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New Valve Design for Flexible Operation of Reciprocating Compressors

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ABSTRACT

For flexible operation of reciprocating compressors the performance of valves has been studied for fixed and adjustable bottom support of the valve springs. Computer simulations have been compared with experimental results.

NOMENCLATURE

n	compressor speed	rpm
Pc	cilinder pressure	bar
Pi	indicated power	kW
Ps	suction pressure	bar
Psm	mean pressure in suction plenum	bar
Pd	discharge pressure	bar
Pdm	mean pressure in discharge plenum	bar
Pvd	discharge valve losses	%
Wd	displacement of the discharge valve	m
Θ	crank angle	deg.

INTRODUCTION

In the oil and gas market there is an increasing demand for reciprocating gas compressors which can perform efficiently over a wide range of operating conditions (e.g. compressor speed, suction pressure, gas molweight). As manufacturer of horizontal balanced opposed double acting reciprocating compressors a bottleneck in designing a compressor for flexible operation is the performance of the valves. These valves (self acting, plate type) are designed for a specific operating condition at which they operate with maximum efficiency. Within a limited range of operating conditions beyond this design point the valves operate "acceptable", however with a reduced efficiency and/or life time. Computer simulation programs have been used to study the performance of valves for flexible operation.

THE COMPUTER SIMULATION PROGRAM

The Soedel computer simulation program of positive displacement type compressors presented in [1,4,5,] and the program presented in [6] was the base for the present simulation program. As manufacturer of double acting reciprocating compressors the computer simulation program is accomplised for a double acting type compressor. For the resonator approach of two resonators in serie with an anechoic pipe and a common discharge and suction plenum of the double acting compressor the equations 2.5.2. and 2.5.6. of ref. [5] were modifed into;

$$p_{Id} = \frac{c_{od}^{2}}{V_{oId}} \int_{o}^{t} \dot{m}_{d}\, dt - \frac{\rho_{od} c_{od}^{2} A_{Id}}{V_{oId}} \xi_{Id} + \frac{c_{od}^{2}}{V_{oId}} \int_{o}^{t} \dot{m}_{dc} dt$$

$$L_{Id} A_{Id} \rho_{od} \ddot{\xi}_{Id} + D_{Id} \dot{\xi}_{Id} + \rho_{od} c_{od}^{2} A_{Id}^{2} \left(\frac{1}{V_{oId}} + \frac{1}{V_{o2d}} \right) \xi_{Id}$$

$$- \frac{\rho_{od} c_{od}^{2} A_{2d} A_{Id}}{V_{o2d}} \xi_{2d} = \frac{A_{Id} c_{od}^{2}}{V_{oId}} \int_{o}^{t} \dot{m}_{d}\, dt + \frac{A_{Id} c_{od}^{2}}{V_{oId}} \int_{o}^{t} \dot{m}_{dc} dt$$

where

$\dot{m}_d$ = mass flow rate through the discharge valve head end (kg/s)

$\dot{m}_{dc}$ = mass flow rate through the discharge valve crank end (kg/s)

(for further nomenclature ref. [5])

By changing the subscripts d to s and replace $\dot{m}_d$ by $-\dot{m}_s$ leads to similar equations for the suction system.
Other adaptations are the preload of the valve springs and the implementation of damping plates. The effect of the damping plates is modeled by multiplying the velocity of the valve plate with a restitution coefficient above a certain displacement and within a certain area if the velocity of the valve is positive. Also the natural frequency of the valve is modeled as function of the displacement.

VERIFICATION OF THE SIMULATION PROGRAM

Before studying the behaviour of valves of reciprocating compressors the simulation program has been verified on a Thomassen low constant speed air compressor. The necessary experimental results for the simulation program were measured for the valves of the low speed compressor by the Dutch National Aerospace Laboratory NLR.
The results of the simulation program verified with the measured data of the compressor are satisfactory as shown in figure 1.

NEW VALVE DESIGN FOR FLEXIBLE OPERATION

To improve the performance of the valves for flexible operation on running compressors we have developed the idea to realise this by adjusting the bottom support of the valve springs which changes the working area of the valve spring characteristic. For a non-linear valve spring characteristic changes beyond the preload also the mean stiffness. Figure 2 shows a possible construction of a suction and discharge valve. The position of the bottom support of the valve springs can be adjusted by changing the pressure in the flexible bellows. By using the procesgas as medium for the pressure in the flexible bellows a simple control system for the operation of valves on running compressors can be achieved. Other possible constructions are discussed in ref. [3].
As well Thomassen as Hoerbiger (see ref. [2]) have put in a claim for a patent about the ideas on this subject.

THE PROTOYPE

Two prototypes have been built and tested after modification of the existing valves to examen the influence of changing the working area of the valve spring characteristic. One valve was tested on the low constant speed air compressor (422 rpm) and one on a high variable speed compressor (1000 rpm). Figure 3 shows the prototype that has been used on the high speed compressor. A similar construction has been used on the low speed compressor. Three inductive proximator probes were installed to registrate the behaviour of the discharge valve plate. The bottom support of the spring can be adjusted by the manual controled central spindle. An inductive linear variable differential transformer registrates the displacement of the bottom support.

EXPERIMENTAL RESULTS

Tests on the low constant speed Thomassen reciprocating air compressor have shown (see figure 1) that an increase of the preload:

- delayed the opening moment of the discharge valve
- reduced the opening period of the valve
- increased the closing period in which the performance of the valve plate became more instable
- increased the valve losses.

The before mentioned results were measured by keeping the suction pressure, temperature and speed constant.

On the high speed Thomassen reciprocating compressor, built in a close testloop filled with natural gas, the influence of the speed on the behaviour of the discharge valve was tested on the 2nd stage of the compressor. For the high variable speed compressor table 1 shows relevant data of the original pressure range selected valve.

After modifying the original valve into the prototype (figure 3) the discharge valve was tested at the same operating conditions as those for the original valve. At different speeds of the compressor the influence of changing the working area of the valve spring characteristic was tested.

In table 2 are listed the results of the tested prototype at a slightly different working area of the original selected valve springs.

In table 3 are listed the results of the tested prototype at which the working area of the valve spring characteristic is adjusted to the speed of the compressor by changing the bottom support of the valve spring.

Figure 4 and 5 shows the results of 1000, 700 and 600 rpm of table 2 and 3.

CONCLUSIONS

- Springs are necessary to close the valves in time
- Springs increases the losses of the valves
- For a wide operating range valves with adjustable bottom supports of the valve springs are more efficient then those with a fixed bottom support
- The compressor simulation program predicts satisfactory results for the behaviour of valves.

REFERENCES

[1] M. Dahr and W. Soedel, "Compressor Simulation Program with Gas Pulsations", Short Course Text, Ray W. Herrick Laboratories, School of Mechanical Engineering, Perdue University, 1978.

[2] H. Hrabal, "Verfahren zum Anpassen eines Verdichterventils an unterschiedliche Betriebsverhaeltnisse des Verdichters und Verdichterventil zur Durchfuehrung des Verfahrens. European Patent EP-A O 242 349 disclosed 21 october 1987.

[3] D. Moes, "Adjustable Compressor Valves for Reciprocating Compressors", internal report 7 januari 1987, Thomassen International, Rheden, The Netherlands.

[4] W. Soedel, "Introduction to Computer Simulation of Positive Displacement Type Compressors", 1972 Short Course Text, Ray W Herrick Laboratories.

[5] W. Soedel, Gas Pulsations in Compressor Manifolds", 1978 Short Course Text, Ray W. Herrick Laboratories, School of Mechanical Engineering, Perdue University, West Lafayette, Indiana 47907.

[6] S. Touber, "A Contribution to the Improvement of Compressor Valve Design", 1976 Delft University of Technology, The Netherlands.

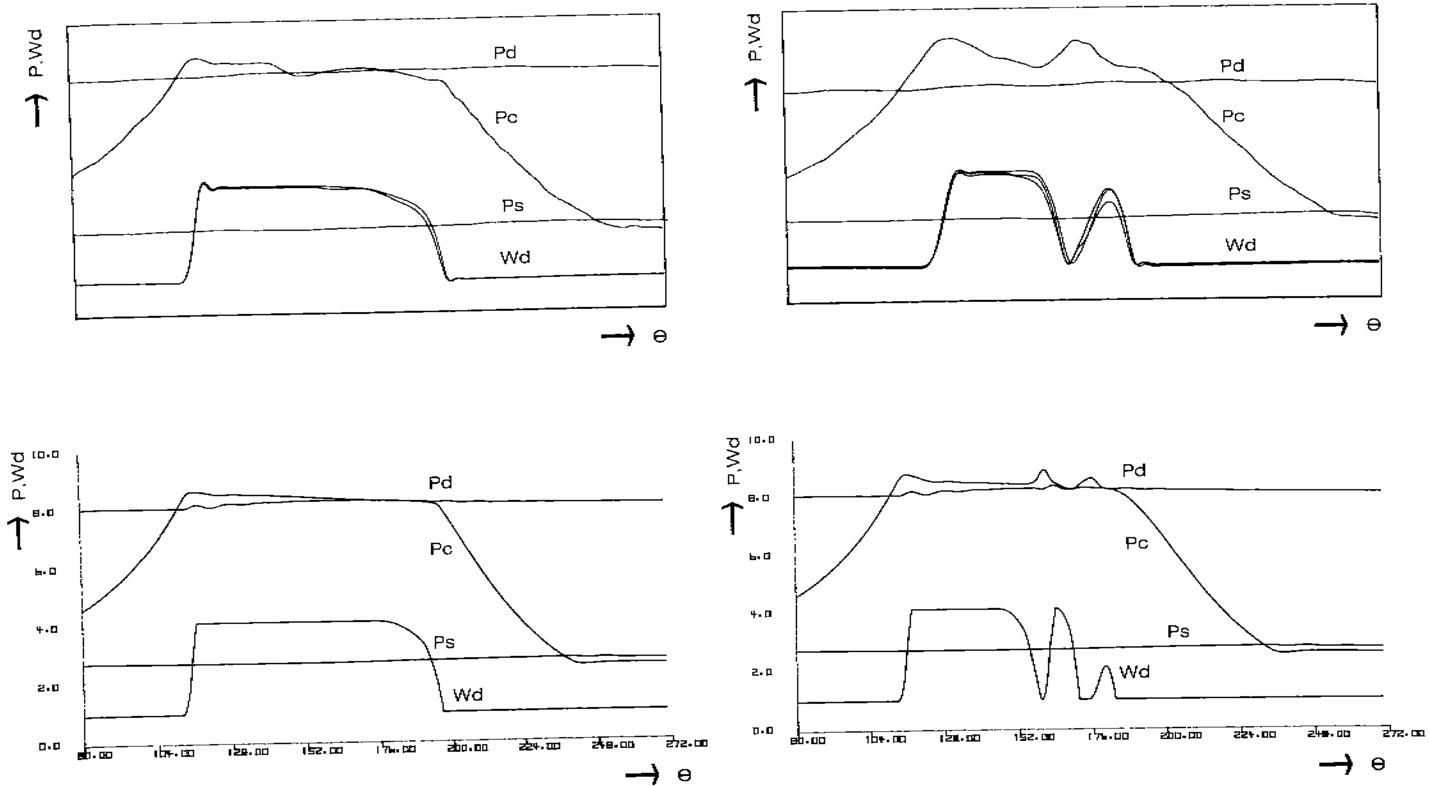
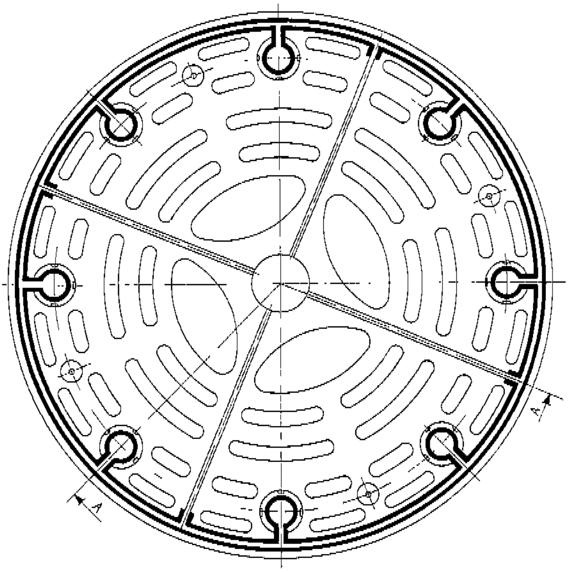
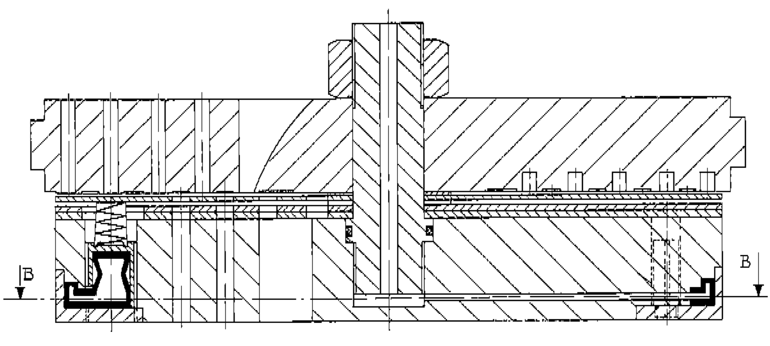


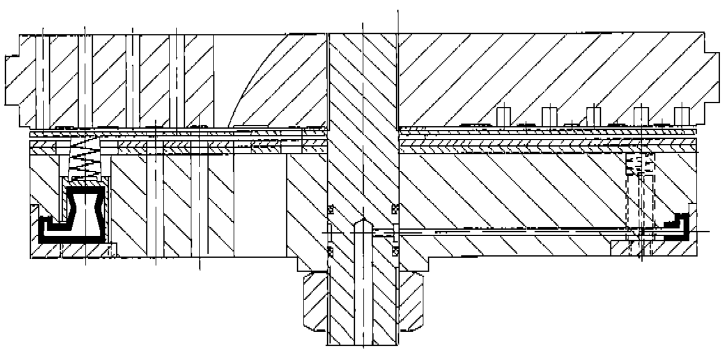
Figure 1. Simulated (beneath) and measured (above) results on a Thomassen low speed air compressor.



SECTION B-B



SECTION A-A



SECTION A-A

Figure 2. Adjustable suction and discharge compressor valve.

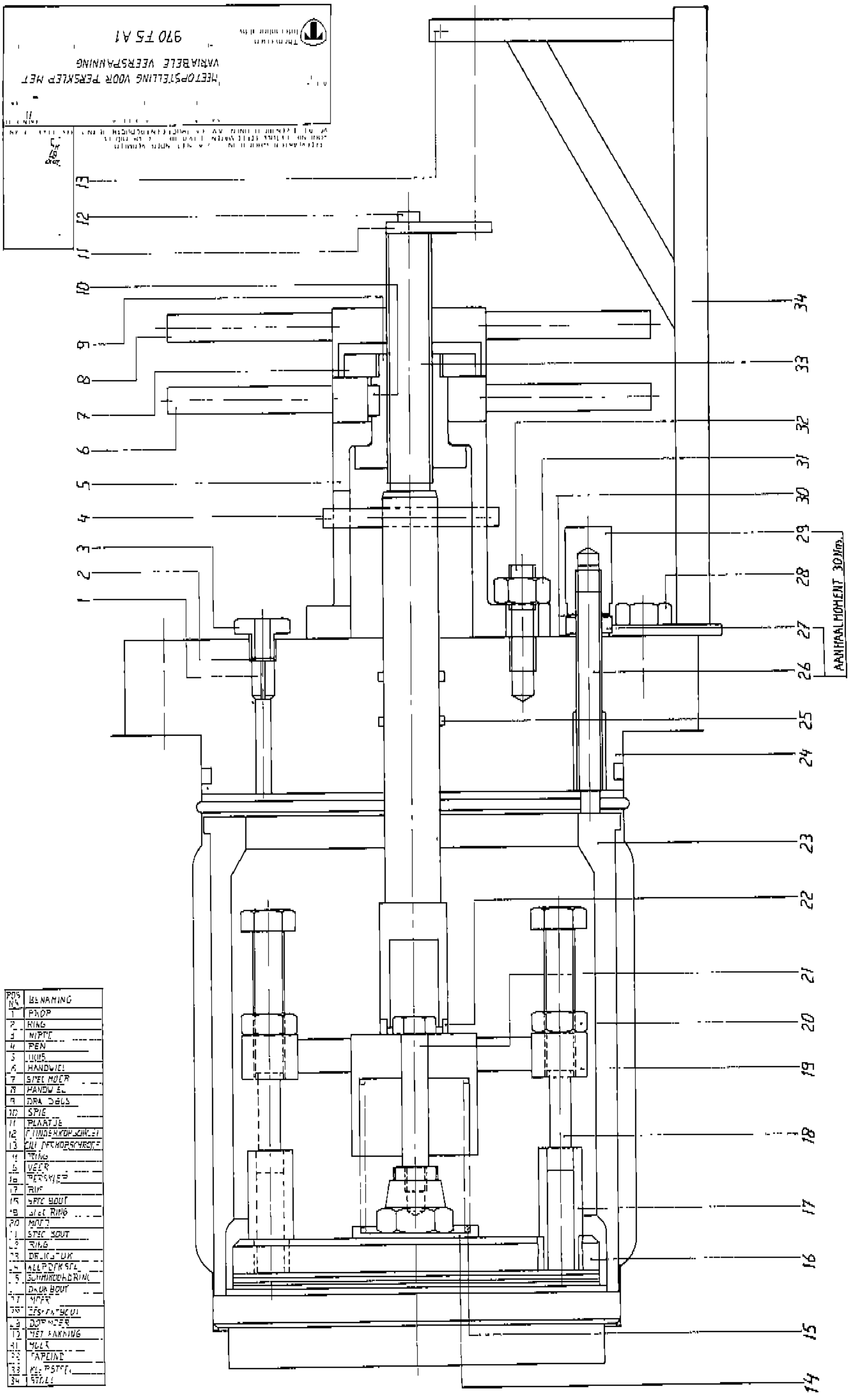


Figure 3 Tested prototype on a Thomassen high speed compressor with natural gas.

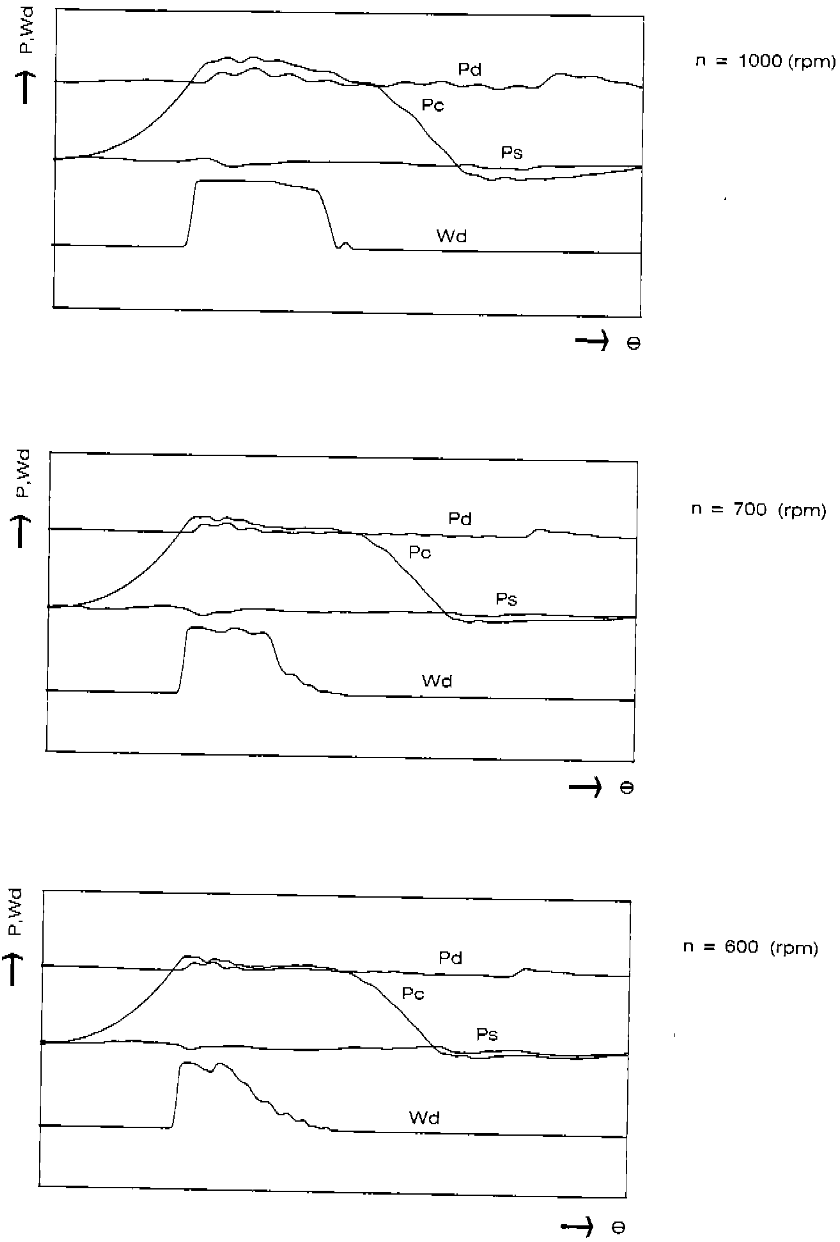


Figure 4. The performance of the pressure and discharge valve on a high speed compressor 2nd stage with a fixed working area of the valve springs characteristic.

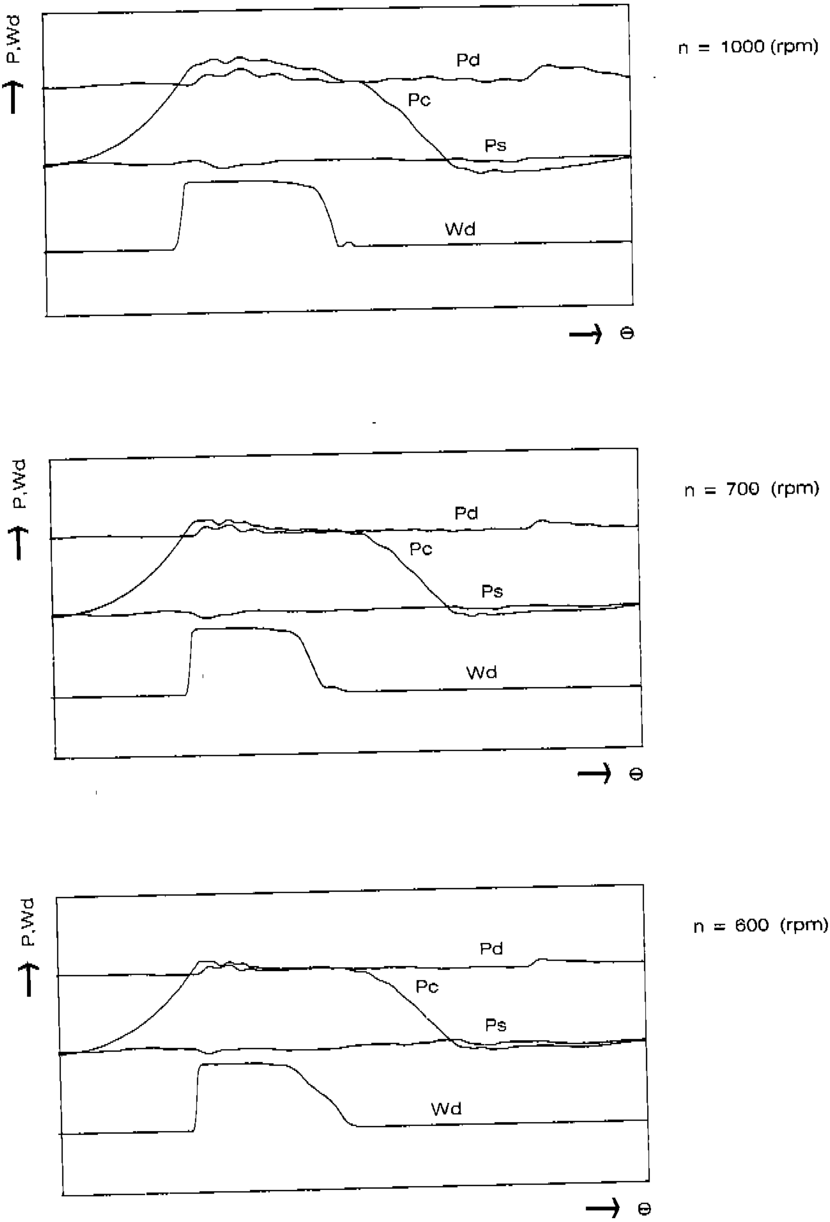


Figure 5. The performance of the pressure and discharge valve on a high speed compressor 2nd stage with a adjusted working area of the valve springs characteristic.

n rpm	Pi kW	Pv_d %	Psm bar	Pdm bar
1000	60.60	12.79	15.22	22.97
900	53.08	11.15	15.27	23.06
800	45.56	9.31	15.31	23.05
700	38.15	7.61	15.27	22.90
600	31.61	6.13	15.22	22.76

Table 1. Compressor data of a high speed compressor 2nd stage with pressure duty selected valves.

n rpm	Pi kW	Pv_d %	Psm bar	Pdm bar
1000	60.20	10.04	14.86	23.07
900	52.04	8.91	14.73	22.82
800	44.84	8.08	14.87	22.82
700	36.95	5.82	14.63	22.47
600	30.95	4.37	14.54	22.36

Table 2. Compressor data of a high speed compressor 2nd stage with a fixed working area of the valve spring characteristic.

n rpm	Pi kW	Pv_d %	Psm bar	Pdm bar
1000	59.99	10.09	14.81	22.97
900	51.96	8.73	14.68	22.75
800	44.61	7.45	14.75	22.70
700	36.93	5.27	14.51	22.37
600	30.58	3.37	14.33	22.13

Table 3. Compressor data of a high speed compressor 2nd stage with an adjusted working area of the valve spring characteristic.