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ABSTRACT

A brief overview is given on compressor valve failures and possible failure causes are summarized. The paper presents the mechanical properties of materials used in valve manufacturing. Fatigue data for reversed and pulsating bending load, including Goodman diagrams, are shown. The maximum bending stress for a pulsating loading mode to which a material of certain suction or discharge valve can be exposed is given.

INTRODUCTION

In recent years developments in compressor technology have been directed towards higher efficiency both for refrigeration and air conditioning compressors. These latest developments deal also with the considerations of an alternative refrigerent to chlorofluorcarbons, CFC. Due to temperature and pressure differences on condenser and evaporator sides increased mechanical loading on compressor components is expected. These changes necessitate a higher reliability of compressor parts including valve components.

Compressor reliability, particularly in the hermetic area, is high and compressor failure rates are very low. However, there are compressor failures which are due to broken components. Examinations showed that most failures of moveable parts are caused by fatigue. Fractographic studies document that most of the valve failures are caused by bending fatigue loading or by impact of the valve reed on the seat or on the valve stop (1-5).

In this paper, recommendations are given concerning valve design, valve processing and dimensioning. Some aspects concerning suitable selection of materials are presented. Anything which may improve or prolong the operation of the compressor valve system is meaningful to machine operation reliability and profitability.

FAILURE ANALYSIS

Improval of valve reliability often depends on accurate failure analysis. Examinations are performed with simple light optical microscopy, LOM, or in a scanning electron microscope, SEM, when more detailed results of the fracture mode are desired.

The valve failure analysis can indicate design overstresses resulting from insufficient stress control or valve failures caused by failure of other compressor components like piston rods, shafts etc.

Most fatigue fractures of compressor valves initiate on the surface or on the edge where the highest dynamic loading occurs. Therefore, primary fatigue crack origins are surface defects, wear marks, corrosion pits or edge defects from valve blanking.

In order to increase volumetric efficiency there is a tendency to decrease the slot between valve plate and valve reed to a minimum. In many cases the distance is approx. 1.0 mm (0.04 in.) or less. It should be kept in mind that in this case the proper tumbling treatment of the valve reed edges is very difficult and during high loading stresses the unproperly treated edge could be sensitive to fatigue cracking (6).

Small gaps at the valve mounting rivets could be a cause of premature valve failure when this area is exposed to dynamic stresses. In such cases there is a high fracture probability caused by the stress concentration at geometric defects, in this case fitting holes, and the rest defects from blanking at the fitting hole edges.

Precise positioning of the valve reed over the seat is very important. It has been shown that unproper valve positioning or narrow valve seat design has an influence on the valve fatigue performance (7).

Another cause of valve failure is overloading of the valve component, sometimes caused by overheating or a sudden increase of the discharge pressure (8). In air or gas compressors operating in a marine atmosphere there is a risk of corrosion damage which could develop into fatigue cracking.

Fatigue life can also be reduced by incorrect lubrication causing valve plate sticking which can retard reseating of the valve. A delayed closing causes excessive loading which could also reduce expected fatigue life.

Extensive case studies indicate that valve failure causes can be classified as follows:

Material defects:	rolling defects, rough surface scratches, structural inhomogeneities, brittle oxidic inclusions exceeding a critical size, insufficient tensile strength
Design and manufacturing:	narrow slots and holes, edge defects from blanking, surface damage from excessive tumbling
Improper assembly:	inaccurate positioning of the valve rèed vs valve seat, valve fitting faults
Environmental effects:	corrosive elements (not in hermetic compressors), foreign particles, excessive wear marks, improper lubrication
Overloading:	valve flutter, slugging at delayed closings, multiple impacting from gas pulsations, overheating

VALVE MATERIALS

Three standard, hardened and tempered valve steels are presented here. The chemical compositions are shown below:

	weight % nominal					
Grade	C	Si	Mn	Pmax	Smax	Ni
Eberle 18	1.0	0.25	0.40	0.015	0.012	-
Eberle 13	0.70	0.25	0.50	0.015	0.012	-
Eberle 15N2	0.75	0.30	0.40	0.015	0.012	2.0

The grade Eberle 18 is used for suction and discharge valves in hermetic compressors, particularly for the thinner sizes 0.15 - 0.60 mm (0.006 - 0.024 in.). The maximum standard thickness is 1.0 mm (0.04 in.). For thicker valves, above 1.0 mm (0.04 in.), the grades Eberle 13 or Eberle 15N2 are recommended. The 2 % Ni alloy is for improved hardenability.

MECHANICAL PROPERTIES

The established standard for flapper valve steels is that with decreasing thickness the tensile strength increases. The reason for the correlation between the thickness and tensile strength is material blankablility. Sizes above 1.0 mm (0.04 in.) are difficult to blank when the tensile strength exceeds 1700 - 1800 MPa (246 - 261 KSI).

Table 1 lists the tensile strength, Rm; yield strength, Rp0,2; and proof stress, Rp0,01 for standard valve steel sizes.

Thickness		Tensile strength		Yield strength		Proof stress	
		Rm	UTS	Rp0,2		Rp0,01	
mm	inch	MPa	KSI	MPa	KSI	MPa	KSI
		+- 50	+- 7	+-100	+-14	+-100	+-14
0,152	.006	2050	297	1820	264	1580	229
0,203	.008	2000	290	1780	258	1540	223
0,254	.010	1950	282	1740	251	1500	217
0,305	.012	1900	275	1690	245	1460	212
0,381	.015	1850	268	1650	238	1420	206
0,508	.020	1750	253	1560	226	1350	195
0,80	.0315	1700	246	1510	219	1310	190
1,00	.0394	1650	239	1470	213	1270	184

Table 1 Static strength for valve steel Eberle 18 for thickness ≤1.0 mm (0.04 in.). For thicker sizes the grades are Eberle 13 or 15N2. Mechanical properties corresponding to those for 1.0 mm are valid for all thicknesses in this range.

The temperature dependance of the ultimate tensile strength and yield strength for Eberle 18 is shown in Fig. 1. Up to approx. 200 °C there is no reduction in the tensile and yield strength. Eberle 13 and 15N2 exibit similar behaviour. Knowing that the fatigue strength is a direct function of the tensile strength one can expect that up to 200 °C there will be no reduction in fatigue strength. This covers the application range of the hermetic compressors.

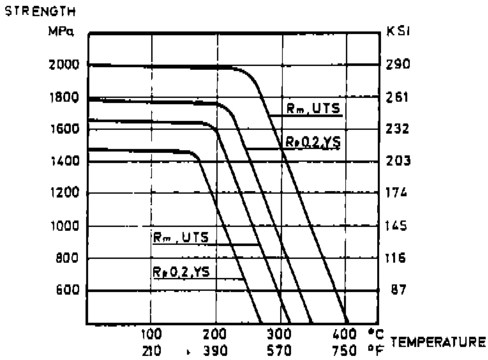

Fig. 1 Ultimate tensile strength and yield strength vs temperature for hardened and tempered flapper valve steel Eberle 18, thickness 0.203 mm (0.008 in.), Rm 2000 MPa UTS 290 KSI) and 1.0 mm (0.04 in.), Rm 1650 MPa (UTS 239 KSI).

FATIGUE PROPERTIES

The bending fatigue testing was performed on machine type Bosch. The vertical deflection of the samples was made by a double excenter. The testing frequency was approx. 25 Hz, the sample width 10 mm (0.4 in.). Length, 25 - 75 mm (1 - 3 in.) depended on the thickness.

Data are presented for bending fatigue strength for the thickness range 0.15 - 1.50 mm (0.006 - 0.06 in.). The fracture probability was determined for P0.5 and P50 %. The fatigue limits were determined for 10^7 loading cycles for reversed bending, stress ratio $Rs = \sigma min/\sigma max = -1$, Fig. 2 and for pulsating bending, stress ratio Rs = 0, Fig. 3. The increasing fatigue strength with the decreasing thickness is documented here for the thickness range 0.2 - 1.0 mm (0.008 - 0.04 in.). For the sizes above 1.0 mm (0.04 in.) there is only one tensile strength standard for all three grades. It is apparent that the increased tensile strength up to approx. 2000 MPa (290 KSI) gives an increased fatigue strength (4). Due to this correlation the thinner sizes can be exposed to a higher fatigue load which is in a good agreement with the practical experience for hermetic compressors.

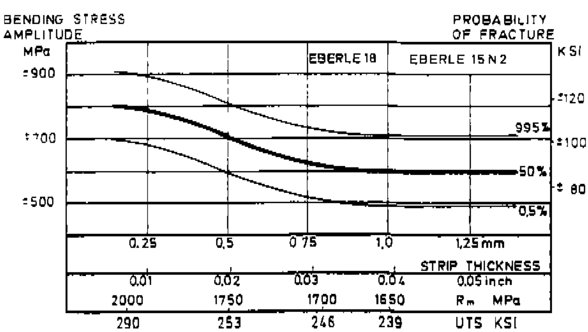

Fig. 2 Fatigue strength, reversed bending, stress ratio Rs = -1. Fatigue limits are valid for 10^7 loading cycles for various thicknesses (Note, there is no linear relation between thickness standard and UTS).

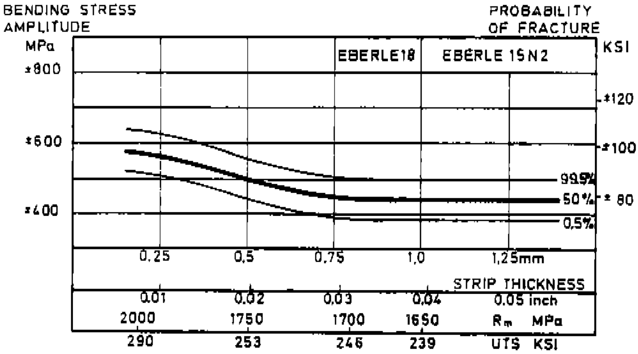

Fig. 3 Fatigue strength, pulsating bending, stress ratio Rs = 0. Fatigue limits are valid for 10^7 loading cycles.

Goodman diagrams are presented in order to estimate the maximum bending stress, σ_{max}, to which a valve material of certain suction or discharge valve can be exposed. A conservative approximation of the Goodman diagram for the thickness 0.152, 0.305 and 1.0 mm (0.006, 0.012 and 0.04 in.) is given in Fig. 4, 5 and 6 for P0.5 and P50 %. Considering a simplified loading of a suction valve, thickness 0.305 mm (0.012 in.) a maximum fatigue stress at pulsating bending for the considered material Eberle 18 is ca. 1000 MPa (145 KSI) at P0.5 %. For comparison, the thickness 1.0 mm (0.04 in.) shows approx. 740 MPa (107 KSI) at P0.5 %.

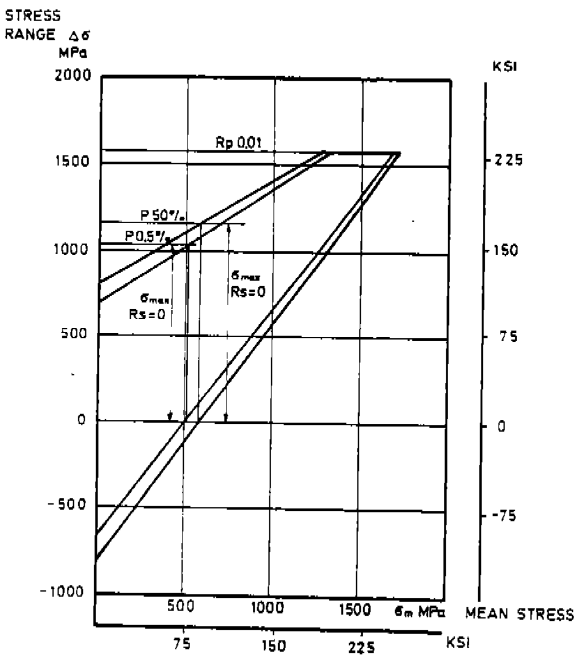

Fig. 4 Goodman diagram, bending fatigue 10^7 loading cycles. Flapper valve steel Eberle 18, thickness 0.152 mm 0.006 in. Rm 2050 MPa (UTS 297 KSI). Stress range $\Delta\sigma = \sigma_{max} - \sigma_{min}$.

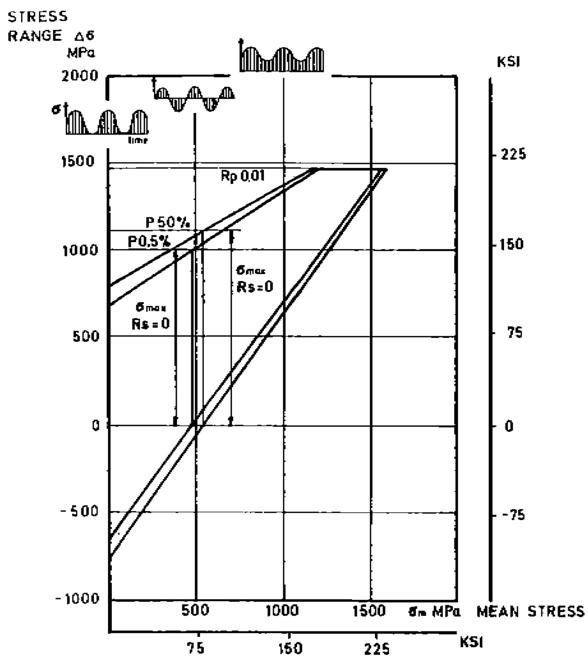

Fig. 5 Goodman diagram, bending fatigue 10^7 loading cycles. Flapper valve steel Eberle 18, thickness 0.305 mm (0.012 in.), Rm 1900 MPa (UTS 275 KSI).

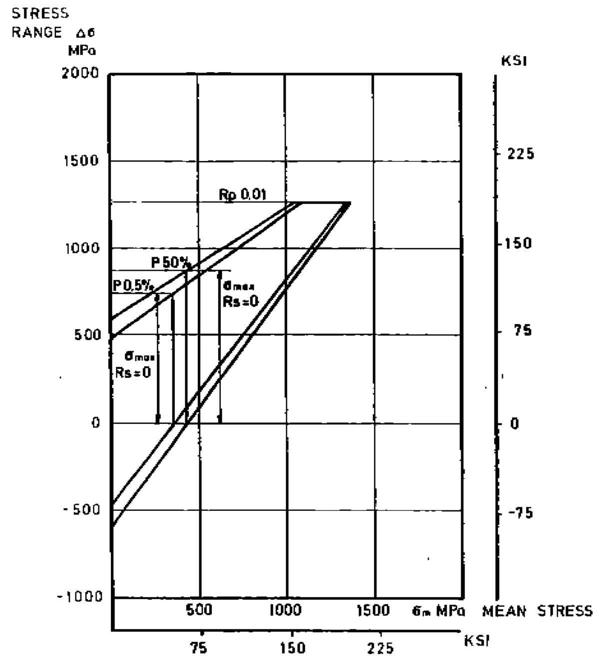

Fig. 6 Goodman diagram, bending fatigue 10^7 loading cycles. Flapper valve steel Eberle 18, thickness 1.0 mm (0.04 in.), Rm 1650 MPa (UTS 239 KSI).

With increasing mean stress σ_m, there is a decrease in amplitude of the fatigue strength. For high strength materials there is a linear Goodman relationship for the stress amplitude

$$\sigma_a = \sigma_{rb} \left(1 - \frac{\sigma_m}{R_m} \right)$$

σ_a stress amplitude
σ_{rb} reversed bending stress amplitude, $R_s = -1$
σ_m mean stress $\frac{\sigma_{max} + \sigma_{min}}{2}$
R_m tensile strength

The above expression is well valid for high strength valve steels, tool steels, spring steels etc. at bending or tensile fatigue loads (9).

FATIGUE AND DIMENSIONING ASPECTS

There are different methods for determining the actual stresses to which valve components are exposed. The most common is the finite element method, FEM, which gives the stress distribution for static and dynamic conditions. Calculated stress contours should be evaluated with respect to the determined maximum stresses and put in relation to the fatigue strength of the valve material. It should be noted that safety factors should be calculated for the considered bending load mode and the fracture probability required for the actual compressor valve system.

Despite mathematical analysis, practical testing in a laboratory compressor combined with accelerated life test and analysis of used or failed valves is important. Because of the complexity in determining valve operation mode and the corresponding stress system, this is the only way to obtain the necessary verification of the significance of the calculated data.

CONCLUSIONS

- Compressor valve failure analysis is an important tool in determining valve functional reliability.
- Most of the compressor valve fractures are caused by fatigue failures which initiate on the valve surface or edge.
- Valve design, valve treatment, assembly and positioning play a critical role in determining fatigue performance.
- For standard valve steel materials there exists a direct correlation between UTS and fatigue strength up to Rm 2000 MPa, 290 KSI. Thinner valve steel sizes used in hermetic compressors can be exposed to the higher fatigue loading.
- Reference data can be used to aid the design engineer. Design calculations must however be backed up by field testing.

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REFERENCES

(1) J. S. Laub:
Some Considerations in Refrigerent Compressor Valve Structural Reliability and Failure Mechanisms. Proceedings of the 1980 Purdue Compressor Technology Conf. pp 386 - 389, Purdue University, West Lafayette, In., USA.

(2) How and Why Compressor Valves Fail, Hoerbiger Bulletin HCA 8200, 1982.

(3) R. Dusil:
Studies of Faults in Used Valves. Case Studies. Proceedings of the 1976 Purdue Compressor Technology Conf. pp 99 - 105, Purdue University, West Lafayette, In., USA.

(4) R. Dusil & B. Johansson:
Material Aspects of Impact Fatigue of Valve Steel. Proceedings of the 1978 Purdue Compressor Technology Conf. pp 116 - 123, Purdue University, West Lafayette, In., USA.

(5) A. Futakawa et al:
Dynamic Stress of Ring Type Discharge Valve in Refrigerating Compressor. Mitsubishi Electr. Corp. Report DMN 1538, 1974.

(6) R. Dusil:
On Blanking, Tumbling and Shot-Peening of Compressor Valves. Proceedings of the 1978 Purdue Compressor Technology Conf. pp 129 - 134, Purdue University, West Lafayette, In., USA.

(7) R. Dusil & B. Johansson:
Influence of Seat Positioning and Seat Design on Valve Fatigue Performance. Proceedings of the 1980 Purdue Compressor Technology Conf. pp 368 - 373, Purdue University, West Lafayette, In., USA

(8) W. Soedel:
Design and Mechanics of Compressor Valves pp 110 - 112, Ray W. Herrick Laboratories, Purdue University, West Lafayette, In., USA, 1984.

(9) Z. Dylag & Z. Orlos:
Material Fatigue and its Testing, pp 29 - 32, SNTL, Prag, 1968.

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