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PLASTIC DEFORMATIONS OF DISCHARGE VALVES IN HERMETIC COMPRESSORS

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ABSTRACT

After running hermetic compressors under hard conditions for a long period, one now and then observes a deformation of the discharge valves across the seats. The deformation, the hardness and the residual stress have been measured on a number of valves, and it proves that the hardness and the residual stress have been lowered from the nominal values. The drop in hardness is caused by a tempering of the steel, and the lowering of the residual stress is probably due to stress relaxation. The tempering also lowers the yield strength of the steel. It also proves, however, that the tempering alone does not explain the plastic deformations. A probable explanation of the phenomenon is given, and the practical aspects of of a plastic deformed discharge valve are briefly discussed.

INTRODUCTION

In hermetic compressors, which have operated in extreme load conditions for long periods of time, we have sometimes observed plastic deformations of the valve across the discharge opening. The valves were discoloured and in cases where the temperature level was especially high incipient carbon formation was observed on the valves. The deformations can not be seen until the valves have been cleaned and polished, and this is perhaps the reason why this problem has not been discussed before, to the authors knowledge.

This paper investigates a valve type that is widely used in hermetic compressors, see fig. 1. First we shall deal with typical deformations, hardness and residual stress in the surfaces, after which we shall determine the maximal elastic stress in the valve in closed state as a function of the outlet pressure.

Even assuming a drop in yield stress on account of tempering, the deformations can not be attributed to an overloading. It must therefore be assumed, that it is a question of creep of the valve steel, but this can not be determined until creep tests have been made.

Finally we shall deal with the important question whether a deformed discharge valve can shorten compressor life. It will be demonstrated that even large deformations of the valve do not affect its function to any great extend, and we may conclude that life time is not influenced.

DEFORMED VALVES, ELASTIC STRESSES

In fig. 1 is shown a result of a series of tests mentioned in a later section

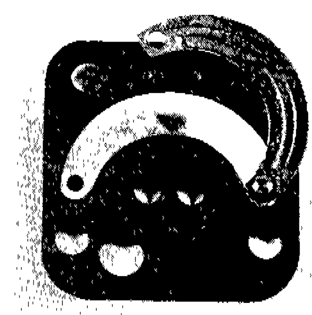

Figure 1.

The valve was loaded hydraulically, and in order to obtain large deformations across the discharge opening, the valve was made of brass sheet. The photograph has been included in order to illustrate the deformations. These are of course far less serious on the real valves.

On a number of deformed valves we have measured the deformations and the hardness, and also certain residual stresses in the surfaces using X-ray diffraction.

Typically values were as follows

Maximal deformations: 22 µm
Hardness : 622 HV

For a new valve corresponding nominal values are

Hardness : 655±30 HV

The drop of residual stresses were about 20% and it was observed that the stress relaxation evidently was greatest in one direction.

No general conclusion could be dedused from these measurements. We could nearly determine that hardness and residual stresses were redused probably as a result of tempering and stress relaxation respectively.

To determine the maximal stress in the closed state, the valve is approximated with a disc having a diameter corresponding to the width of the valve in symmetry, suppurted by the valve seat and loaded by an evenly distributed load.

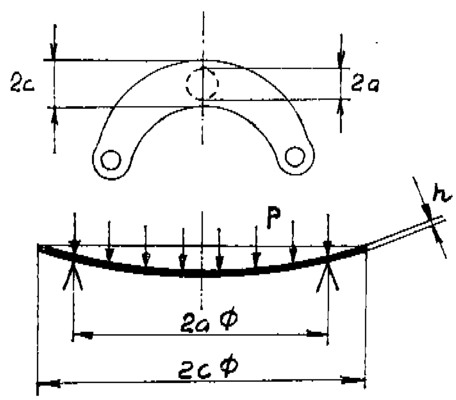

Figure 2.

The first step in calculation of stresses is the superposition of the elementary cases

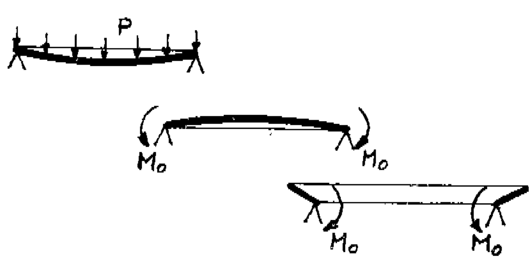

Figure 3.

From this we find the torque distributed along the seat and the torque in radial and tangential direction respectively

$$(1) \qquad M_o = \frac{p\, a^3}{8\left(a + \frac{c}{\alpha^2 - 1}\left(\frac{1}{\alpha^2} + \frac{1+\nu}{1-\nu}\right)\right)}$$

$$(2) \qquad M_r(r) = -\frac{p(3+\nu)}{16}\,(a^2 - r^2) + M_o$$

$$(3) \qquad M_\varphi(r) = -\frac{p}{16}\,(a^2(3+\nu) - r^2(1+3\nu)) + M_o$$

where ν = POISSON's ratio

$\alpha = c/a$

From the valve in question we have

$h = 0.152$ mm
$a = 2.2$ mm
$c = 3.4$ mm
$\nu = 0.3$

as substituted in (1), (2), (3) give

$$M_r(r) = (0.2062\, r^2 - 0.8270)\cdot p$$
$$M_\varphi(r) = (0.1187\, r^2 - 0.8270)\cdot p$$

and the stresses will thus be seen from

$$(4) \qquad \sigma_r(r) = \frac{6}{h^2} M_r = (53.55\, r^2 - 214.77)\cdot p$$

$$(5) \qquad \sigma_\varphi(r) = \frac{6}{h^2} M = (30.83\, r^2 - 214.77)\cdot p$$

The maximum stresses occur either at the seat or in the center of the disc. At the seat we have $r = a$ as substituted in (4) and (5) giving

$$\sigma_r = -0.57\cdot p$$
$$\sigma_\varphi = -65.55\cdot p$$

If the radial length of the seat is Δr, we have the following contact stress

$$\sigma_z = \frac{\pi a^2 p}{2\pi a \Delta r} = \frac{ap}{2\Delta r}$$

and if $\Delta r \cong 0.05$ mm we get

$$\sigma_z = 22.0\cdot p$$

To determine the reference stress we use v. MISES's yield hypothesis

$$(6) \qquad \sigma = \sqrt{\sigma_r^2 + \sigma_\varphi^2 + \sigma_z^2 - \sigma_r\sigma_\varphi - \sigma_r\sigma_z - \sigma_\varphi\sigma_z}$$

which gives

$$(7) \qquad \sigma = 78.73 \cdot p$$

In the center of the disc we have r = o and from (4) and (5) we get

$$\sigma_r = \sigma_\varphi = -214.77 \cdot p$$

We also have very nearly plane stress conditions and from (6) we get

$$(8) \qquad \sigma = 214.77 \cdot p$$

By comparing (7) and (8) we may conclude that in the given circumstances maximum stresses occur in the center of the disc.

TEMPERING, CREEP

If the valves are exposed to excess temperatures for long periods of time, annealing of the valve material occurs, and if in addition the valve is under load for most of the time creep is likely to occur in the loaded areas. In the following we shall assume that these two factors can be dealt with separately.

Tempering reduces the yield strength of the steel and if the maximum elastic stresses in extreme operating conditions become greater than the yield strength of the tempered steel, the deformations can be explained as a tempering phenomenon. If this is not the case, the cause is probably creep, and this requires examination of the creep characteristics of the material.

From our steel supplier we have received the figures 4 and 5 showing the changes in the characteristics of the steel at excess temperatures. Figure 4 shows the relation between temperature, time, hardness and ultimate strength. The steel has been tempered at a given temperature for a given time and the hardness and the ultimate strength have been determined at ambient temperature. The tempering temperature T in ^oK and the time t in minutes are substituted in

$$(9) \qquad X = T(\,15.4 + \log t\,)$$

which expresses as x-coordinate. The new hardness and the ultimate strength is read from the curve.

From figure 5 we get the ultimate strength at given temperatures after brief heating. There are curves both for the normally used quality and for a corresponding stainless.

The temperature in the gas flow near the discharge valve was measured at various operating conditions. For a valve operating in normal conditions we found

$$T_{max} \cong 185\ ^oC$$

and if we assume that the valve has operated at this temperature for 2ooo h we get from (9)

$$X = (185+273)(15.4+\log(2ooo \cdot 6o) = 9379$$

At this value the hardness is read from figure 4

$$HV \cong 620$$

The nominal hardness of the steel is 655 HV and thus the hardness has been reduced with the ratio

$$\Delta = 620/655$$

According to the supplier the yield strength ($\sigma_{o.ol}$) of the steel is

$$\sigma = 167\ kp/mm^2$$

and if we assume that this stress is reduced to the same extent as the hardness, we get

$$\sigma = 167 \cdot (620/655)\ kp/mm^2$$

which is the yield strength of the tempered steel at room temperature.

From figure 5 it will be seen, that brief heating to 185 ^oC reduces the ultimate strength from 1945 N/mm^2 to 1900 N/mm^2. The newly determined yield strength must be reduced correspondingly

$$\sigma = 167 \cdot (620/655) \cdot (1900/1945) = \underline{154\ kp/mm^2}$$

giving the yield strength at 185 ^oC after 2000 h at this temperature.

A temperature ratio of -5/+90 ^oC must be considered as being extreme operating conditions for a R12 compressor. The corresponding pressure conditions applies

$$p \cong 27\ kp/cm^2 = 0.27\ kp/mm^2$$

and substituted in (8) the maximum stress in the valve is

$$\sigma = 214.77\ 0.27 = \underline{58\ kp/mm^2}$$

Thus in operating conditions that only occur in extreme cases the maximum stress in the valve is 2.7 times lower than the yield strength. On this basis we must discard the tempering theory and attribute the deformations to creep.

So far, as the author is aware, no creep tests have been made on cold rolled band steel at temperatures below 200 ^oC. The majority of tests have been made on high alloy steels and at far higher temperatures than these mentioned here. In order to be able to explain the deformations fully at some later date, we have arranged with our supplier a number of creep tests to be carried out on cold rolled band steel at the relatively low temperature that occur in the discharge valve of a hermetic compressor.

THE EFFECT OF A PLASTIC DEFORMED DISCHARGE VALVE

Leakage of a discharge valve will cause an increase in the temperature of the outlet gas and a drop in capacity of the compressor.

Measurements have shown that the rise in temperature, among other things, depends on the operating conditions and the extend of the leakage, and rises in temperature up to 50° C are not unrealistic. This will bring about an increase in the temperature level of the compressor, and this can mean a reduction of compressor life time. It is therefore important to establish whether a valve with plastic deformations across the discharge opening has a leakage.

As mentioned before we have found deformed discharge valves in compressors, which have operated in extreme operating conditions e.g. on our life time panels. During such tests the temperature is constantly measured at various points in the compressor and the tests are always concluded with a measurement of the capacity. On compressors which subsequently proved to have deformed discharge valves abnormally high temperatures were not measured, and after the tests the capacities were normal.

In addition we have made a number of tests in which the valve in question was exposed to very high hydraulic pressures in order to test the tightness of the valve after deformation. It was not possible to deform the steel valves since the head gaskets blew out before the valve became deformed. The test was therefore performed on a valve made of brass, and after the load of approximately 100 kp/cm^2 plastic deformations of several times the thickness of the valve occured, see figure 1. After this the tightness of the valve was checked and this proved to be in order in all cases.

From these examples we may conclude that a discharge valve with plastic deformations across the discharge opening will not affect compressor operation and life, and must therefore be regarded as a "cosmetic curiosity".

ACKNOWLEDGEMENTS

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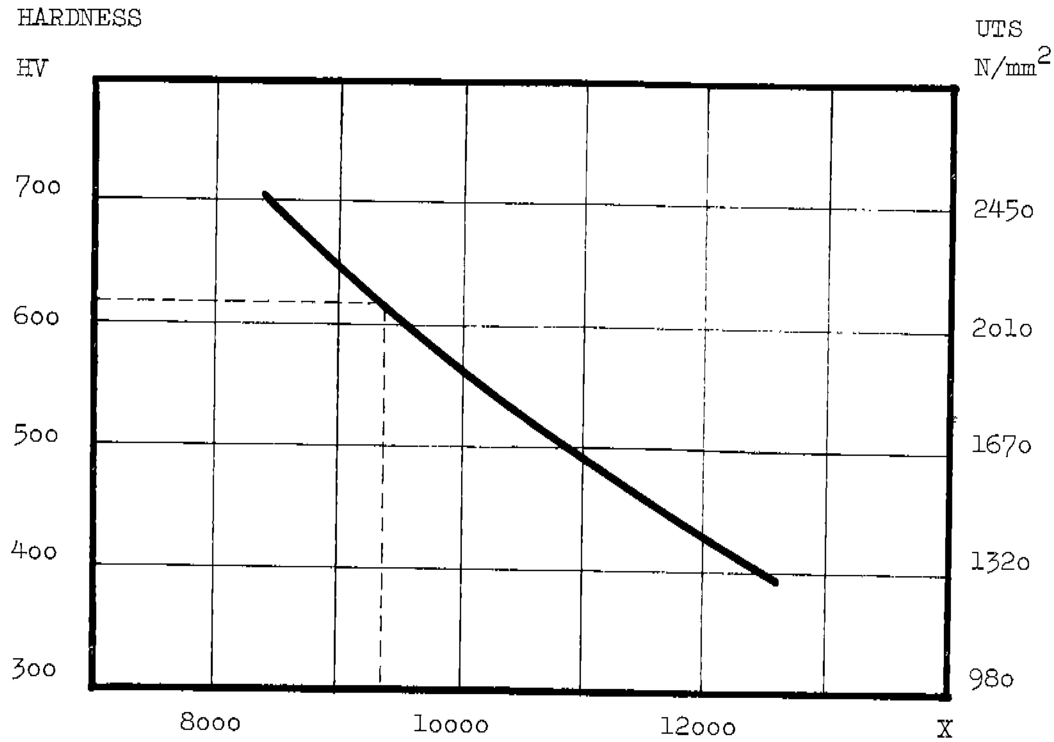

Figure 4. Hardness and tensile strength versus temperature, time.

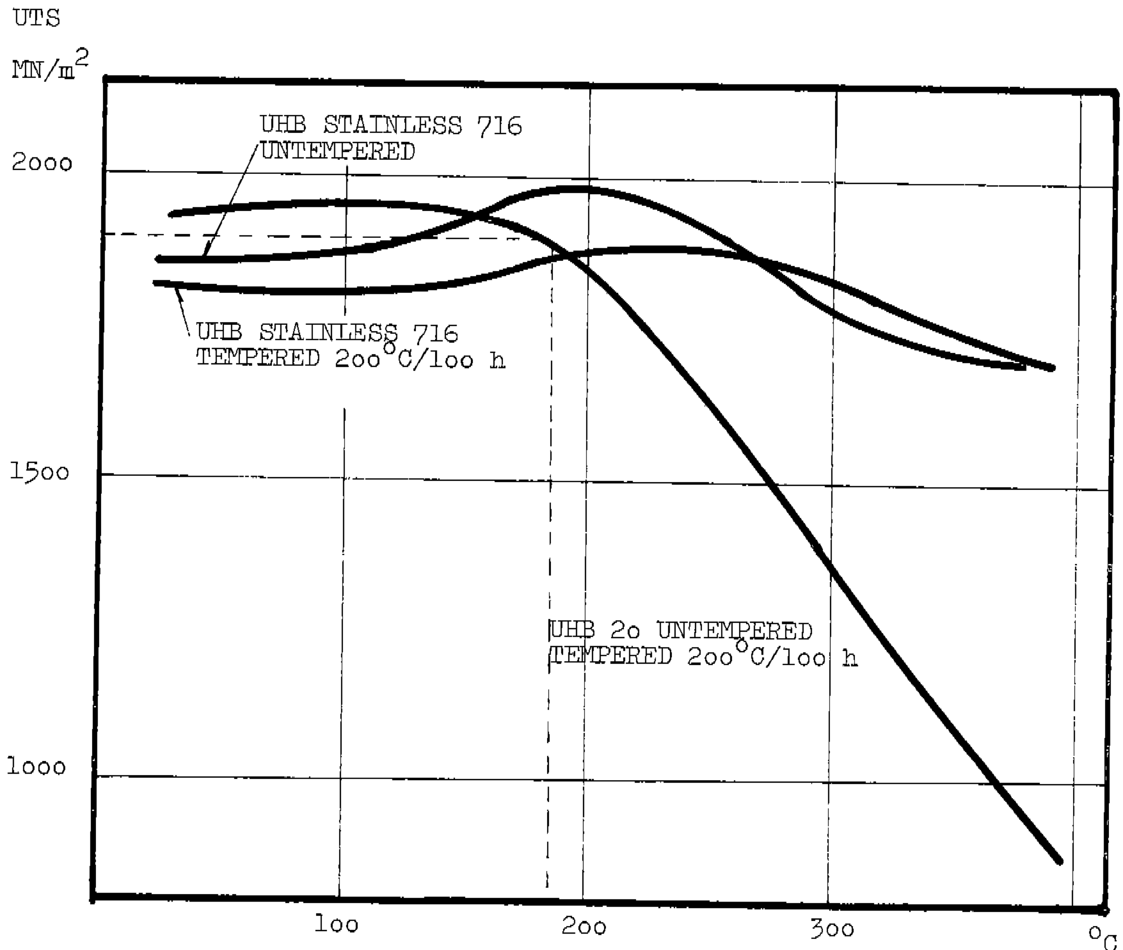

Figure 5. Tensile strength versus temperature.