

## Alternative FEM Algorithm of Determining Piston Ring Pressure Distribution on a Cylinder to a Contact Simulation

Marcin Kaliszewski, Paweł Mazuro

Institute of Heat Engineering, Warsaw University of Technology  
 Nowowiejska 21/25, 00-665 Warsaw, Poland,  
 email: markaliszewski@gmail.com, pawel.mazuro@itc.pw.edu.pl

Received November 17.2016; accepted December 21.2016

**Summary.** An alternative FEM algorithm of finding piston ring pressure distribution to a contact simulation is introduced. The method is basing on an analytical determining of required nodal displacement boundary conditions. Its several configurations are tested using APDL and compared to a no-separation contact simulation of a simple 2D finite element model of a two-stroke piston ring made of Titanium alloy. Each of the methods tested in the paper brings displacement result and Huber-Misses equivalent stresses close to each other. However, only one of those brings resulting contact pressure close to a no-separation contact simulation. Nonetheless, the obtained configuration occurred to be less computationally efficient than no- separation contact simulation performed in an ANSYS software.

**Key words:** Finite Element Method, alternative algorithm to contact simulation, pressure distribution, APDL, two-stroke piston ring.

tested piston ring is in fact conical with 15° to 30° inclination and a rectangular gap. However, the role of its conical shape is only to allow faster lapping process take place (for details, see [9, 11, 17]) and, therefore, is neglected in the FEM model which is two-dimensional. Due to a symmetry of the ring, only half of it is modelled and symmetry boundary constraint on the left hand side of the model is applied. Since the aim of following simulations is not to test a geometry of a ring but methods of determining its pressure distribution, it was chosen to divide geometry into only 80 elements along circumference of a ring. The material for the ring was chosen to be a titanium alloy. Ring properties are briefly outlined in Table 1. Every of performed simulation is compared to a no separation contact solution (for details, see [12, 13, 14, 16]) with no friction and penetration tolerance of a constant value equal to  $10^{-8}$ .

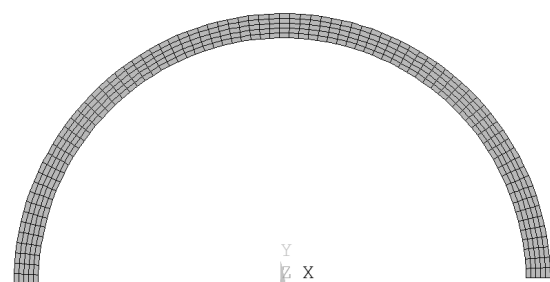
### INTRODUCTION

The aim of this paper is to discuss a certain method of determining pressure distribution for piston ring of an arbitrary geometry which would be faster and more stable than a standard contact simulation. Those properties are requested in case of optimisation process. In this paper an alternative method to contact simulation is proposed. The results of its testing on the basis of a simple 2D model are presented in the second section of the paper. Altogether, there were considered three configurations of the method which are briefly described in the subsequent section. However, only one of them turned out to bring the results close to the ones obtained by means of a no separation contact simulation. Conclusions are delivered in the last section of the paper.

Methods discussed in this research were tested by using the geometry of a ring defined by outer nodes obtained from C++ program written by R. Smoliński [15]. The ring is fully modelled by 2D PLANE182 quadrilateral elements [7] in plain stress state as shown in Figure 1. The shape of the

**Table 1.** Ring properties

Width	g	2.5 mm
Depth	h	1 mm
Deflected ring diameter	D	55 mm
Young's modulus	E	625 GPa
Poisson ratio	$\nu$	0.21



**Fig. 1.** Tested ring mesh

ALTERNATIVE FEM METHOD

LINEAR METHOD

An alternative to standard contact methods bases on a simple displacement constraint in a radial coordinate system and treats the model as a standard linear simulation. Since a cylinder has a round shape, it is possible to compute required displacement by formula (1), where  $x_i$  and  $y_i$  are nodal coordinates. However, such a method generates inaccuracy due to ANSYS formulation of such a constraint. Radial nodal coordinate system in fact is not really a radial coordinate system but a Cartesian system aligned to a radial direction ( $x$  nodal coordinate is collinear with a line between a node and a centre of global coordinate system). Because of that reason a node can move along a certain straight line instead of moving along a circle as it is visualised in Figure 2 (for details, see [5, 6, 7, 8]). Due to such a phenomenon, the most inaccurately displaced node after tested linear simulation is  $\varepsilon_R = 0.077 \text{ mm}$  far from the circumference of a circle which stands at 0.28% of the circle radius (as it is presented in Figure 3).

$$\Delta u_i = (x_i^2 + y_i^2)^{0.5} - R. \quad (1)$$

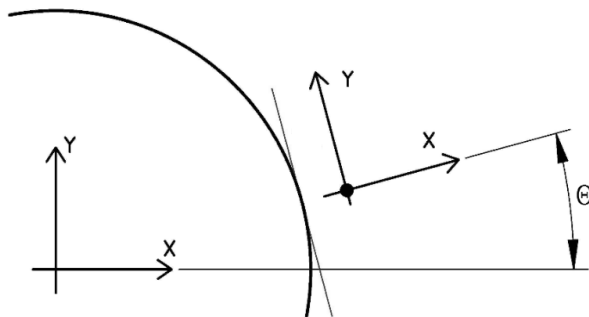


Fig. 2. Scheme of the first step of alternative method

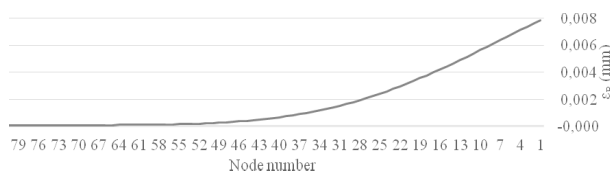


Fig. 3. Distance of displaced nodes from the cylinder circle

Inaccuracies of the position of such values have a huge impact on the pressure distribution. It results in an inaccuracy of 112% in the resulting contact pressure values compared to a no separation contact solution of the same geometry. Resulting pressures for both geometries are presented in Figures 4 and 5. The difference between pressure distributions is presented in Figure 6. The contact pressure distribution in both cases is computed via APDL commands by reading nodal forces and dividing them by associated areas as described in equations (2-3) (see also [10]). In those formulae  $l_i$  and  $l_{i+1}$  stand for the length along cylinder circumference of the two nearest elements to the node after it is deflected in a simulation. For the first and last node  $l_i$  and  $l_{i+1}$  were substituted by zeros respectively, because they are associated with one element only.

$$p_i = F_i \cdot A_i^{-1}, \quad (2)$$

$$A_i = (l_i - l_{i+1}) \cdot h \cdot 2^{-1}. \quad (3)$$

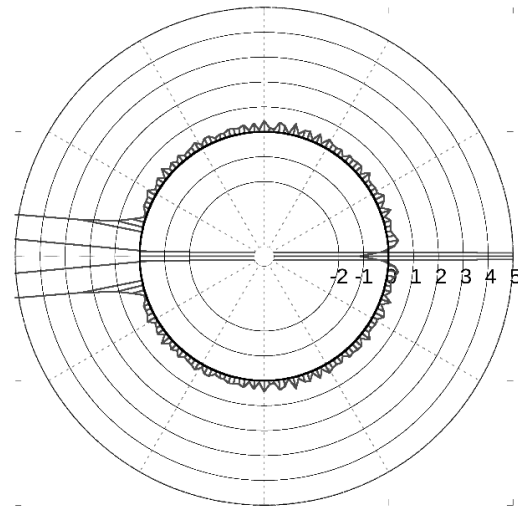


Fig. 4. Pressure distribution for the linear simulation (MPa)

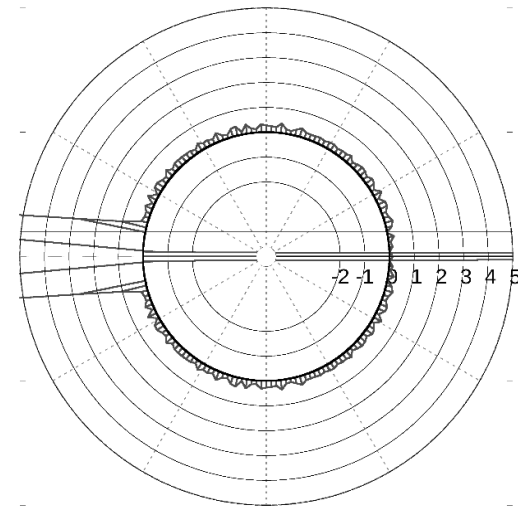


Fig. 5. Pressure distribution for the contact simulation (MPa)

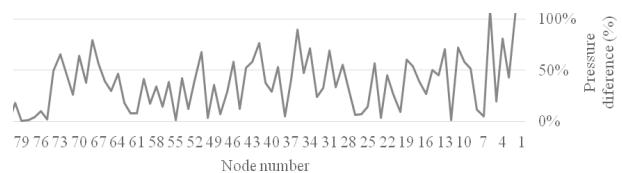


Fig. 6. Pressure difference between contact simulation and the first step of the linear method

The distribution of pressure may be considered as atypical for a two-stroke ring. It comes from a mistake while using R. Smoliński program, however, was accepted by the authors for the purpose of this paper which is a test of the method of determining contact pressures.

TWO NEXT STEPS OF LINEAR METHOD

In order to improve the algorithm, it was decided to perform the two next steps of the simulation. Each of these steps consists of aligning nodal coordinate system to a radial

coordinate of a node in the position after a previous step of simulation, as it is shown in Figure 7. The angle of rotation of nodal coordinate system is computed by formulae (4-5). Since a nodal coordinate system changes its rotation it is also required to change value of nodal constraints. They may be computed from formula (6). Thanks to such modification, the node is allowed to move along another straight line which is much closer to the proper localisation.

$$\theta_i = \tan^{-1}[(y - \Delta y_i) \cdot (x - \Delta x_i)^{-1}] \text{ for } \theta \in (0^\circ, 45^\circ) \cup (135^\circ, 180^\circ), \quad (4)$$

$$\theta_i = \pi \cdot 2^{-1} - \tan^{-1}[(x - \Delta x_i) \cdot (y - \Delta y_i)^{-1}] \text{ for } \theta \in (45^\circ, 135^\circ), \quad (5)$$

$$\Delta u_i = |-y_i \cdot \sin(\theta_i) - x_i \cdot \cos(\theta_i)| - R. \quad (6)$$

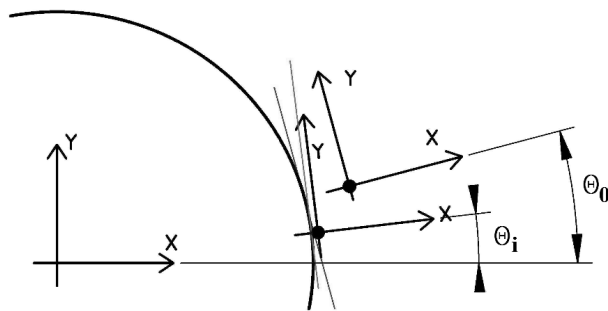


Fig. 7. Scheme of further steps of alternative method

After 3 first steps of the algorithm, maximal simulation misalignment of a node dropped to an acceptable value of only  $\varepsilon_R = 7,11 \cdot 10^{-15} \text{ mm}$ . Contact pressure changed as well, however, it still differed from the contact simulation and the differences were far from being acceptable as it is shown in Figure 8.

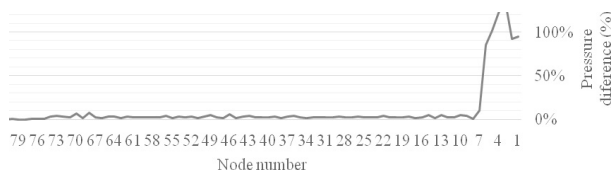


Fig. 8. Pressure difference between contact simulation and the third step of the linear method

NONLINEAR METHOD

The same algorithm was utilized once more, yet, with nonlinear solver settings with an automatic number of substeps (for details of nonlinear analysis, see [2, 3, 4, 14]). The number of substeps can vary from 5 to 1000 and is 10 substeps by default. The resulting pressure distribution improved significantly. The difference in the pressure distribution between the contact simulation and three-step nonlinear algorithm dropped to 0.41% as presented in Figure 9. FEM model results of nodal displacements and Huber-Mises equivalent stresses are presented in Figures 10-11.

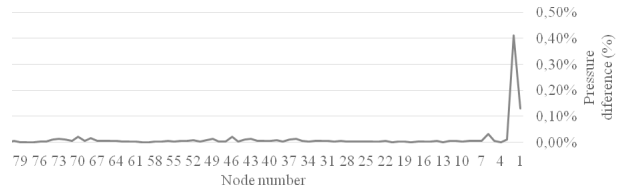


Fig. 9. Pressure difference between contact simulation and the third step of the nonlinear method

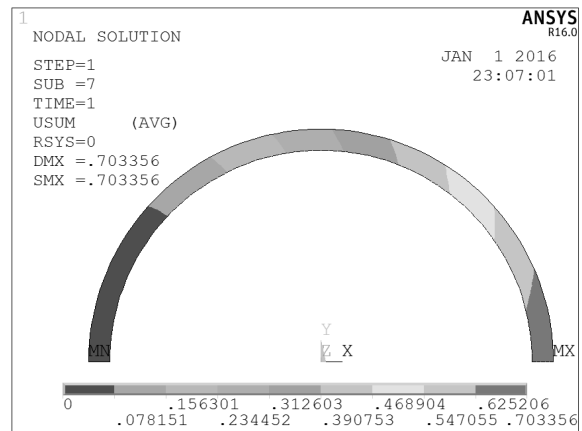


Fig. 10. Displacement of a ring (mm) (3<sup>rd</sup> step of nonlinear sim.)

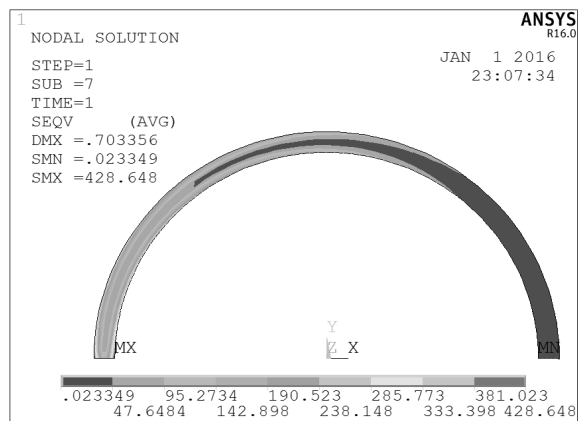


Fig. 11. Equivalent von Mises stresses (MPa) (3<sup>rd</sup> step of nonlinear sim.)

## CONCLUSIONS AND POSSIBLE LINE OF FURTHER IMPLEMENTATION

All of the algorithms have a common disadvantage – they can bring positive as well as negative resulting pressures. In case of a piston ring, it is not physical and could deliver completely inaccurate results. However it hopefully could be neglected in case of inter alia some optimisation algorithms (e. g. Simulated Annealing Method [1, 10, 18]), where negatives pressure values are far from the optimal distribution and lead to high objective function values.

All of the above-considered methods lead to similar nodal displacements and equivalent von Mises stresses. The differences between them are not higher than 1.2% and 0.06%, respectively. However, they result in a huge inaccuracy of ring pressure distributions. Only 3 steps of nonlinear simulation and no-separation contact simulation brought the same pressure distribution (with the maximum difference of 0.41%). In case of the last two simulations, the differences in nodal displacements and equivalent von Mises stresses are lower than 0.001%.

The three steps of a nonlinear simulation take approximately 150% more time to obtain a solution than a no-separation contact simulation of the same model. It could be explained by 3 repetitions of a nonlinear solution (in case of contact simulation it is only 1 solution) and a significant number of APDL loops and equations which are not adjusted to parallel computing by an ANSYS software.

The generated algorithm could be used both in 2D cases and in 3D ones, even for some complex cross-section and lock geometry. In the case of 3D geometry, possible displacement of nodes would be constrained to planes rather than lines as it was in the case of the 2D model. The equations which were presented in the paper would be valid for a 3D case, however, they have not been tested yet.

## ACKNOWLEDGMENTS

The presented work is a part of the research which received funding from the Polish-Norwegian Research Program operated by the National Centre of Research and Development under the Norwegian Financial Mechanism 2009-2014 in the frame of Project Contract No Pol-Nor/199058/94.

## REFERENCES

1. **Bertsimas D. and Tsitsiklis J. 1993** Simulated annealing *Statistical Science* **8** 10-5
2. **Brzoska Z. 1979** *Wytrzymałość materiałów* (Warszawa: PWN)
3. **Bijak-Żochowski M., Jaworski A., Krześciński G. and Zagrajek T. 2004** *Mechanika Materiałów i Konstrukcji tom 1* (Oficyna Wydawnicza Politechniki Warszawskiej)
4. **Bijak-Żochowski M., Jaworski A., Krześciński G. and Zagrajek T. 2004** *Mechanika Materiałów i Konstrukcji tom 2* (Oficyna Wydawnicza Politechniki Warszawskiej)
5. ANSYS Inc. April 2009 *Programmer's Manual for Mechanical APDL*
6. ANSYS Inc. November 2009 *Contact Technology Guide*
7. ANSYS Inc. November 2009 *Element Reference*
8. ANSYS Inc. November 2013 *ANSYS Mechanical APDL Command Reference*
9. **Iskra A. 1996** *Studium konstrukcji i funkcjonalności pierścieni w grupie tłokowo-cylindrowej* (Poznań: Wydawnictwo Politechniki Poznańskiej)
10. **Kaliszewski M. and Mazuro P. 2016** *Analysis of optimisation method for a two-stroke piston ring using the Finite Element Method and the Simulated Annealing Method* (preprint)
11. **Kozaczewski W. 2004** *Konstrukcja grupy tłokowo-cylindrowej silników spalinowych* (Warszawa: Wydawnictwa Komunikacji i Łączności)
12. **Nackenhurst U. and Wriggers P. 2006** *Analysis and Simulation of Contact Problems* Springer
13. **Reddy J. 2005** *An Introduction to the Finite Element Method* McGraw-Hill Education
14. **Reddy J. 2015** *An Introduction to the Nonlinear Finite Element Analysis* Oxford University Press
15. **Smoliński R. 2013** *Projekt i optymalizacja wielopierścieniowego uszczelnienia tłoka* (Master's Thesis in Faculty of Power and Aeronautical Engineering, Warsaw University of Technology)
16. **Taylor R.L. and Zienkiewicz O.C. 2000** *The Finite Element Method vol 1* Butterworth-Heinemann
17. **Wajand J.A. and Wajand J.T. 2005** *Tłokowe silniki spalinowe średnio- i szybkoobrotowe* (Warszawa: Wydawnictwo Naukowo Techniczne)
18. **Yang X-S. 2010** *Engineering Optimization: An Introduction with Metaheuristic Applications* (United Kingdom: John Wiley and Sons)

**Streszczenie.** W pracy przedstawiono alternatywne użycie Metody Elementów Skończonych w celu wyznaczania rozkładu nacisków pierścienia tłokowego na gładź cylindra. Metoda ta bazuje na analitycznym wyznaczeniu wymaganych warunków brzegowych dla danych węzłów. Kilka konfiguracji danej metody zostało przetestowanych w języku APDL i porównanych do symulacji kontaktu bez separacji na przykładzie prostej dwuwymiarowej geometrii pierścienia dwusuwowego wykonanego ze stopu tytanu. Każda z przetestowanych w pracy konfiguracji przyniosła zbliżone wyniki przesunięć i naprężeń zredukowanych, jednak tylko jedna z testowanych metod zwróciła rozkład nacisków powierzchniowych bliski do symulacji kontaktu bez separacji. Ostatecznie otrzymana metoda okazała się być mimo wszystko mniej wydajną obliczeniowo od metody kontaktu bez separacji zaimplementowanej w programie ANSYS.

**Słowa kluczowe:** Metoda Elementów Skończonych, alternatywny algorytm symulacji kontaktu, rozkład nacisków, APDL, pierścień dwusuwowy.