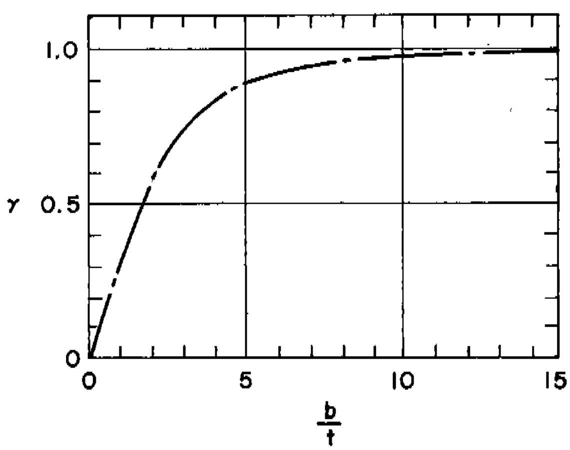

FIGURE 2a

The Load Relieving Factor, γ, as a Function of the "Notch Proximity," b/t

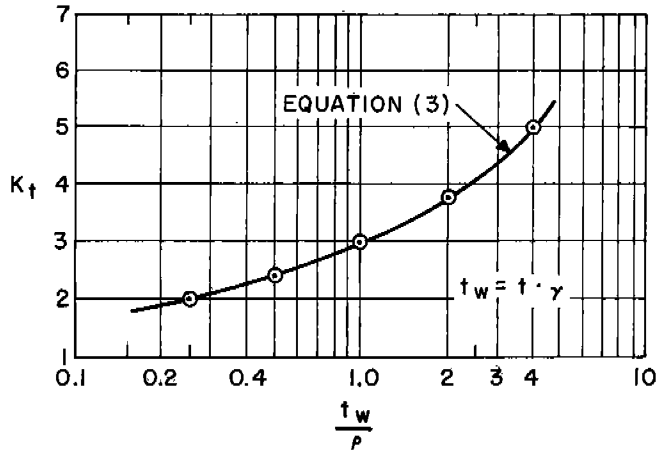

FIGURE 2b

Elastic Stress Concentration Factor for Multiple Surface Scratches

Volume Defects

Volume defects such as voids, second-phase particles, non-metallic inclusions, and microcracks must be treated as three-dimensional problems; consequently, the simplifying assumption of plane strain loading conditions is most often invoked. Voids or hollow cavities will not be treated here since they should never occur in the wrought materials of interest for compressor valves. Hard, second-phase particles are of interest since high carbon steels frequently contain iron carbide, Fe_3C, precipitated as a second phase in the ferrite (α - Fe) matrix.

With this class of problem, approximate solutions are generally the rule, although a few special cases where symmetry prevails have been solved exactly in closed form. Chu and Conway [2] give a series of solutions for spherical and cylindrical inclusions starting with a perfectly rigid sphere; i.e. $E_i = \infty$, for the inclusion. For this case, the maximum stress occurs on the particle axis in line with the maximum tensile stress and corresponds to a K_t of 1.94. Because of symmetry and the assumptions of continuum mechanics, no size effects or relative dimensions are necessary in this solution. Obviously in real materials, this solution becomes less accurate as the inclusion size approaches that of the surrounding matrix structure. Other solutions given are for rigid cylindrical inclusions of various aspect ratios. The effect of shoulder radius is also considered. Finally, the elastic solution for spheres and cylinders is given as a function of the modulus of elasticity ratio, E_i/E_e, where E_i is the modulus of the particle and E_e is the modulus of the surrounding material matrix.

The numerical method used provides any needed degree of accuracy, depending on the number of points selected on the matrix-particle boundary. It should be appreciated that the problem is solved only for axisymmetric inclusions aligned with the direction of principal tensile stress. These results indicate that for the most part, hard particles represent stress concentration factors between 1.0 and 2.0.

These solutions are presented in continuum mechanics terms, implying particles large with respect to the matrix (grain), and do not include a lower limit on the particle corner radius. Angular inclusions with sharp radii will cause higher K_t's.

When the particle is small in comparison to grain dimensions, we can assume that continuum mechanics concepts again prevail in that the surrounding matrix, now the grain material with its crystal lattice characteristics and atomic structure, is homogeneous. Naturally, the material is no longer isotropic; the lattice parameters, dislocations, molecular and atomic forces, etc., determine the strength and resistance to slip of the surrounding medium.

Samples of AISI 6150 steel heat treated to 231 ksi tensile strength were found to have many sharp, angular carbide particles of roughly one micron in diameter in a grain of d = 25 microns. The stress concentrations at corners were calculated to be between 1.4 and 1.7. This material was more notch sensitive than one of the conventional AISI 1095 plain carbon steels heat treated to 260 ksi tensile strength. The latter material was also found to have carbides dispersed through the grain matrix, but these were much less angular and uniformly smooth when viewed by scanning electron microscope.

Microcracks

An analysis of the effect of microcracks, i.e., cracks on the order of length of grain dimensions in real materials, cannot be based on classical continuum mechanics because of the relative sizes of the effects involved. Here we have notches or disturbances which are strongly affected by lattice arrangement and are frequently interrupted by grain boundaries or carbide particles. The root radius of the crack tip ranges upward from half the atomic spacing, $\bar{b}/2$, which is on the order of angstroms for brittle or elastic cracks. The assumptions of homogeneity and isotropy are no longer applicable. Sharp crack fracture mechanics theory provides some of the necessary concepts to evaluate this problem.

The equation for the stress concentration caused by completely brittle elliptical cracks or voids is:

$$K_t = \frac{\sigma_{max}}{\sigma_{nom}} = 1 + 2\sqrt{\frac{a}{\rho}} \qquad (4)$$

where a is half the length of the crack (the semi-major axis) and ρ is the radius at the tip. For very shallow ellipses approaching the shape of a crack:

$$K_t \simeq 2\sqrt{\frac{a}{\rho}} \quad \text{for } \rho \ll a \qquad (5)$$

In order for this equation to be useful for other than idealized, completely brittle materials, ρ must be modified, since as $\rho \to 0$, $K_t \to \infty$.

Numerous methods of adjusting Equation (5) to reflect the behavior of real technical materials have been put forward. One which seems to relate to crystalline structures is developed in [1].

$$K_t \simeq 1 + 2\sqrt{\frac{a}{2\eta}} \qquad (6)$$

where η is the width of a particle at the notch root. This particle has dimensions small enough so that the stresses on its sides can be considered uniform. Various interpretations have been given for the proper dimensions of η; it is always considered to be a material constant, and for the present case, it seems useful to assume that it corresponds to d/2 when the crack is of this order of magnitude. Making this assumption, for microcracks one grain diameter in length where a = d/2 and η = d/2, then

$$K_t \simeq 1 + 2\sqrt{\frac{1}{2}} \simeq 2.4 \simeq K_{t\ min}$$

for a microcrack.

The above analysis outlines one compressor valve material failure mechanism in simple terms. If notches or hard angular carbides occur in the critically stressed area of a valve, there is a high probability that critical K_t's will exist, and the valve will fail early on a severe cyclic test. If no disturbances of critical strength exist within critically stressed sections, the valves will live indefinitely on the same test.

3 FATIGUE CHARACTERISTICS - THE DISTRIBUTION OF DEFECTS

Failure rate characteristics of the material may be found by plain and notch fatigue testing. For thin coil stock materials, pulsating tensile fatigue tests are conveniently made on conventional equipment. Staircase testing as outlined in ASTM STP91-A is a convenient means of obtaining both the mean fatigue strength and enough data at various stress levels to study the failure distribution characteristics of the material. Figure 3 shows the results of a test program on one of the common valve materials, a Swedish 1095 plain carbon steel. In this figure, the mean fatigue strength is shown by a dashed line, the finite life portion of which is obtained by a fit through the mean lives of the samples tested above 130 ksi. The 20% failure rate and 10% failure rate fatigue lives are also shown. The sloping line portions of these curves are obtained by fitting lines through the relevant failure percentages obtained from cumulative distribution plots of the samples tested at the various stress levels. The intersection of the horizontal portion of the life line (the fatigue strength limit for the various probabilities) is obtained from the graph of probability of failure versus cyclic stress level shown in Figure 4. Background development for the statistical methods can be found in [3], [4] and [5].

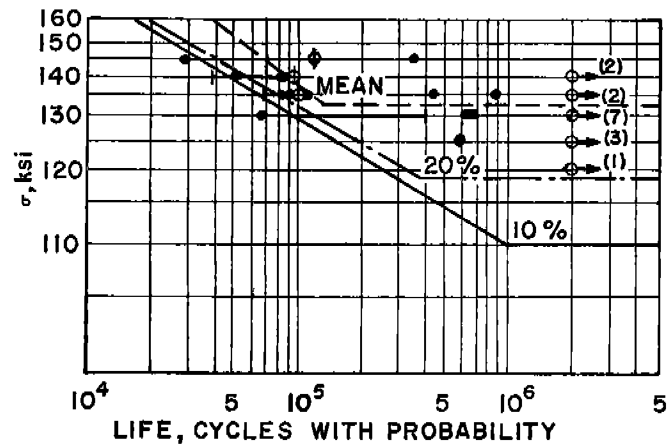

FIGURE 3

Swedish 1095 Plain Carbon Steel Fatigue Characteristics on Pulsating Tension, Smooth Sample, Constant Load Range Tests

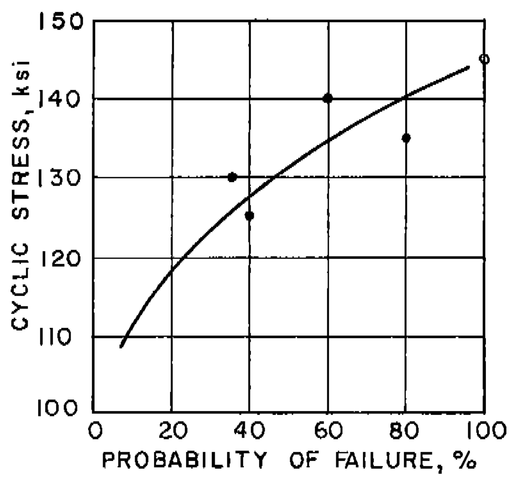

FIGURE 4

Swedish 1095 Plain Carbon Steel Failure Proportion Characteristics

The probability of failure graph, Figure 4, is also obtained from the distribution plots. The fraction of each sample which failed (the number of survivors is shown by the runout symbols in Figure 3) is plotted here. This is a direct estimate of the probability of encountering a critical defect in the volume of material tested. In the present case, this volume is 14×10^{-3} cubic inches.

The failure rate characteristics of finished valves are similar to those of the material. Proprietary finishing processes and shot blasting may change the level of the fatigue strength significantly, but the distributional characteristics of failures will remain essentially unchanged. A representation of such a distribution is shown in Figure 5, a cumulative Weibull distribution function plot with time to failure. The Weibull cumulative distribution function is:

$$F(x) = 1 - \exp\left[-\left(\frac{x-\alpha}{\theta}\right)^{c}\right] \qquad (7)$$

with a corresponding probability density function:

$$f(x) = \frac{c}{\theta}\left(\frac{x-\alpha}{\theta}\right)^{c-1} \exp\left[-\left(\frac{x-\alpha}{\theta}\right)^{c}\right] \qquad (8)$$

The failure rate is:

$$r(x) = f(x)/1 - F(x) \qquad (9)$$

α, c, and θ are the Weibull location, shape, and scale parameters, respectively, while x is the failure time.

From the shape parameter, c, we recognize that the data of Figure 5 are distributed as random failures; they have a constant failure rate with time. This is because the Weibull distribution with shape parameter equal to 1 simplifies to the negative exponential distribution. The data have been adjusted by inserting a location parameter, α, also known as the minimum life, of 0.05 hours to the failure times of the 65 failures in a test sample of 172 valves. All of these valves were run under identical stroke controlled bending cyclic testing. The failures abruptly cease at about 0.3 hours, equivalent to 130,000 cycles. Some valves were cycled beyond this time for many hours without further failures. It must be concluded that the probability of encountering a fatal defect in these valves under the working conditions is 38%. This number is the likelihood of encountering a fatal defect in the volume of material subjected to the test stress in this particular valve line. This critical volume is computer to be 1.82×10^{-6} cubic inches.

By generating a fatigue life with probability graph, such as Figure 3 for the finished material, a direct comparison of the stress concentration distribution causing the failure distribution seen in Figure 5 could be obtained from the 40% failure line on the fatigue graph. From Figure 5, we see that the failed valves follow a distribution which intuitively appears reasonable for a distribution of defect strengths as discussed in Section 2 above. The remaining portion of the sample has been truncated by the fatigue strength limit characteristic of the material. This reveals the likelihood of encountering a defect severe enough to cause failure at the test stress level.

The joint distribution of defect strength and spatial location can be deduced from the cumulative distribution function plot. In order to relate this function to various valve designs, notched fatigue tests are performed with the size and severity of the notches producing peak stresses in the test specimen comparable to the critically stressed volume in the valves.

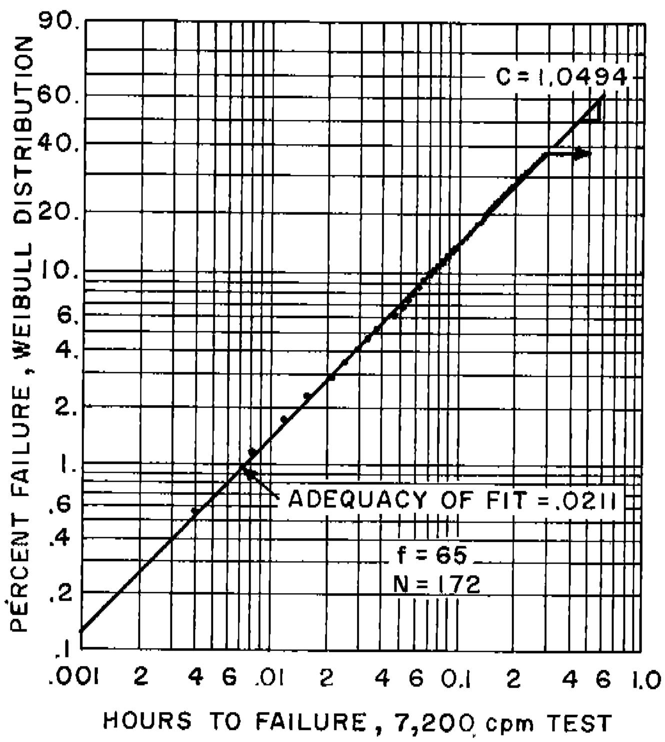

FIGURE 5

Adjusted Valve Failure Data for 65 Defect Failures. α, the Location Parameter, is Estimated at 0.05 Hours. The Probability of Failure is 38%.

4 VALVE DESIGN AND THE CRITICAL VOLUME OF MATERIAL

The design of the compressor valve and its mode of operation will determine the volume of material which is subjected to the peak stress. A further distribution of operating severity is encountered in actual refrigerant compressor operations. In most applications, this distribution is limited over a relatively narrow range and will not be considered further since unusual load cycles occur due to some malfunction in the refrigeration system.

Valve designs which stress the valve uniformly over the surface are most tractable, since the stress can be easily computed. In these valves, the critical volume is large, but it is easy to keep the maximum stress at a suitably low percentage of failure value. In spoked valves, the maximum stress is located in small enclaves at transition points in the spokes, and calculation of this stress is difficult and inaccurate. An experimental method such as outlined by Lang in [6] can be used to good advantage for this type of valve.

Notch fatigue tests which duplicate the critical volume of any given valve design can be carried out to determine the safe operating stresses which can be supported by any material. Further, by selecting representative notch severities, K_t from 2 to 4, comparisons of notch sensitivity may be made between materials. With data as shown in Figures 3 - 5 above, the fatigue performance and reliability of valve designs can be categorized. As an example, the spoked valve of Figure 5 was subjected to a stress level which produced approximately a 40% failure rate. From Figure 4, we see that this corresponds to approximately 127 ksi, which in turn provides a fatigue limit at 130,000 cycles on Figure 3. Since no further failures occur in this valve beyond the 130,000 cycle life, this defines the fatigue limit and the inflection point on the fatigue curve.

Using a similar slope to those already obtained from material tests, an estimate of the fatigue strength reduction factor, K_f, can be made. From the final failure at the three-hour point to the initial failure at 0.004 hours or about 1,700 cycles implies a stress range from 127 ksi to about 250 ksi (this last stress by linear extrapolation). Using 127 ksi as the fatigue limit, the maximum K_f is about 2. The corresponding K_t can be found by a method such as that of Peterson [7].

5 CONCLUSIONS

Compressor valve fatigue performance and the concomitant reliability is a joint function of the distributions of defect strengths, spatial density, and operating load severity. The distribution of failures represents this compound probability. From a study of the failure rate characteristics along with the results of material tests, the distribution of defect strengths and the likelihood of encountering a fatal defect can be found.

Valve design determines the critical volume of material subjected to potentially harmful stresses. An alternative characterization is that the distribution of stress concentrations will produce failure at the maximum stress location for a given operating stress level. Compressor usage determines the severity of the operating conditions. These are generally well controlled and need not be taken into consideration in the mechanics of materials computations. The valve material along with its fabrication sets the defect density and strength.

Conventional materials and commercial valves are generally well controlled and clean with a range of stress concentration factors between 1 and 2. Any mechanical damage due to mishandling, improper assembly or circulating debris in the refrigeration system may produce surface damage with stress concentrations as high as 4 or greater. Naturally, this type of damage leads to rapid failure.

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