

Modelling the load of the piston system in an axial piston pump by means of the adina software

Tadeusz Zloto, Piotr Stryjewski

Institute of Mechanical Technologies
Częstochowa University of Technology

Received April 11.2013; accepted June 18.2013

Summary. The paper analyses the load of the kinematic pair piston-cylinder block in an axial piston pump. The surface compressive stress occurring between the piston and the cylinder block were determined by means of the commercially available software Adina, which was used for constructing numerical models and representing results of the research.

Key words: axial piston pump, piston load, software Adina.

INTRODUCTION

Axial piston pumps have numerous industrial applications. Since they can operate by high pressures and power, they are characterized by high values of power efficiency, defined as the ratio of power to mass or volume [11, 12, 19]. Axial piston pumps are most often used in the drives of complex devices requiring high efficiency [13, 20]. Needless to say, attempts are constantly made at improving the design and exploitation parameters of pumps to further increase their efficiency and reliability [6, 7, 9, 10, 14, 17, 18, 21].

As was mentioned, axial piston pumps have a constantly growing range of industrial applications, including such branches as

- aircraft industry,
- automotive industry (presses, CNC systems, injection molding machines),
- heavy industry (pressure foundries, rolling machines, cokeries),
- building industry (excavators, loaders, extension arms),
- agriculture and forestry (cranes, elevators, grill rigs, mowers, harvesters),
- military vehicles (multifunction vehicles, constructing bridges).

LOAD OF A PISTON IN AN AXIAL PISTON PUMP

From among all hydraulic machines, the highest pressures are attested in axial piston pumps [1, 8, 22]. High pressures cause high loads of the basic kinematic pairs in these machines. Fig. 1 presents forces acting on the piston system in an axial piston pump.

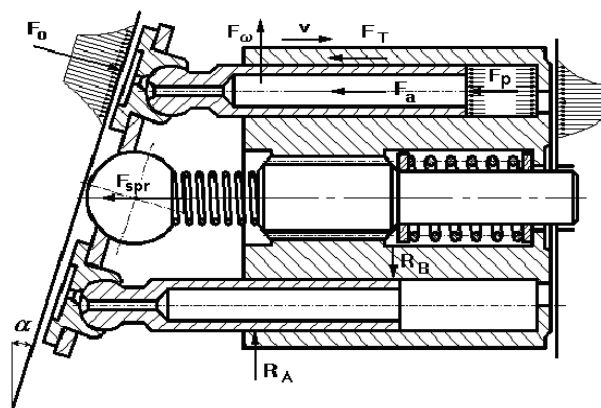


Fig. 1. Load of an axial piston pump

The following forces can be observed: Force F_p originating from pressure in the displacement chamber, dynamic force F_a of the piston system, centrifugal force F_w and friction F_T . The friction force F_T operates between the cylinder and the piston when the latter is moving in the course of the work. In the pump work cycle, friction increases load of the valve plate, but in the motor work cycle it causes relief of the valve plate.

One of the main sources of energy losses in axial piston pumps is the system piston-cylinder block. The two types of cylinder lining and the corresponding piston motions are presented below [3, 5, 16]:

- piston travels in a sleeveless cylinder (Fig. 2a),
- piston travels in a sleeved cylinder (Fig. 2b).

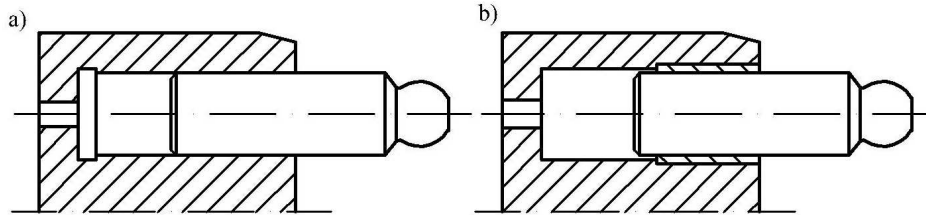


Fig. 2. Piston moving in: a) sleeveless cylinder, b) sleeved cylinder

ANALYSIS OF STRESS FORCES
IN THE PISTON SYSTEM

In the piston-cylinder system in axial piston pumps operating on the wash plate principle, a radial force F_{wy} coming mostly from the force of displacement pressure acts on the piston joint. This causes the piston to skew, and subsequently, significant reactions R_A, R_B occur between the piston and the cylinder.

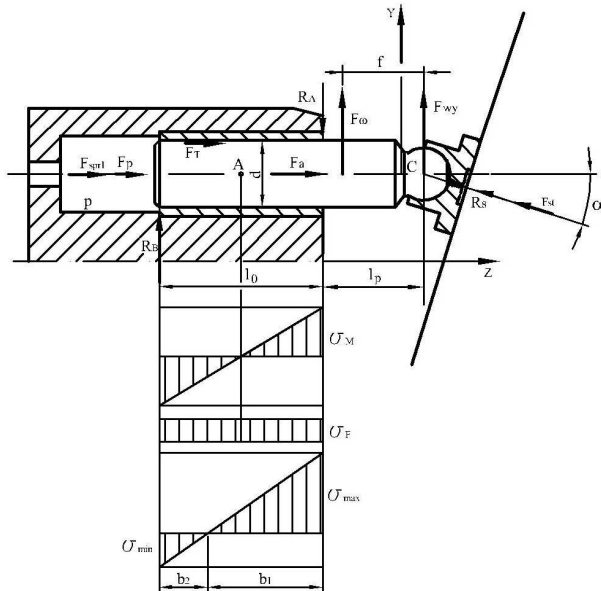


Fig. 3. Computational model of surface compressive stress

The surface compressive stress coming from the radial force are represented by the following formula [2, 4, 19]:

$$\sigma_F = \frac{F_{wy}}{d \cdot l_0}, \tag{1}$$

and the surface compressive stress coming from the torque of the radial force are represented as [4]:

$$\sigma_M = \frac{6 \cdot [F_{wy} \cdot (l_p + \frac{1}{2} l_0)]}{d \cdot l_0^2}. \tag{2}$$

The maximum surface compressive stress between the piston and the cylinder are [4, 19]:

$$\sigma_{max} = \sigma_M + \sigma_F. \tag{3}$$

Fig. 4 presents the maximal and minimal values of surface compressive stress as the function of the angle ϕ of the cylinder block rotation in the model under consideration.

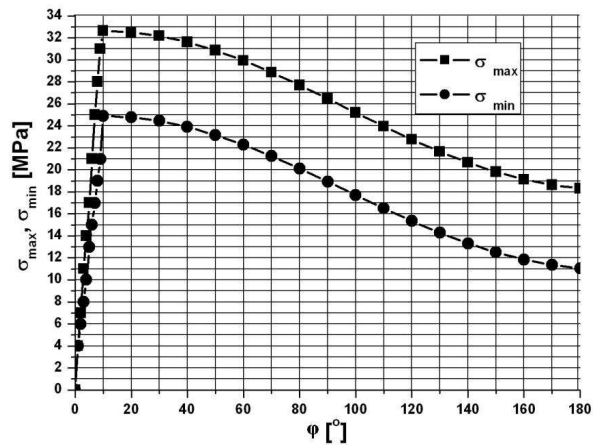


Fig. 4. Maximal and minimal surface compressive stress as the functions of the angle ϕ of the cylinder block rotation

MODELLING THE PISTON-CYLINDER
CONTACT ZONE

In the study, the ADINA (Automatic Dynamic Incremental Nonlinear Analysis) system was used to simulate numerically the effective stress distribution of the kinematic pair piston-cylinder in an axial piston pump. The software is designed as a dedicated tool for solving problems related to fluid mechanics and heat flow.

Fig. 5 presents the algorithm of the simulation subsequently applied in the Adina system.

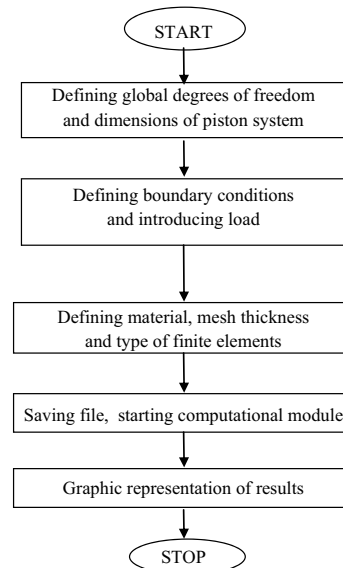


Fig. 5. Algorithm of the simulation in the Adina system

The piston-cylinder contact is analysed as a case of flat deformation in a two-dimensional space. Such a model includes some simplifications, but on the other hand, it offers important advantages, such as simplicity of modelling, compact size, and consequently, short computation time. Therefore, the model proves to be useful for testing computations related to various problems.

Fig. 6 presents the discretization region of the piston system.

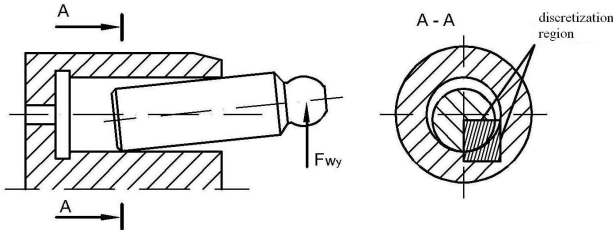


Fig. 6. Discretization region

The discretization uses 4-nod elements of the type 2D-SOLID.

Fig. 7 presents a mesh of the two-dimensional model of the steel piston-steel cylinder system.

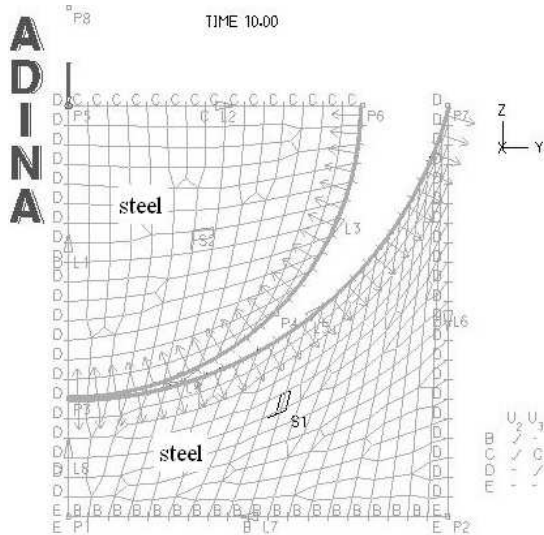


Fig. 7. Two-dimensional model mesh (steel-steel)

Fig. 8 presents smoothed field of effective stress in the contact zone, obtained for the piston-cylinder system.

Fig. 9 shows a mesh of the two-dimensional model of the steel piston-bronze cylinder system.

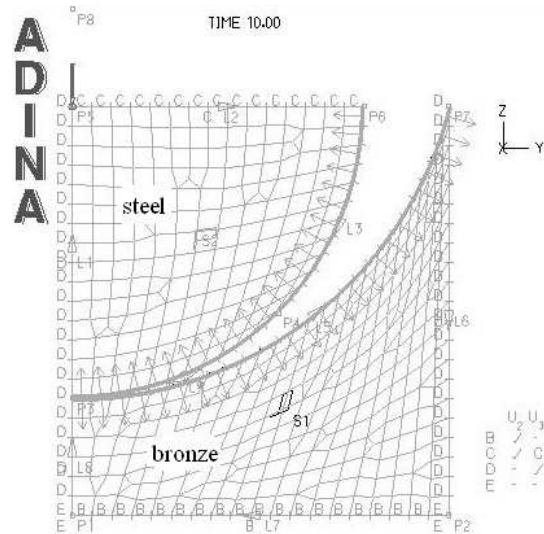


Fig. 9. Two-dimensional model mesh (steel-bronze)

Fig. 10 presents smoothed field of effective stress in the contact zone, obtained for the steel piston-bronze cylinder system.

Table 1 presents selected stress values obtained in the piston-cylinder contact zone in an axial piston pump for the analytic model not taking into account the material of which the parts are made and the numerical models of the steel piston-steel cylinder system and the steel piston-bronze cylinder system.

The absolute differences in unitary stress were obtained for the numerical models and the analytic model with respect to the analytic model. The relative values for the numerical model in the steel piston-steel cylinder case are 1.6 – 13.6 %, whereas in the case steel piston-bronze cylinder the relative values are 1.9 – 8.7% [15].

The study carried out therefore lends support to the thesis that the construction with a steel piston and a bronze cylinder should be preferred over the alternative construction of a pump in which both the piston and cylinder are made of steel.

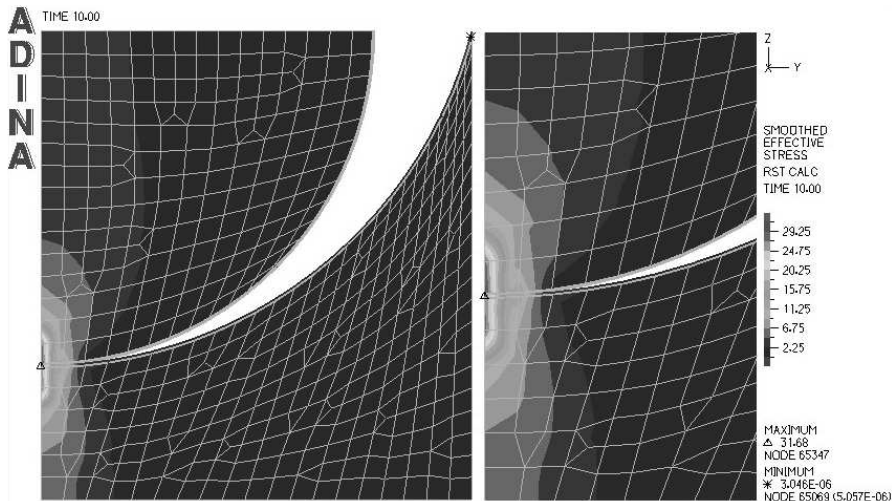


Fig. 8. Huber-von Mises smoothed effective stress distribution in the contact zone, obtained for the steel piston-steel cylinder system

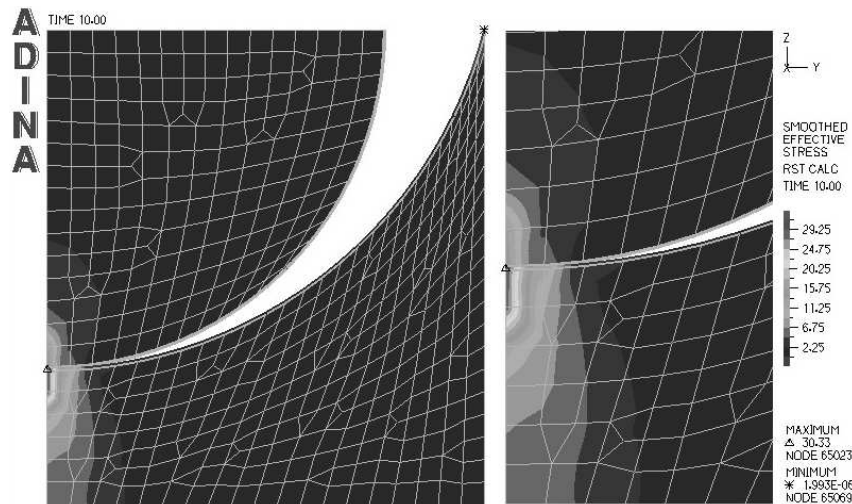


Fig. 10. Huber-von Mises smoothed effective stress distribution in the contact zone, obtained for the steel piston-bronze cylinder system

Table 1.

No.	Cylinder block rotation angle φ [°]	Radial force F_{wp} [N]	Analytic model σ [MPa]	Numerical models			
				Steel piston and cylinder σ [MPa]	Relative values [%]	Steel piston and bronze cylinder σ [MPa]	Relative values [%]
1	0	1864.11	32.21	31.68	1.6	30.33	5.8
2	45	1835.63	30.42	31.18	2.5	29.85	1.9
3	90	1774.53	26.55	30.15	13.6	28.86	8.7

CONCLUSIONS

On the basis of the present study, the following conclusions can be put forward:

1. The numerical models developed with the use of the software Adina enable determining the values of stress forces in the kinematic pair piston-cylinder of an axial piston pump.
2. In the case when the piston is made of steel and the cylinder block is made of bronze the surface compressive stress are significantly smaller than in the case when both the piston and the cylinder are made of steel. Supposedly it is caused by different strain and material properties of steel and bronze.

REFERENCES

1. **Balas W. 1966:** Łożyska hydrostatyczne w osiowych pompach tłokowych. Przegląd Mechaniczny, Nr 11, 329-331.
2. **Bronsztejn I.N., Siemiendajew K.A., Musiol G., Mühlig H. 2004:** Nowoczesne compendium matematyki. WN PWN, Warszawa.
3. **Heyl W. 1979:** Ermittlung der optimalen Kolbenanzahl bei Schrägscheiben-Axialkolbeneinheiten. Ölhydraulik und Pneumatik, 23, Nr 1, 31-35.
4. **Ivantsyn J., Ivantsynova M.:** Hydrostatic Pumps and Motors. Akademia Books International, New Delhi 2001.
5. **Jang D.S. 1997:** Verlustanalyse an Axialkolbeneinheiten. Dissertation RWTH, Aachen.
6. **Kögl Ch. 1995:** Verstellbare hydrostatische Verdrängereinheiten im Drehzahl- und Drehmomentregelkreis am Netz mit angepaßtem Versorgungsdruck. Dissertation RWTH, Aachen.
7. **Kraszewski D. 1975:** Dobieranie parametrów konstrukcyjnych łożyska hydrostatycznego o powierzchni płaskiej i sferycznej w zastosowaniu do maszyn tłokowych. Rozprawa doktorska. Pol. Wrocławska, Wydział Mechaniczny, Wrocław.
8. **Murrenhoff H. 2005:** Grundlagen der Fluidtechnik. Teil 1: Hydraulik, Shaker Verlag, Aachen.
9. **Niegoda J. 1978:** Badanie możliwości zastosowania tłoków z bezprzegubowym podparciem hydrostatycznym w pompach i silnikach wielotłoczkowych osiowych. Rozprawa doktorska. Pol. Gdańska, Wyzd. Budowy Maszyn, Gdańsk.
10. **Olems L. 1999:** Berechnung des Spaltes der Kolben-Zylinderbaugruppe bei Axialkolbenmaschinen. Ölhydraulik und Pneumatik, 43, Nr 11-12, 833-839.
11. **Osiecki A. 1998:** Hydrostatyczny napęd maszyn. WNT, Warszawa.
12. **Орлов Ю.М. 1993:** Авиационные объемные гидромашины с золотниковым распределением. ПГТУ, Пермь.
13. **Osiecki L. 1999:** Badanie zjawisk zachodzących w zespolu tłoczek-stopka hydrostatyczna-dławik śrubowy

- maszyny wielotłoczkowej osiowej. Rozprawa doktorska. Pol. Poznańska, Wydz. Budowy Maszyn i Zarządzania, Poznań.
14. **Osiecki A., Osiecki L. 1998:** Prace rozwojowe nad nową konstrukcją pomp wielotłoczkowych osiowych. *Hydraulika i Pneumatyka*, Nr 4, 4-9.
 15. **Paleczek W. 2003:** *Mathcad Professional*. Akademicka Oficyna Wydawnicza EXIT, Warszawa.
 16. **Полозов А.В. 1976:** Выбор оптимальных параметров объемного роторного насоса. *Вестник Машиностроения*, № 1, 3-8.
 17. **Renius K.H. 1973:** Experimentelle Untersuchungen an Gleitschuhen von Axialkolbenmaschinen. *Ölhydraulik und Pneumatik*, 17, Nr 3, 75-80.
 18. **Renius K.H. 1975:** Reibung zwischen Kolben und Zylinder bei Schrägscheiben-Axialkolbenmaschinen. *Ölhydraulik und Pneumatik*, 19, Nr 11, 821-826.
 19. **Stryczek S. 1995:** Napęd hydrostatyczny. Tom 1, WNT, Warszawa.
 20. **Ryzhakov A., Nikolenko I., Dreszer K. 2009:** Selection of discretely adjustable pump parameters for hydraulic drives of mobile equipment. *Polska Akademia Nauk, Teka Komisji Motoryzacji i Energetyki Rolnictwa*, Vol. IX, 267-276.
 21. **Złoto T., Stryjewski P. 2012:** Load of the kinematic pair piston-cylinder block in an axial piston pump, *PAN, TEKA Komisji Motoryzacji i Energetyki Rolnictwa*, Vol. 12, No 2, 269-274.
 22. **Zhang Y. 2000:** Verbesserung des Anlauf- und Langsamlaufverhaltens eines Axialkolbenmotors in Schrägscheibenbauweise durch konstruktive und materialtechnische Maßnahmen. *Dissertation RWTH, Aachen*.

MODELOWANIE OBCIĄŻENIA ZESPOŁU TŁOCZKA
POMPY WIELOTŁOCZKOWEJ Z WYKORZYSTANIEM
PROGRAMU ADINA

Streszczenie. Praca zawiera analizę obciążenia pary kinematycznej tłoczek-cylinder pompy wielotłoczkowej osiowej. Do określenia nacisków powierzchniowych występujących pomiędzy tłoczkiem i cylindrem w pompie wielotłoczkowej wykorzystano komercyjny program komputerowy Adina. Zbudowano modele numeryczne w tym programie i zaprezentowano wyniki badań. **Słowa kluczowe:** pompa wielotłoczkowa osiowa, obciążenie tłoczka, program Adina.