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SEAT OPTIMIZATION OF PLATE VALVES

by

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ABSTRACT

Increasing the flow area of reciprocating compressor valve seats by employing larger valves is limited by the cylinder bore size and the dead space volume. Therefore efforts to increase the valve seat flow area are restricted to optimizing the valve design under the constraint of a constant outer diameter. This was the task of the project presented here.

Since the flow areas in a valve are interdependent, the geometries of the flow channels between the valve plate and the seat and those of the stroke limiter had to be considered simultaneously with the flow areas in the valve seat. Finite element calculations gave additional rating criteria.

It was possible to get an increase in flow area in the valve seat of about 13% with equal deformation and better stress distribution in comparison to the present design.

INTRODUCTION

Designs of reciprocating compressor valve seats in use today have demonstrated a long service life. There are no reports of stability and/or deformation problems. On the other hand, the flow area of the valve seat is the parameter limiting the compressor delivery. Increasing the flow area by employing larger valves is limited by the cylinder bore size and the dead space volume. Therefore efforts to increase the valve seat flow area are restricted to optimizing the valve design under the constraint of a constant outer diameter. This was the task of the project presented here.

The flow area was maximized under the following boundary conditions:

- The outer diameter of the valve seat was to remain unchanged in comparison to the present design.
- For the deformation of the valve seat under pressure a maximum level was defined.
- The appropriate distribution of stress should not show extreme concentrations.
- The valve plate lift was limited to values which give sufficient lifetimes in accordance with today's experience.

Since the flow areas in a valve are interdependent, the geometries of the flow channels between the valve plate and the seat and those of the stroke limiter have to be considered simultaneously with the flow areas in the valve seat. The optimization criterion chosen was the requirement of equal gas velocity in all channels of the valve, which can be expressed by means of a system of non-linear equations. A computer program was written to solve this system of equations.

At the beginning of the optimization stress and deformation of the present designs of valve seats were determined. Their flow areas were known. The finite element calculation was made for static load. The results of this calculation were the basis of the optimization work. To choose variants for finite element calculations we started from systematic investigations of geometries using our optimization program. Only such new seat geometries were chosen which gave an essential increase in flow area. The results of the FE-study gave additional rating criteria. The optimization procedure employed is described in detail for two valve seats of different sizes and the results are discussed.

FLOW CHANNELS AND VALVE GEOMETRIES

Presented in Fig. 1 is a compressor valve design in use today, the drawing referring to a suction valve. The gas flowing through the valve is directed downwards. The flow channels in the seat are covered and sealed by the ring-shaped surfaces of the valve plate in the closed position of the valve. In the open position, the valve and damper plates are depressed against the stroke limiter. It is in this position that the gas flows through various channels of the valve. Because the flow channels of the valve and damper plates, and those in the stroke limiter are radially displaced with respect to the flow channels in the seat, the gas must undergo two flow direction changes on the way from the seat channel to the stroke limiter channel.

Referring to Fig. 2, a valve is shown in its open position. The flow filament emerging from a seat channel is divided into two parts. On leaving the seat channel, each partial flow filament is deflected by about 90° before reaching the stroke-dependent flow area between the valve plate and the seat. After another 90° deflection, the partial flow filament enters the corresponding flow channels formed by the valve and damper plates, and the stroke limiter, all having the same cross sections. The partial flow filament considered joins here the corresponding counterpart coming from the neighbouring seat channel. As we see, one partial stream exiting a valve seat channel is directed to a valve plate channel having the radius smaller than that of the seat channel, whereas the other partial stream enters a valve plate channel with the radius greater than that of the seat channel. Therefore, individual flow channels within a plate valve are mutually dependent. A change in seat channel flow area results in a change of the valve and damper plates and stroke limiter geometries.

The distribution of flow channels in a valve seat in common use today is shown in Fig. 3, the channels being marked black. Depicted are two seats having different sizes. The smaller seat on the right is of the type with four radial bridges. The larger seat (left) has four radial bridges in the interior area. From the middle diameter outwards, there is a jump to 6 radial bridges. All bridges have equal thicknesses, the value of which is constant over the radius. Likewise, all black-marked flow channels have constant wall thickness, the channels itself being also equally displaced with repect to each other in the radial direction. As the total flow channel area amounts now to only 25-30% of the total mounting area of the valve seat (which is a bad usage of this area), a question can be posed as to what is the maximum value of the sum of all flow channel areas that can be attained. The optimization procedure described herein is demonstrated on the two above described seat variants.

COMPUTER PROGRAM FOR MAXIMIZATION OF FLOW CHANNEL AREA IN VALVE SEATS

Referring to the schematical representation of Fig. 4, the important flow areas are marked with broken lines.

The diameters Da and Di can be considered as prescribed, the former being determined by the mounting conditions in the compressor and the latter by the valve concept itself. The central screw and the suspension of the valve and the damper plates on the coaxially arranged guide springs determine the internal mounting diameter Di.

The factor that further limits the achievable flow area in the valve seat is the existence of radial bridges. Their number and width are determined by two opposing considerations, since they must provide for a sufficient valve seat strength without taking up too much valuable flow area. The experiences acquired in numerous strength calculations indicate that the valve seat deformation should be the major rating criterion.

The last factor that determines the total flow area is the number of radially arranged annular channels. It is to note that the flow area increases as the number of flow channels (with correspondingly increasing channel width) decreases. The channel width however is strongly related to the valve plate stroke. Since the stroke cannot be set to an arbitrarily large value (dynamics of the valve and damper plates), it follows that the number of annuli is also limited.

The task is then to achieve as large as possible flow area in the valve seat, subject to the constraints imposed by the diameters Di and Da, and maintaining realistic value of the plate stroke. The choice of variables encompasses the number of annuli, and for each of them, the number and the width of the radial bridges.

The boundary conditions pertaining to the problem specified are as follows:

- the ratio of the total flow area in the valve seat to the corresponding flow areas in the valve plate, damper plate and the stroke limiter should remain constant and equal to a prescribed value, and
- throttling of the gas stream in all of the flow channels between valve plate and seat should be constant over the valve radius.

Only pure geometrical areas will be considered (see Fig 4).

The above described relationship can be described by means of a system of quadratic equations, the unknown to be solved for being the diameters of the seat flow channels. A computer program was written to facilitate solving this equation system. The following input quantities are required for the solution:

- the mounting diameters Di and Da,
- the number of annuli in the valve seat (in the radial direction),
- the number and width of radial bridges connecting the neighbouring annuli.

In addition to the diameters of the valve seat and stroke limiter annuli, the iteratively obtained solution of the system of equations contains also the sum of the flow channel areas and the required valve plate stroke. It is the later quantity that determines the realizability of the valve variant calculated.

OPTIMIZATION OF THE FLOW AREA IN A SMALL VALVE SEAT

As the first example, we demonstrate the optimization of a small valve seat (Fig. 3 right). The starting geometry has four flow channels in the radial direction, divided by four constant-width radial bridges. The underlying geometry is summarized in Table 1 as variant A. Specified in the table are the radial bridge data, namely the number and width per flow channel from the center outwards. The data suggest that with four flow channels, an increase in the flow area is only possible when the bridge width is reduced. The flow channel area gain is, however, not large. This indicates that the starting geometry is already optimized to a high degree.

By employing three radial flow annuli, one achieves a noticeable increase in the flow area (variants E to L). However, this is achieved at the cost of a longer valve plate stroke. If the radial bridge geometry remains unchanged (variant E), the flow channel area rises by 5.4%. By varying the bridge geometry in both the valve seat and stroke limiter (variant H), one can theoretically attain a 21% flow channel area gain, the stroke increasing to 136% relative to the variant A value. The flow channel width is not constant anymore.

For practical reasons, the stress and strain calculations are shown only for variants I and K, and compared to the reference variant A. Referring to Fig. 5, variant A is shown in a deformed state (broken lines), caused by a constant load due to pressure difference over the valve seat. It is noticeable that the ring-shaped areas, interconnected by the radial bridges, are twisted, reducing sealing action in the valve seat - valve plate pair. In addition, there are pronounced stress concentrations at a middle radius in the bridge area (at the bottom side transition toward the flow channel).

Presented in Fig. 6 are deformations of the variant I (solid lines). Apparently, only the outermost ring section is lightly twisted. In comparison with A, the bending amounts to 90.3%, the corresponding stress amounting to 98%. The stress concentration zones are by far not so pronounced.

In the deformed state of variant K, there is no more twist to be seen (Fig. 7, solid lines). The bending amounts to 53.2% of the variant A value, at the same thickness and load. In comparison with A, the maximum stress of variant K is distributed over a much larger volume, its value amounting to only 67% of the corresponding A value.

On the basis of the stress and strain analyses, variant K is the best solution. The functional tests will have to prove whether the 36% increase in the valve plate stroke gives rise to function and life penalties. This variant brings about a 13.3% increase in the flow channel area.

OPTIMIZATION OF THE FLOW CHANNEL AREA IN A LARGE VALVE

This optimization relates to the valve seat geometry presented in Fig. 3 left. Contrary to the above described small valve seat, the number of radial bridges is in this case not linearly related to the seat radius (Table 2). Referring to the variant A, the first two flow channel annuli have four radial bridges each, to be followed by a jump to the annuli with six bridges each. The same geometry applies to the stroke limiter. All of the bridges have equal widths, the same being true for the flow channels. Similarly to the small valve seat, in the case of eight radial flow annuli one can achieve a gain in flow channel area only by reducing the bridge width and simultaneously varying the number of bridges. With the exception of an extreme variant (D) that can not be realized in practice, the maximum gain amounts to 8.7% (variant C) relative to the variant A.

A reduction of the number of flow annuli brings about more than 10% area gain. The solutions with six flow annuli (variants H and I) result in unrealistically large valve plate strokes. With seven annuli (variant G) one obtains max. 111.2% of the reference (variant A) flow area, the valve plate stroke increase amounting to only 17-18%. This case is also characterized by variable-width flow channels over the valve radius.

The stress and deformation studies of the reference variant (A) revealed a sharp stress concentration in the volume transition from the four to the six radial bridges. The stress maximum appears in the ring-shaped segment where the number of bridges jumps from four to six, and not in the radial bridge region, as in the case of small valve seats. This discontinuity of the radial bridge distribution is from the standpoint of stress analysis unfavourable. The broken lines in Fig. 8 represent the deformed structure. The front cross section is situated in the vicinity of one of the four inner radial bridges. Clearly visible is the angular displacement at the outer bridge in the third flow channel from the valve center outwards. The invisible cross-section at the rear of the seat segment drawn is situated in an outwards projecting radial bridge. Here also one finds twisted ring-shaped parts of the seat, the outermost ring displaying this quite clearly.

The variant F proved to be the optimal new solution. This valve seat with eight consecutive radial bridges is represented in the deformed state in Fig. 9 (solid lines). The twist of the ring-shaped surfaces remains in a narrow range. This solution features a flow area gain of 10.9%, the bending amounting to only 79% of that computed in the reference case A under the same conditions (seat thickness and load). The stress concentation is distributed over a larger bridge volume in the outermost flow channel. The maximum reference stress amounts to only 70.8% of the variant A value.

CONCLUSIONS

The ring-plate valve type used in contemporary compressors continues to prove itself as a mature design which can be further optimized only by investing a considerable effort. The goal of attaining a better usage of the valve mounting area was achieved. In spite of the high effort involved, the spatial stress analyses are absolutely necessary as a means of arriving at a material distribution in the valve seat such that in a complicated geometry no pronounced stress peaks occur. The geometry of this "heart valve" of the compressor is to be chosen so as to guarantee a high degree of operational reliability. The finite element methods offer a potential for the optimization of plate valves.

In the geometry optimization one should not forget that the flow through the valve should experience as small as possible pressure drop. A complete optimization should take this effect also in account.

seat variant	no. of annuli	valve seat				stroke limiter	stroke	flow area
		radial bridges (no. x width [mm])						
		annulus number						
		1	2	3	4		%	%
A	4	4x12	4x12	4x12	4x12	orig.	100	100,0
B	4	4x4	4x4	4x4	4x4	new	95	108,1
C	4	4x2	4x2	4x2	4x2	new	100	109,8
D	4	4x2	4x2	8x2	8x2	new	100	107,5
E	3	4x12	4x12	4x12	-	orig.	136	105,4
F	3	4x4	4x4	4x4	-	new	133	114,1
G	3	4x4	4x6	4x4	-	orig.	133	113,3
H	3	4x4	4x6	4x4	-	new	136	121,0
I	3	4x8	4x6	4x4	-	new	106	110,5
K	3	8x4	8x6	8x4	-	new	136	113,3
L	3	8x2	8x4	8x2	-	new	137	118.6

Table 1: Area optimization for the small valve type Ø 110 mm

seat variant	no. of annuli	valve seat								stroke limiter	stroke	flow area
		radial bridges (no. x width [mm])										
		annulus number										
		1	2	3	4	5	6	7	8		%	%
A	8	4x12	4x12	6x12	6x12	6x12	6x12	6x12	6x12	orig.	100	100,0
B	8	8x4	8x4	8x4	8x4	8x4	8x4	8x4	8x4	new	100	107,9
C	8	6x4	6x4	6x4	6x4	12x4	12x4	12x4	12x4	new	100	108,7
D	8	6x2	6x2	6x2	6x2	12x2	12x2	12x2	12x2	new	100	114,5
E	7	4x12	4x12	6x12	6x12	6x12	6x12	6x12	-	orig.	118	103,8
F	7	8x4	8x4	8x4	8x4	8x4	8x4	8x4	-	new	118	110,9
G	7	4x4	4x4	8x4	8x4	8x4	8x4	8x4	-	new	117	111,2
H	6	4x12	4x12	6x12	6x12	6x12	6x12	-	-	orig.	142	107,2
I	6	8x4	8x4	8x4	8x4	8x4	8x4	-	-	new	141	113,8

Table 2: Area optimization for the large valve type Ø 200 mm

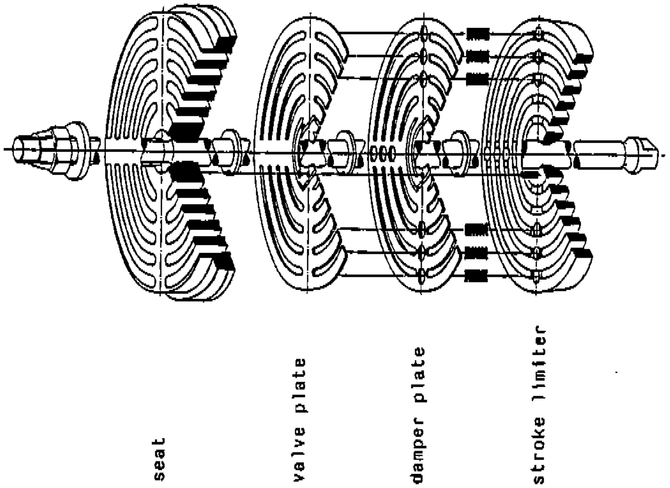

Fig. 1: Frictionless plate valve.

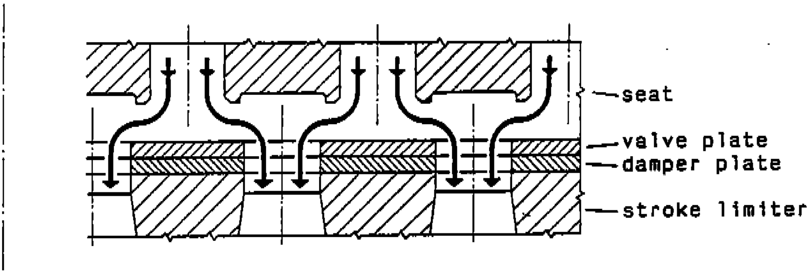

Fig. 2: Flow through valve channels.

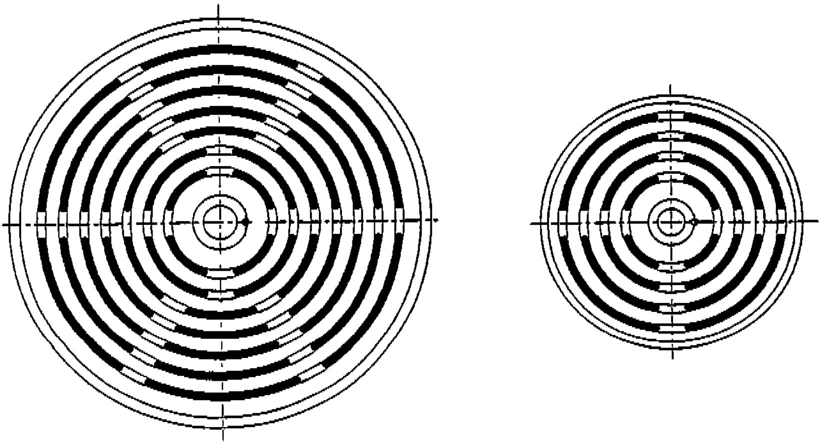

Fig. 3: Distribution of flow channels in valve seats.

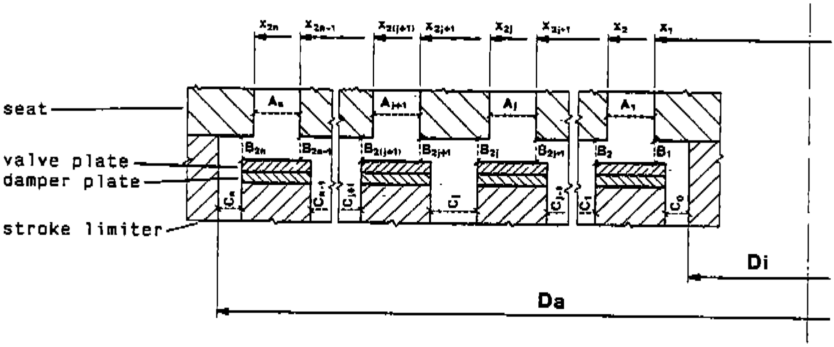

Fig. 4: Flow area relations

Aj, j=1,2,..n	Flow areas through circular channels (annuli) of the seat.
Bj, j=1,2,..2n	Flow areas through slots between seat and valve plate.
Cj, j=0,1,..n	Flow areas through circular channels of the stroke limiter.
xj, j=1,2,..2n	Diameters of circular channels through the seat.

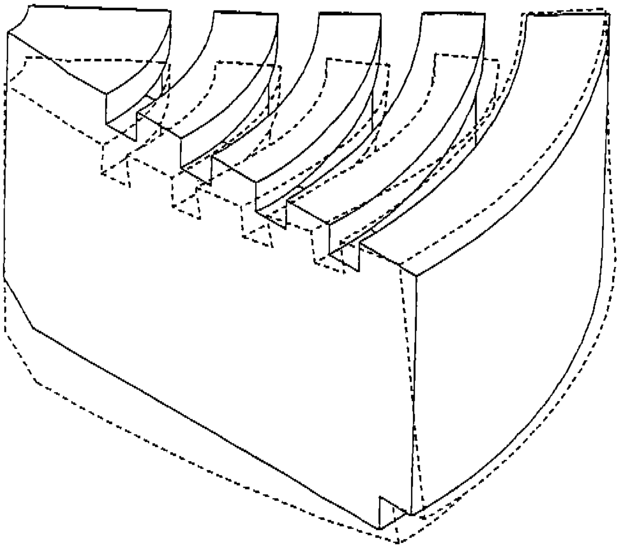

Fig. 5: Deformation of small valve seat, variant A.

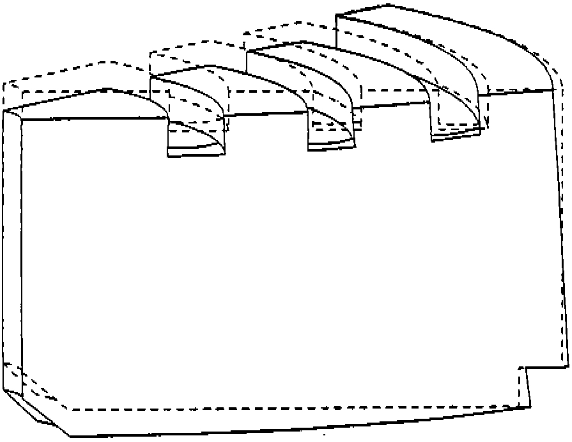

Fig. 6: Deformation of small valve seat, variant I.

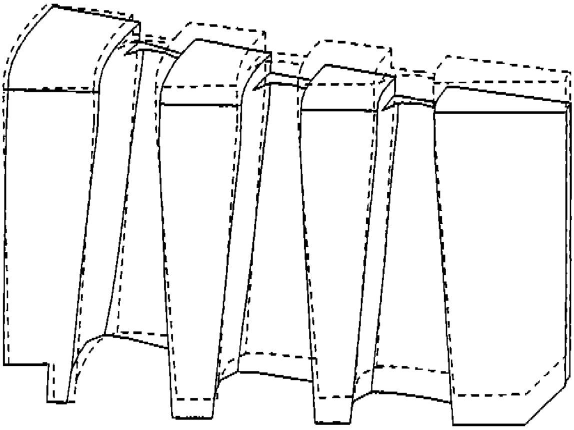

Fig. 7: Deformation of small valve seat, variant K.

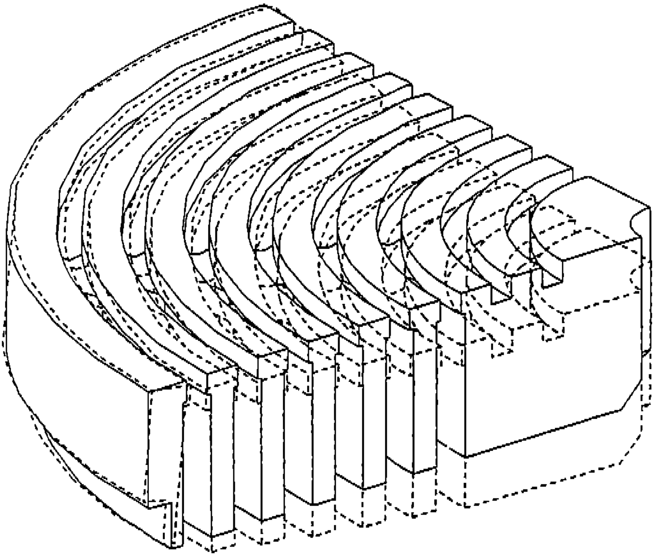

Fig. 8: Deformation of large valve seat, variant A.

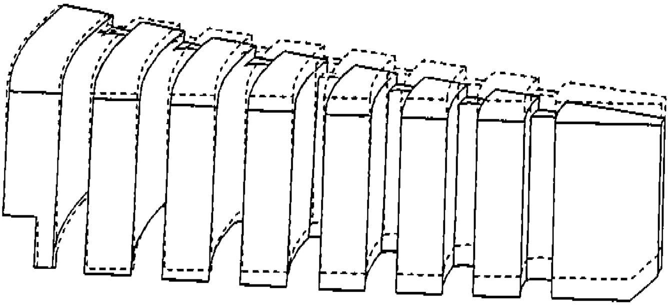

Fig. 9: Deformation of large valve seat, variant F.