Analysis of end-face seals by FEM

J. Juraszek [1]

Abstract: *The end face-seals joint simultaneously the function of seal, bearing and heat exchanger. In order to improve the design of end-face seals the analysis of this type of joint by means of finite element method, was developed. The temperature distribution is indispensable to the analysis of displacement and stress in this type of face seal. In this work the values of temperature and stress in end-face seals were calculated. The numerical results were confirmed by measurements in the laboratory.*

Introduction

In technical applications of the measurements of temperature in many devices are possible only in selected points. The problems of determining of temperature distribution are very important particularly in metal-ceramic joints, which we can meet in end-face seals. This problem is often encountered in various pumps of used in car and power industry.

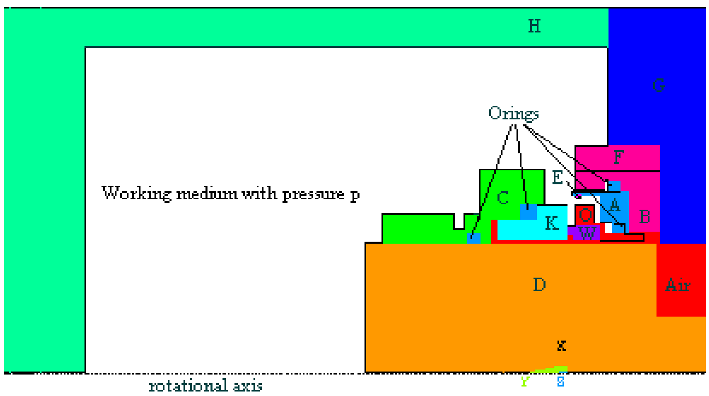
The subject of analysis is the end-face seal, which is numbered among the second loading group in the range of working medium pressure (included in the range of 0,1 to 1,0 MPa) as well as the third loading group taking into consideration the average rotational speed of one of rings [1], [2]. When determining the pressure and rotational speed (it can be done without any problem) the determination of distribution of temperatures in a free (any) element of seal encounters significant difficulties because the temperature measurement is possible to carry out only in selected points. So it is impossible to determine distribution of temperatures in seal, especially in a slide pair.

The aim of our work is to determine the temperature and strain distribution in end-face seals in order to avoid a leakage in this type of joint.

The problem of determining of temperature distribution is essential for ceramic-metals joints occurring in end-face seals. As a result of face seals, which are constructed on the basis of different materials with different mechanical and thermal properties, a thermal stress can appear. Regarding the increasing accuracy of numerical simulations, we performed special data base of materials parameters. This data can draw proper material data from tensile curves stored by Instron tensile machine and automatically convert to the integrated input FEM method.

Features of the numerical model

The end-face seal can be performed as an axisymmetrical problem. In this work, the numerical model of end-face seal is discretized by finite elements [3]. The model has over 3800-node, 3560 continuum elements (plane 42, 13, contact 48) and 3793 degrees of freedom were employed.



The distribution of temperature and stress were modeled by a thermal solid element and coupled-field solid element taken from the Ansys System FE. Fig. 1 shows a model of testing stand chamber with the development seal.

Fig.1 Seal elements development in the testing stand chambe r – a calculation model.

- K, W, O, A, C, E - denote components of elements of seal;
- B, D, F, G, H - denote components of elements of testing stand chamber.

[1] *J. Juraszek*, University Bielsko-Biała, Willowa 2, 43-309 Bielsko-Biała, Poland.
(Reviewed, revised version received August, 14 2002)

Elements of slide pair were made of carbon-silicon composite. The other seal elements were made of stainless steel [4]. For example, in table 1 are the values of material parameters.

Tab.1 Material parameters used in the end-face-seal.

Physical quantity	Unity	Moving ring	Immobile ring	Other elements
Density	[kg.m^{-3}]	3100	1800	7900
Young's modulus	[MPa]	3,95.10^{5}	10,468.10^{3}	1,93.10^{5}
Poisson ratio		0,17	0,22	0,3
Thermal conduction coefficient	[K^{-1}]	4,5.10^{-6}	4,5.10^{6}	17.10^{-6}
Thermal expansion coefficient	[W.m^{-1}.K^{-1}]	100	20	15,1
Specific heat	[J.kg^{-1}.K^{-1}]	1000	750	510

Boundary and loads conditions

As an initial condition, temperature of 20°C was assumed for the whole model. Boundary conditions were determined on the basis of measurements conducted on the testing stand. The measurement of temperature was carried out in the selected points of testing chamber and connected with elements of the power unit. Depending on the place in which a measurement was performed, temperature was evaluated by using thermocouples or a laser instrument.

On the basis of the obtained results, the following boundary conditions were formulated:
To the 1 st. type boundary conditions (Dirichlet's condition), applied in the analysed system, belong:
- temperature of driving shaft, at the development seal,
- temperature inside of the chamber frame on the section between bearings and the seal.

A 3 rd. type of boundary condition was the Fourier's condition (fig.2), that is a relation between the temperature of fluid and its boundary derivative:

$$-\lambda \left.\frac{\partial t}{\partial n}\right|_b = \alpha\left(t_b - t_o\right) \tag{1}$$

where:
λ - thermal conduction coefficient, α - convective heat-transfer coefficient,
t_b - temperature of the boundary area, t_o - temperature of fluid.
$\left.\frac{\partial t}{\partial n}\right|_b$ - derivative of temperature of the boundary area.

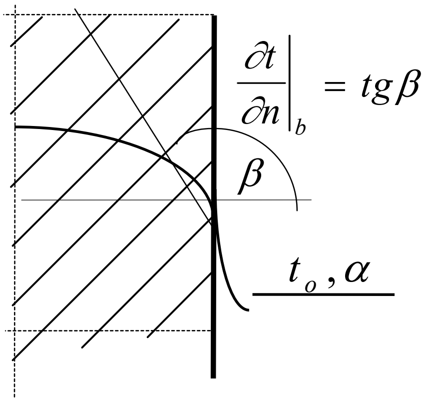


Fig.2 Diagram of 3 rd type boundary conditions [8].

One of indispensable elements to formulate above boundary conditions is a knowledge of temperatures
- of the working medium,
- of air (surrounding the testing chamber).

The temperature of working medium was measured at the distance of 40 mm from the side of the chamber. The measurements were carried out when the temperature of the medium was held on the same level for 30 minutes. The temperature of air surrounding the testing chamber was measured 1 m from its cylindrical surface. For the used testing chamber, it was impossible to determine convective heat-transfer coefficients for surfaces contacted with the working medium as well for external surfaces of the chamber testing blown by air.

To determine convective heat-transfer coefficients, criterial equations were used which were determined experimentally for different kinematics conditions performed in work [5], [6], [7]. Surfaces blown by the working medium, for which convective heat-transfer coefficients are determined, are performed in fig.3.

The mean value of convective heat-transfer coefficients for external surfaces of testing chamber was calculated and its value equals to α_z=5,2 $\left[\frac{W}{m^2K}\right]$.

In the air-gap between the rotated shaft and the seal elements, the thermal conduction coefficient was assigned taking into account a convection action and radiation in this place.

In table 2 are given the values of material parameters of convective heat-transfer coefficients [8] whose values correspond to surfaces denoted in fig. 3.

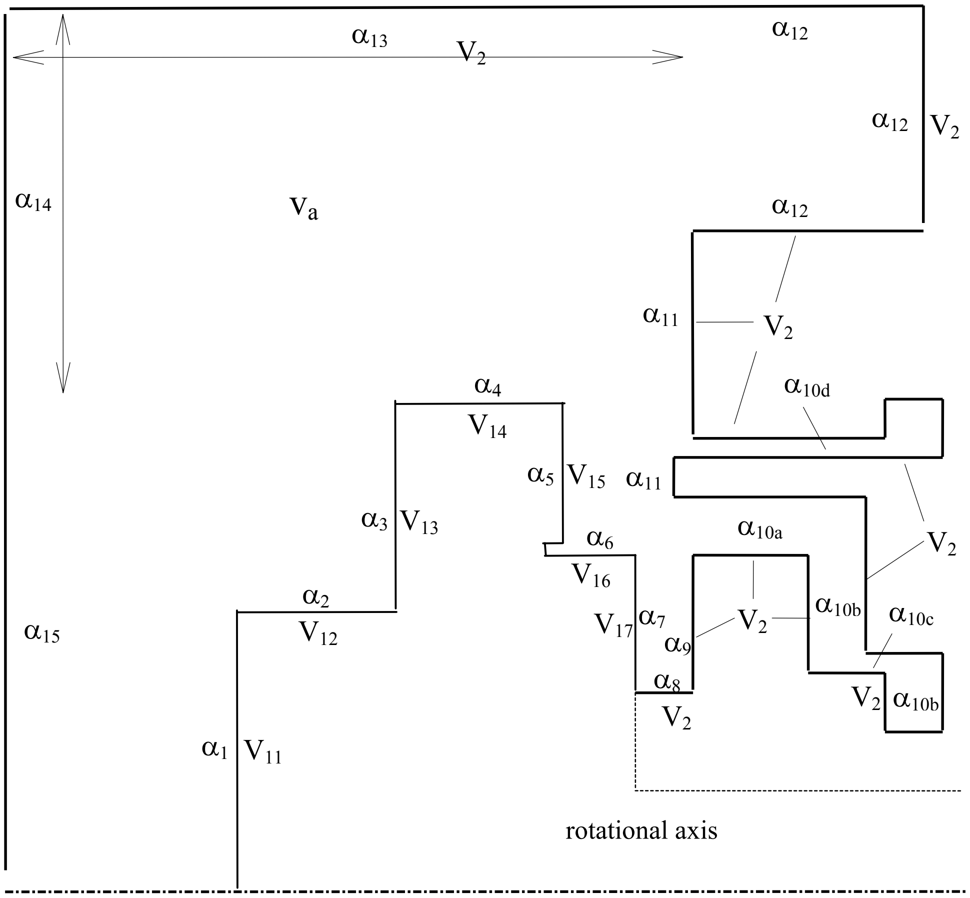
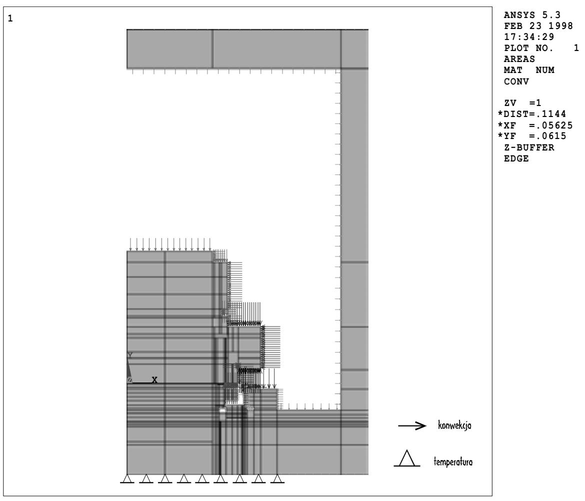


Fig.3 Diagram of partition of surface testing stand and seal elements in the areas with differential convective heat-transfer coefficients (the diagram has not a proportion of dimensions), ($\alpha_1 - \alpha_{15}$) - convective heat-transfer coefficients for particular surfaces, (v_{11} - v_{17}) – average rotational speed on the surfaces of moving elements of seal, $v_2 = 0$ - rotational speed of immobile elements.

Tab.2 Convective heat-transfer coefficients for surfaces blown by working medium at temp. t=61°C.

Surface	**1**	**2**	**3**	**4**	**5**
α [Wm^{-2}K^{-1}]	14026	16613	27514	22518	28649
Surface	**6**	**7**	**8**	**9**	**10a**
α [W/m^{-2}K^{-1}]	18217	24372	17029	18468	5373
Surface	**11**	**12**	**13**	**14**	**15**
α [W/m^{-2}K^{-1}]	23195	7118	6891	7183	8695



The analyzed system was loaded by introducing a heat source in the zone of contact of slide rings. The whole analyzed system with the introduced boundary and loaded conditions is performed in fig.4.

Fig.4 Boundary and loaded conditions of model introduced to the thermal analyses of testing chamber.

Results of analyses

We tried to approach the problem mathematically by using the conduction and convection equation (2) [7]:

$$\rho \cdot c \cdot \left(\frac{\partial T}{\partial t} + \{v\}^T \{L\} T \right) = \{L\}^T \left([D] \{L\} T \right) + \overset{ooo}{q} \, ,$$

where :

ρ - mass density, T - temperature (=T(x,y,z,t)), t - time, c - specific heat , $\overset{ooo}{q}$ - heat generation.

$$\{L\} = \begin{Bmatrix} \frac{\partial}{\partial x} \\ \frac{\partial}{\partial y} \\ \frac{\partial}{\partial z} \end{Bmatrix}$$ - vectorial operator, $$\{v\} = \begin{Bmatrix} v_x \\ v_y \\ v_z \end{Bmatrix}$$ - speed vector;

$$[D] = \begin{bmatrix} \lambda_{xx} & 0 & 0 \\ 0 & \lambda_{yy} & 0 \\ 0 & 0 & \lambda_{zz} \end{bmatrix}$$ - coefficients of thermal conduction.

The time of analyses of the studied phenomenon was in a good correspondence with the real time of achieving the equilibrium on the testing stand.

The results obtained in the conducted calculation enable to perform the temperature distribution and gradients of temperature in the whole analyzed system. In fig.5, the distribution of temperature for the whole testing chamber is performed, whereas in fig.6, distribution of temperature in slide pair elements of seal is shown.

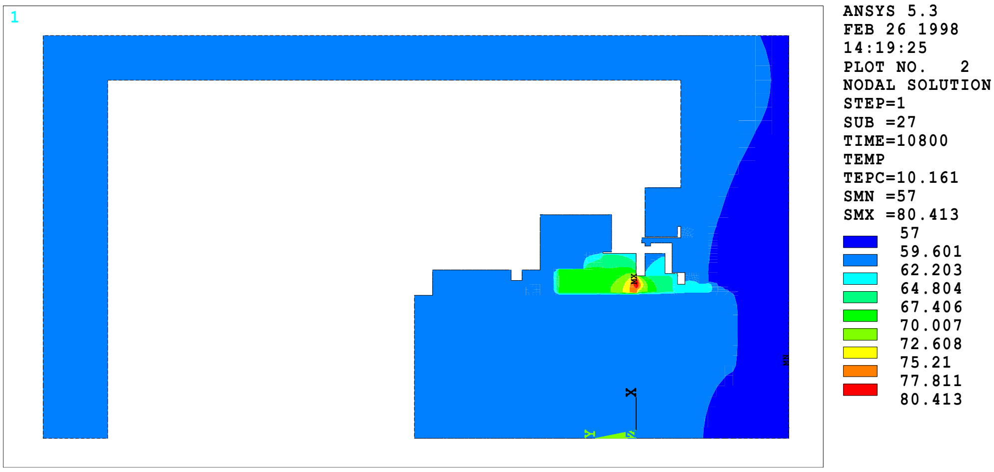


Fig.5 Distribution of temperature in the tested chamber (with the development seal) at the rotational speed n = 1420 $\left[\frac{rot}{\min}\right]$.

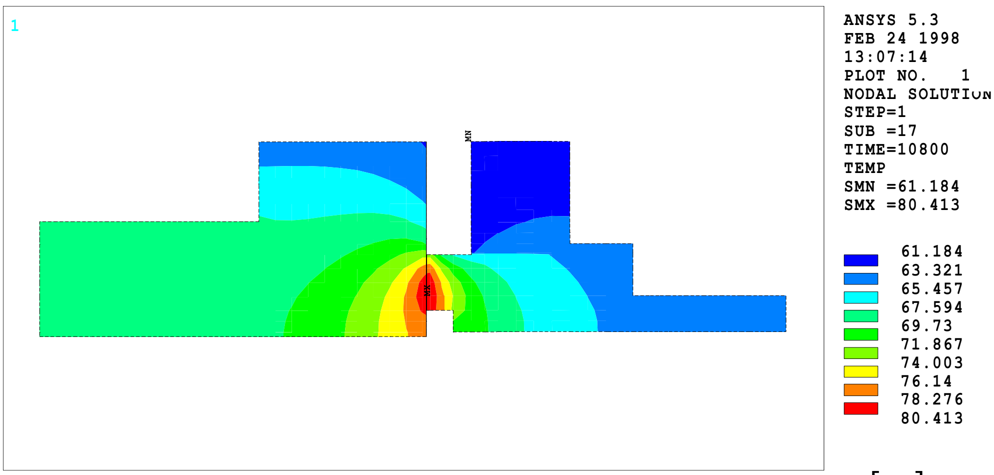


Fig.6 Distribution of temperature in slide rings of seal, performed at the rotational speed n = 1420 $\left[\frac{rot}{\min}\right]$.

Strain analysis

A computer aid simulation enables the determination of strain distribution of seal rings. The rings, which are loaded only by the fluid pressure (fig.7), undergo a strain so that the interspace at the thickness $0{,}11.10^{-6}$ m rises from the inside (space without the working medium – dry space).

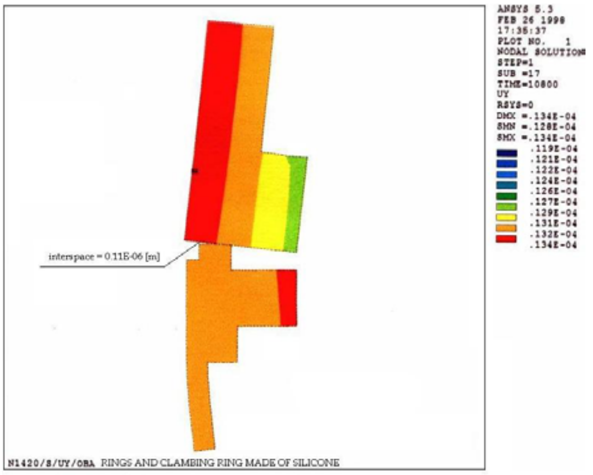


Fig.7 The interspace form under the influence of loaded seal elements (made of 100% silicone) by pressure (overscaled 2000 times).

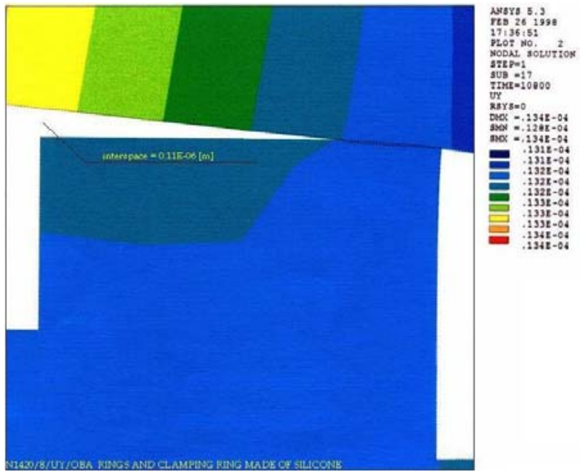


Fig.8 The magnified interspace formed under the influence of loaded seal elements (made of 100% silicone) by pressure (overscaled 2000 times).

This strain is not harmful for the right working of seal. Fig.8 shows the magnification of rings of contact zone.

The seals loaded by temperature, determined on the base of measurements in chousen spots, and the computer simulation were inversely strained in comparison with those loaded by pressure only. The interspace is formed from the external side on the side of working medium – wet space and amounts $0{,}24.10^{-6}$ m. Such a strain is danger because it is able to cause the loss of leaktightness of joint.

The main task in the construction of seal joints is a compensation of thermal strains by means of the strains caused by the working medium pressure. The analysis of seal loaded simultaneously by pressure and temperature is performed in fig.9 and 10.

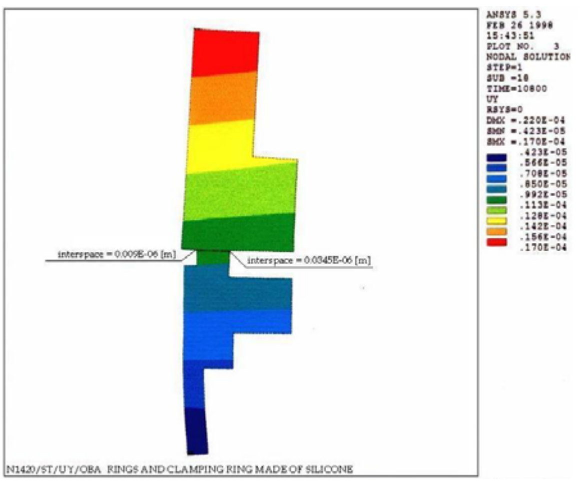


Fig.9 The interspace formed under the influence of loaded seal elements (made of 100% silicone) by pressure and temperature (overscaled 2000 times).

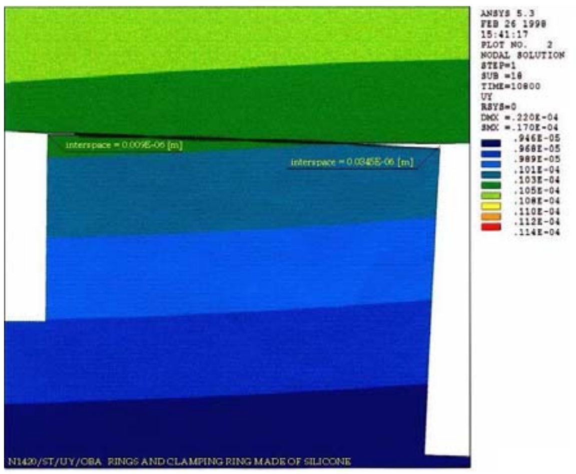


Fig.10 The magnificated interspace formed under the influence of loaded seal elements (made of 100% silicone) by pressure and temperature (overscaled 2000 times).

Thanks to the suitable choice of material, we succeeded to reduce the disadvantageous thermal strain. The interspace on the dry side amounts $0{,}009.\ 10^{-6}$ m, whereas the interspace loaded by the pressure amounts $0{,}11.10^{-6}$ m (that is by 2 order lower quantity). On the wet side it amounts $0{,}0345.10^{-6}$ m in comparison with the interspace ($0{,}24.10^{-6}$ m) loaded by temperature. A permissible value of the interspace on the wet side amounts $0{,}08.10^{-6}$ m.

Experimental verification of the numerical model

Laboratory test

In order to check the most important part of the FE-model for the end-face seal, experimental investigations were conducted in the special industrial laboratory. In the end-face seal made of silicon in the steel frame a big leakage (2 dcm^3/min) was observed. However, the ring made of silicon was not leaking.

Conclusions

As a result of the computer simulations in the end-face seal, the temperature distribution and also the strain and stress distributions in the same area was received. The results are in good agreement with experimental data.

An applicable selection of end-face seals materials allow to reduce disadvantageously thermal strains and to the avoid seal leakage in area of the joint. In previously applied technical solutions, steel seals made of the silicon ring, caused considerable thermal strains due to a higher thermal expansion of steel ALPX = 17.10^{-6} K^{-1} as an opposite to the thermal expansion of silicon, carolit SSiC, ALPX = $4{,}5.10^{-6}$ K^{-1}, i.e. over 3 times higher.

Regarding the increasing of accuracy of numerical simulations, we prepared special data base of materials parameters [9].

In conclusion we can state, that thermal strains in the end-face seal are possible to be compensated by using a working fluid pressure.

References

[1] MAYER, E. „Uszczelnienia czołowe". WNT Warszawa, 1976.
[2] SELEZNEW, K. „Stan cieplny wirników i cylindrów parowych i gazowych turbin", Moskwa 1964.
[3] ANSYS User's Manual For Revision 5.3 - Volume 1-4.
[4] KORONA, L. „Tworzywa z węgla uszlachetnionego nasycone metalami", WNT Warszawa, 1980.
[5] KORONA, L. „Wymiana ciepła w uszczelnieniach czołowych pomp i aparatury chemicznej". Inżynieria i Aparatura Chemiczna, 1973 no 4.
[6] TAYLOR, G.I. „Phil. Trans. R. Soc.". Series A, 1923, 223, 289. - „Stability of a Viscous Liquid contained between Two Rotating Cylinders".
[7] SEBANAND, R.A., JOHNSON H.A. „Heat transfer from a rotating cyl. in oil". NASA, NEMO 4-22-59 W.
[8] KOSTOWSKI, E. „Zbiór zadań z przepływu ciepła", Politechnika Śląska - Skrypty Uczelniane, 1996.
[9] JURASZEK, J., NOWAK, J., RYSIŃSKI, J.:Zeszyt Naukowy OBR Bosmal, Bielsko-Biala nr 16, 2001.