BLADE PISTON COMPRESSOR

Piotr Kalina, Wlodzimierz Chomczyk, Andrzej Irzycki, Krzysztof Snopkiewicz

Institute of Aviation
Krakowska 110/114,02-256 Warsaw
tel.: +48 22 8464495, fax: +48 8465774
e-mail: kalina@ilot.edu.pl

Abstract

The paper describes the principles of building and working of new blade piston compressor. Dynamic force analysis of assembly of crankshaft and first research results are also presented. The working principle of this compressor was patented by prof. W. Chomczyk.
The piston, in shape of a blade, executes a pendulum movement in a circular cylinder. The two bulkheads with valves create four compression chambers for one piston. The compressor features compactness of construction and high efficiency. The lack of side thrust of the piston on cylinder surface is a major advantage of the construction. The lack of friction between crown and cylinder makes it possible to build an oil-free compressor. The strength calculation of chosen elements was executed in MES (Cosmos WORKS program). The acceleration of piston was a parameter which was optimized. The investigation of different tightening was conducted on a single-compression machine. The single compression machine was driven by pneumatic actuator having the task of imitation of piston movement during the compression cycle. The results of investigations of labyrinth seals are described in the paper. The compressor can find use in truck and bus aggregate drives. Paper present analysis of crank and crosshead mechanism's kinematics, and dynamic loads, strength analysis of compressor components and construction of compressor.

Keywords*: mechanics, compressor, MES, dynamic force analysis*

1. Introduction

Blade piston compressor of rotary-turning motion is the construction patented by Prof. Wlodzimierz Chomczyk (leading patent No. 68220). The piston in the form of blade moving in rotary-turning manner inside a circular cylinder realizes two cycles (intake and compression) in four chambers simultaneously. The change of swinging motion to the rotary one has been accomplished by means of crank and rocker mechanism.

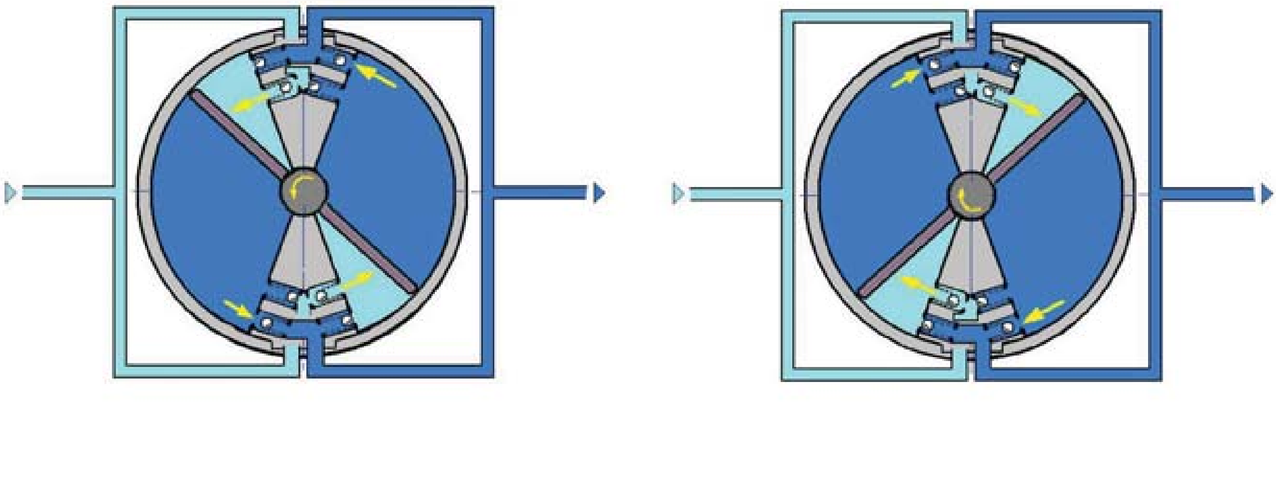


Fig. 1. Principle of operation of blade piston compressor

The presented conception of compressor eliminates the pressure of piston against cylinder wall (as in case of classic piston compressor), which should increase the durability of compressor as well as assure more noiseless operation. During the single revolution of drive shaft four duty cycles (compression) have been realized, what guarantee the higher efficiency of compressor. The construction of compressor is highly compact, the classic head as well as a crankcase are missing and four compression chambers situated in one cylinder permit the considerable reduction of its dimensions. The described compressor having cylinder diameter of 140 mm and height of 70 mm gives swept capacity about 1000 cm^3. Because of its modular structure, there is a possibility of any arbitrary development of compressor to its multi cylinder version.

2. Analysis of crank and crosshead mechanism's kinematics

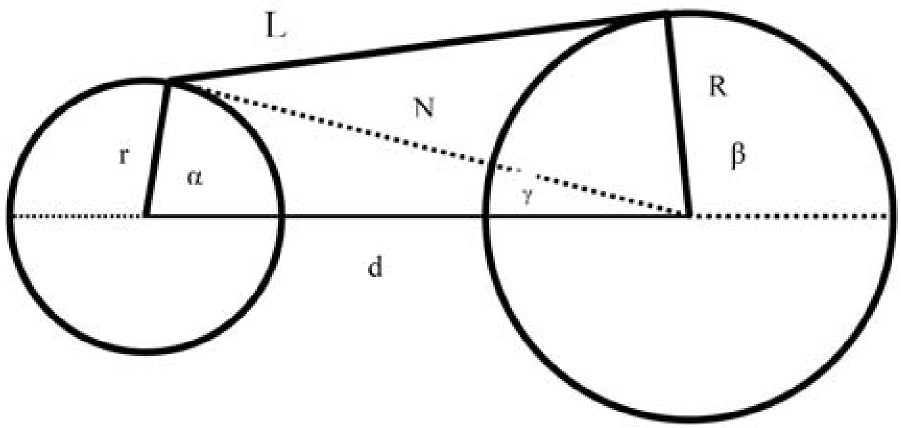


Fig. 2. Scheme of crank and crosshead mechanism of compressor drive, r - crank radius of driving engine; R - crank radius of compressor shaft; L - length of crosshead; d - distance between engine and compressor axes

$$N^2 = r^2 + d^2 - 2dr\cos(\alpha), \tag{1}$$

$$L^2 = N^2 + R^2 + 2NR\cos(\beta + \gamma), \tag{2}$$

$$\frac{N}{\sin(\alpha)} = \frac{r}{\sin(\gamma)} \Rightarrow \gamma = \arcsin\left(\frac{r}{N} * \sin(\alpha)\right), \tag{3}$$

$$\beta = \arccos\left(\frac{L^2 - N^2 - R^2}{2NR}\right) - \gamma, \tag{4}$$

$$\gamma = \arcsin\left(\frac{r}{\sqrt{r^2 + d^2 - 2rd\cos(\alpha)}} * \sin(\alpha)\right), \tag{5}$$

$$\beta = \arccos\left(\frac{L^2 - r^2 - d^2 + 2rd\cos(\alpha) - R^2}{2R\left(\sqrt{r^2 + d^2 - 2rd\cos(\alpha)}\right)}\right) - \arcsin\left(\frac{r\sin(\alpha)}{\sqrt{r^2 + d^2 - 2rd\cos(\alpha)}}\right). \tag{6}$$

3. Analysis of dynamic loads

The successive graphs show the forces calculated for the compressor having following parameters:
Cylinder diameter - 136 mm,
Shaft diameter - 40 mm,
Working angle (compression) - 110º,
Compression pressure - 1 MPa,
Compressor shaft speed 1400 rev/min.

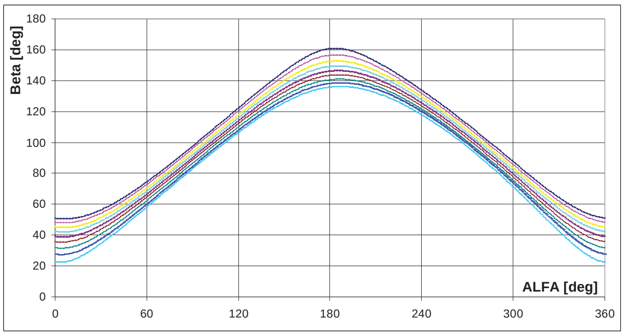


Fig. 3. Graph of blade displacement

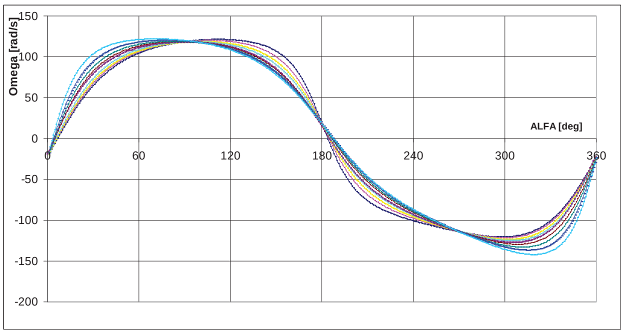


Fig. 4. Graph of blade displacement speed

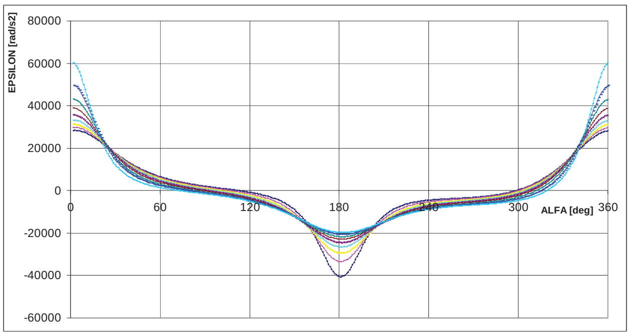


Fig. 5. Graph of blade acceleration

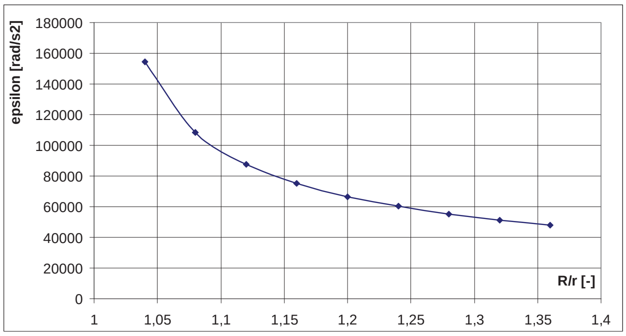


Fig. 6. Graph of blade acceleration versus quotient R/r

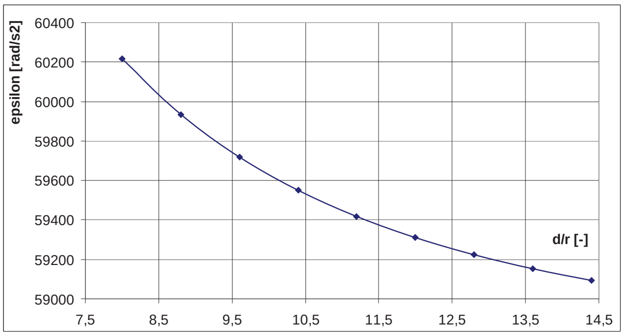


Fig. 7. Graph of blade acceleration versus quotient d/r

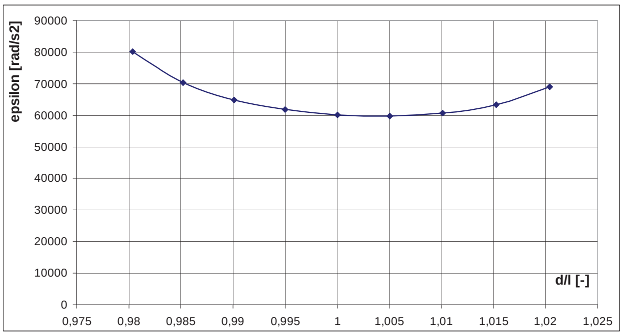


Fig. 8. Graph of blade acceleration versus quotient d/L

4. Strength analysis of compressor components

The strength analysis of compressor components for determined load values was carried out with use of FEM utilizing "COSMOS-Works" programme. The results of calculations for the shaft-blade assembly are showed below.

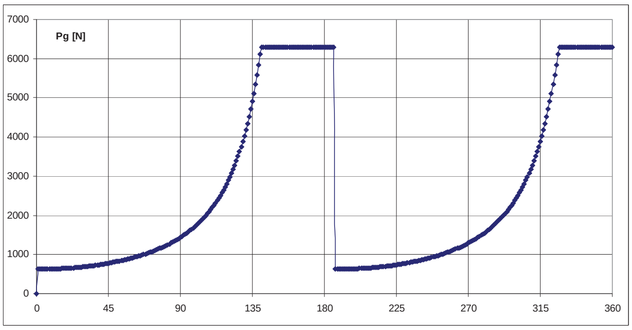


Fig. 9. Diagram of gas force in compressor

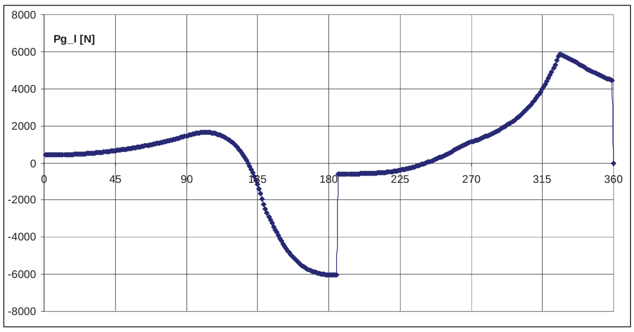


Fig. 10. Diagram of gas force changeability (component along connecting bar)

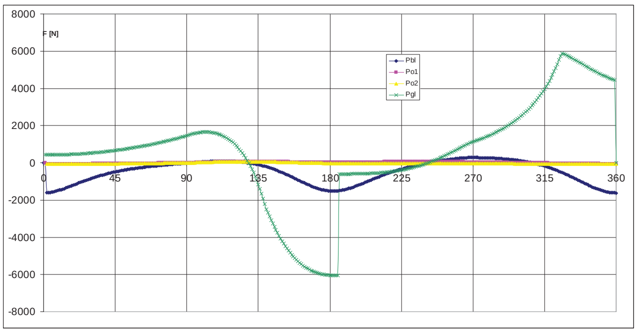


Fig. 11. Diagram of crank-pin forces changeability (Pbl - force of inertia of shaft; Po1,2 – forces of inertia of connecting rod components referred to shaft and driving engine axes)

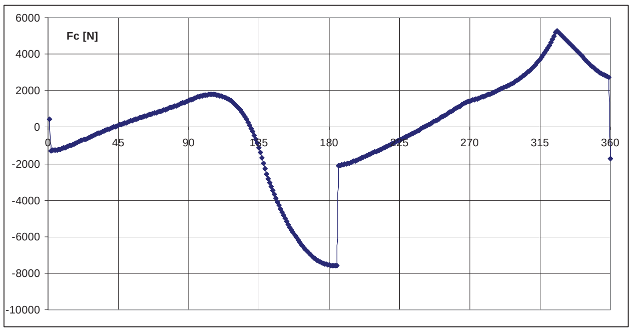


Fig. 12. Diagram of resultant force changeability on crank- pin

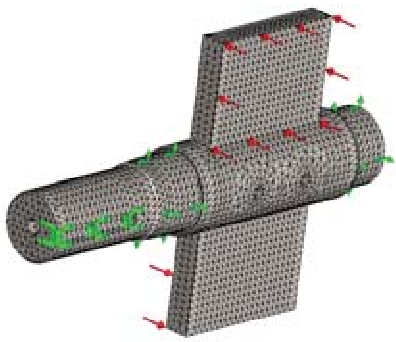


Fig. 13. FEM mesh applied to calculations of shaft

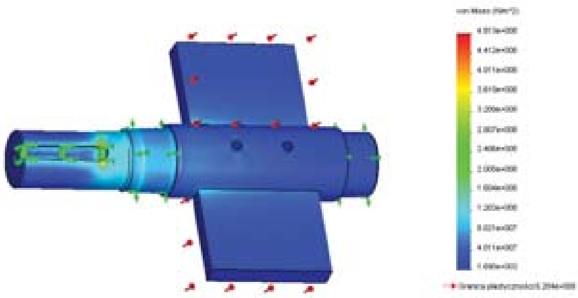


Fig. 14. Stress in blade of thickness 12 mm

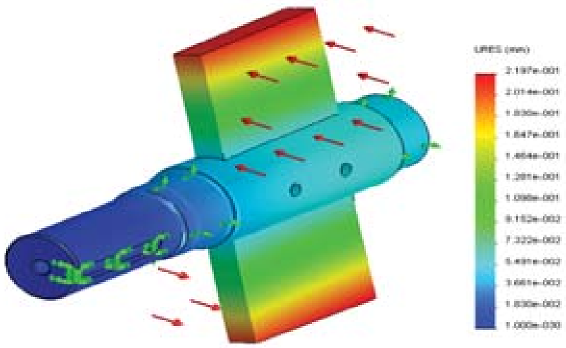


Fig. 15. Total strains in model

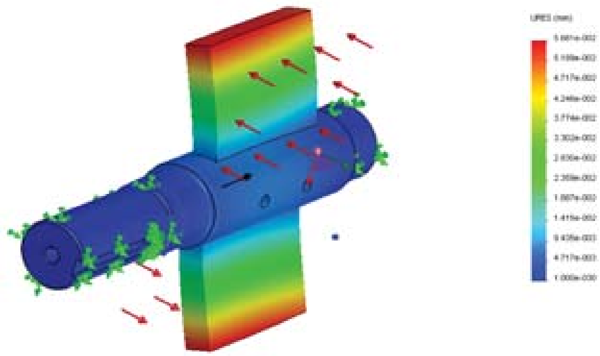


Fig. 16. Bending strains in model (without shift torsion)

The piston equipped with lateral disks was chosen as a second seal variant.

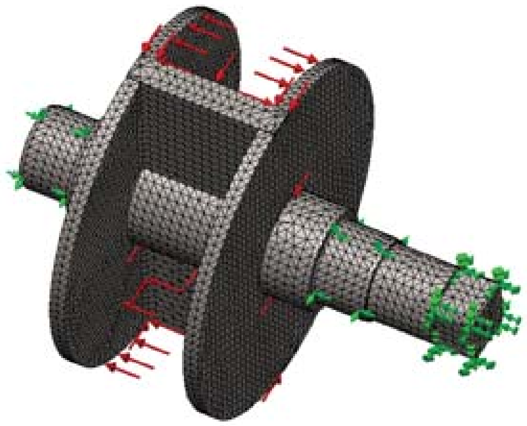


Fig. 17. FEM mesh applied to calculations of disks at shaft-blade assembly with lateral disks

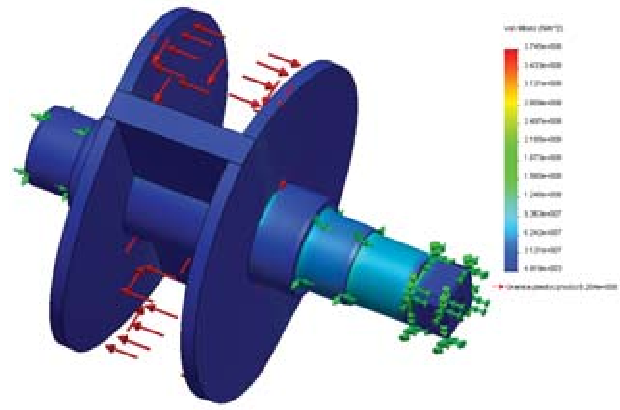


Fig. 18. Graph of stress in shaft-blade assembly with lateral disks at working pressure of 1 MPa

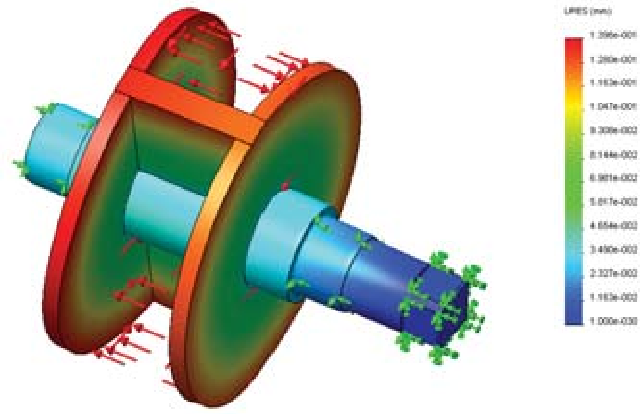


Fig. 19. Total strains in model 0,14 mm

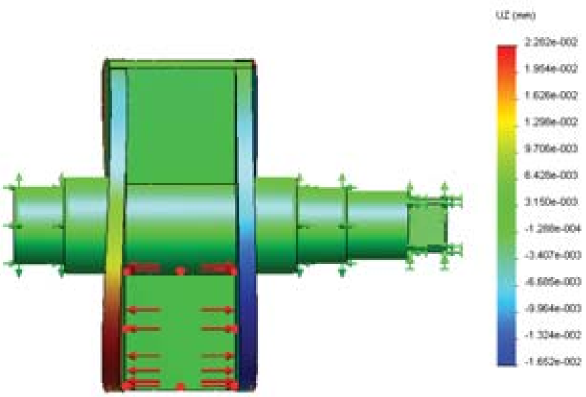


Fig. 20. Strains in axial direction only (0,023 mm)

5. Construction of compressor

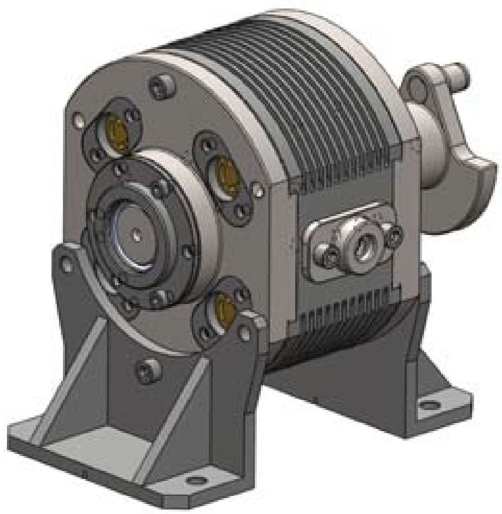


Fig. 21. View of compressor from suction valves

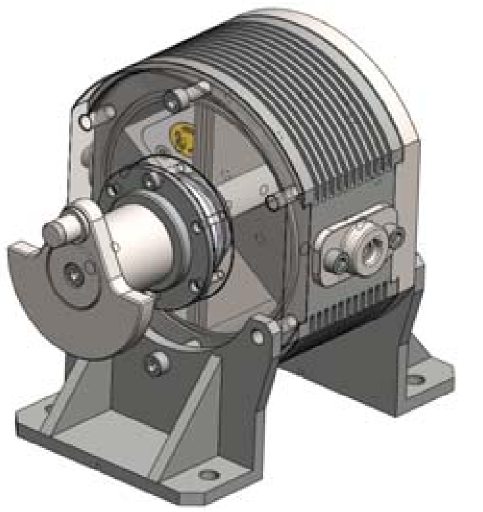


Fig. 22. View of compressor from driving shaft

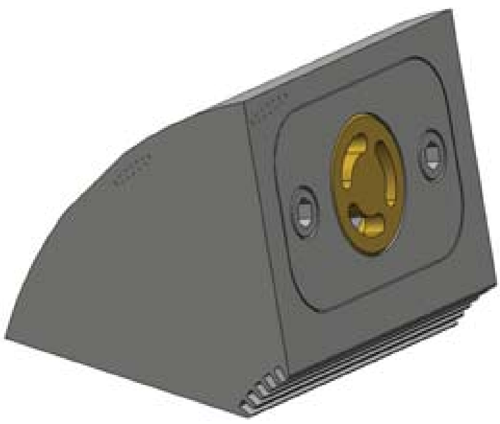


Fig. 23. View of baffle from built-in pressure valve

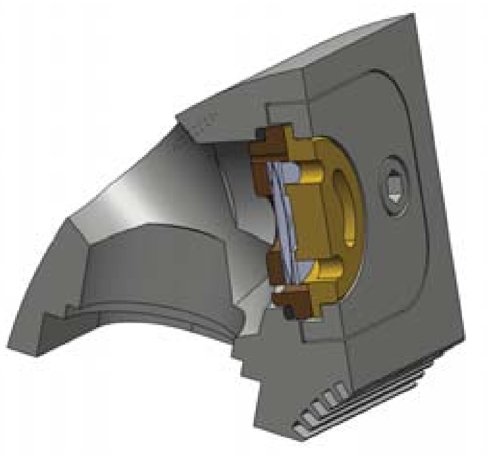


Fig. 24. Cross-section of baffle with pressure valve

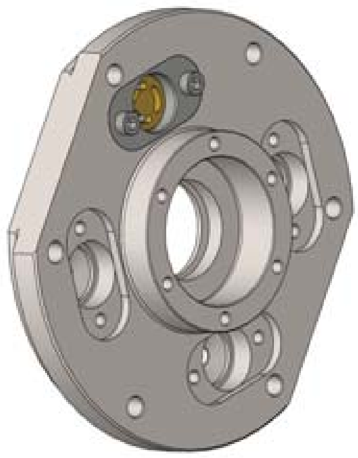


Fig. 25. View of front plate with built-in suction valve

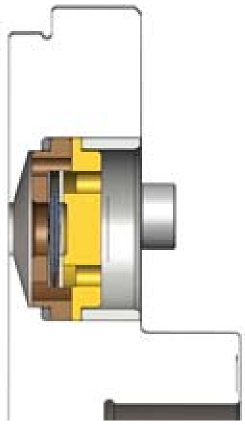


Fig. 26. Cross-section of plate with suction valve

6. The researches of seals

The single compression machine as the test stand provided the motion of the blade in circular cylinder similarly as in target compressor. A hydraulic servomotor served as driving unit of shaft-blade assembly. The hydraulic brake system along with the bumpers stop the motion of servomotor in definite position, fixed through the change of connecting rod length and the position of bumpers. In the chamber formed by piston (blade) and boundary baffle the quick-changeable pressure versus piston position has been measured. The speed of pressure increase as well as the pace of its decrease can be regarded as a quality measure of different seal variants.

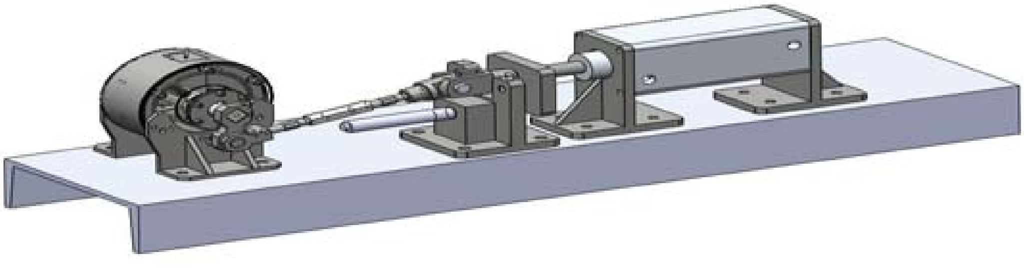
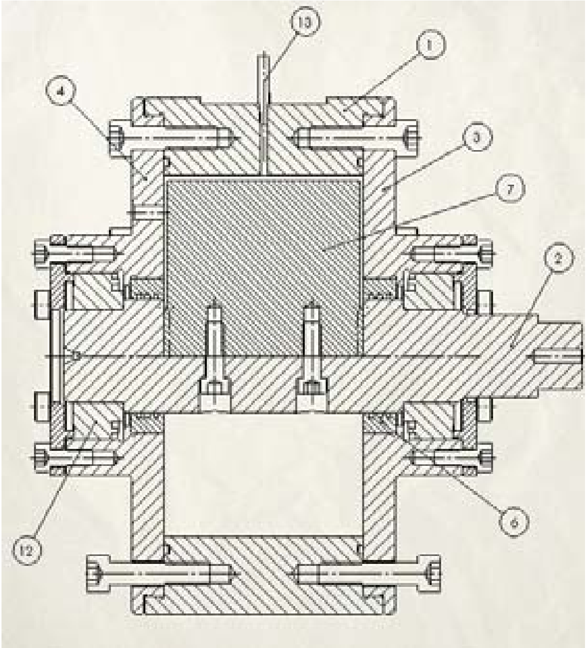


Fig. 27. The 3D-view of seals testing stand



1 - Cylinder
2 - Shaft
3 - Front cover
4 - Rear cover
6 - Axial labyrinth packing
7 - Blade with labyrinth packing
12 - Bearing
13 - Pressure sensor

Fig. 28. Cross-section of single compression machine with above discussed sealing devices

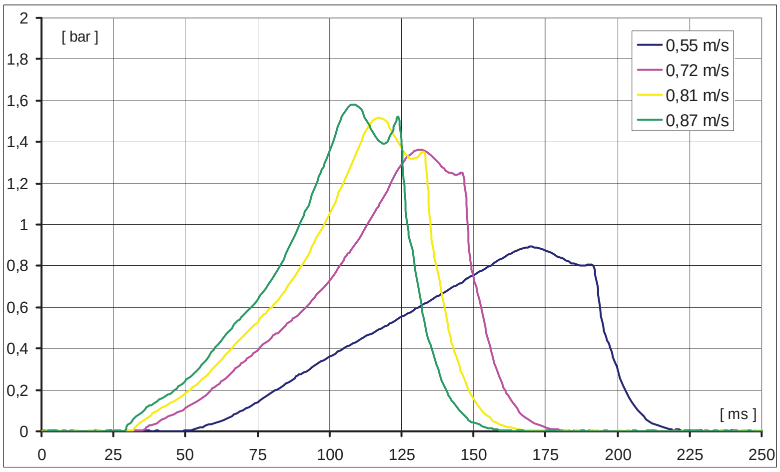


Fig. 29. Comparison of averaging pressure curves for labyrinth seals on piston (blade); the shaft sealing was labyrinth type as well

References

[1] Chlumsky, V., *Spężarki łokowe*, PWT, 1961.

[2] Cyklis, P., Żelasko, J., *Optymalizacja przepływu czynnika w samoczynnych zaworach sprężarek tłokowych*, Materiały XVII Zjazdu Termodynamików, Zakopane 1999.

[3] Cyklis, P., *Model matematyczny sprężarki tłokowej wyposażonej w samoczynne zawory płytkowe*, Materiały XIV Zjazdu Termodynamików, wyd. AGH, pp.98-105, Kraków 1990.

[4] Forsthoffer, W., *Compressors*, Elsevier, 2005.