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TEMPERATURE DISTRIBUTION IN THE RINGS OF LIQUID FACE SEALS - EVALUATION OF SIMPLIFIED ANALYTICAL MODELS USING 3D FEM ANALYSIS

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Abstract

A great amount of work has been dedicated in the last two decades to the evaluation of thermoelastic distortions of seal rings. Although small (several microns), these deformations have the same order of magnitude as the film thickness and significantly affects the operation of Mechanical Face Seals (MFS). It has been also demonstrated that temperature gradients in the seal rings produce larger deformations that those produced by pressure distribution so that temperature is the key factor for MFS operation.

Several simplified analytical models for calculation of temperature distribution in MFS operating with full liquid film have been proposed. These models are either simplified 1D models taking into consideration only the axial temperature variation, or more complex 2D models which include also the radial variation of the temperature. Although simple and easy to use, they include some restrictive assumptions which should be assessed. It is the primary objective of the work presented in this paper to evaluate the differences between two basic analytical models and a more realistic 3D FEM model. The comparative analysis is focused on the evaluation of the ring shape simplification and on the influence of variable convection coefficient, two of the most important limitations of the analytical models. The analysis is made on two typical balanced MFS from vendors catalogues.

KEY WORD: *Lubrication, mechanical face seal, heat transfer, temperature, 3-D FEM*

NOMENCLATURE

A_f	seal interface area, $= \pi\left(r_o^2 - r_i^2\right)$, m^2	r	radius, *m*
B	balance ratio	T	temperature, *C*
H	convection coefficient, $W/(m^2 \cdot K)$	T_f	sealed fluid temperature, *C*
h	film thickness, *m*	t	time, *sec*
k	heat conduction coefficient, $W/(m \cdot K)$	w	width of the active face, $= r_o - r_i$ *m*
L	length of the ring, *m*	η	dynamic viscosity, $Pa \cdot sec$
m	complex dimensionless parameter		
n	rotor speed, *rev/sec*		**Subscripts**
P_f	power, *W*	i	inner
q	heat flux, W/m^2	o	outer

1. INTRODUCTION

It is unanimously accepted that thermal effects must be considered in any realistic analysis of a mechanical face seal (MFS). Temperature distribution in the fluid and in the seal rings is important for the prediction of the operating instabilities due to thermal distortions or due to phase change. Also, temperature is frequently used to control the operating conditions of a MFS.

With few exceptions, the main heat source in a MFS is the friction in the gap [1], and most of the heat is rejected by the seal rings. For typical seal geometries the flow equation -Reynolds equation- can be uncoupled from energy equation. If axisymmetry prevails (assumption generally accepted for noncontacting MFS), Reynolds equation becomes one-dimensional, whereas the conduction in the seal rings is bidimensional. Hence, an analytical solution for heat conduction in the seal rings can be obtained only with supplementary simplifying assumptions. This is the reason that most of THD analyzes focus on heat transfer in the seal rings.

Moreover, in many cases, the heat rejected by the stator is only a few percentage of the total heat produced in the seal gap, and consequently, can be neglected. Thus, thermal analyses are focused primarily on the rotor.

Several simplified analytical models for predicting temperature distribution in MFS operating with full liquid film have been proposed in the last two decades. These models are either simplified 1D taking into consideration only the axial temperature variation, or more complex 2D models which include also the radial variation of the temperature.

In the first category (1-D), one can include all the approaches based on classical one dimensional fin model, first time used by Lymer [2] for a noncontacting water seal to detect the boiling initiation in the seal gap. A similar approach, was developed by Morariu & Pascovici [3] for noncontacting, face to face double seal arrangement. Buck [4] and later Müller, G.S. and Müller, H.K. [5], reconsidered the fin model to estimate heat transfer efficiency for more complex seal rotor shapes. The fin model was also used for transient heat conduction analysis; a review of most important papers on this subject can be found in [6].

All the above mentioned models are isothermal in respect to the fluid film and have as final result a constant temperature in the seal dam. It is worth to be mentioned that, whereas the temperature variation is 1-D (axial direction only) the heat transfer has a 2-D character, the heat flow being both axial and radial (to the surrounding sealed fluid).

On the other hand, there are few published approaches predicting 2-D temperature distribution in a MFS. Most of them are based on numerical solution of heat transfer models, obtained either using finite element method [7] or finite differences method [8,9].

Buck [4] published the first attempt to model analytically 2-D heat flow in a MFS based on classical series development solution for steady-state conditions. His model assumes a constant heat flow entering the rotor and, can be used only for rectangular cross-section rings.

The solution proposed by Pascovici and Etsion in 1992 [10] is based on the assumption that the heat flow path in the rotor can be approximated by a set of straight lines, inclined at a fixed angle to the seal face. Their 2-D solution overestimates the temperature gradient in the radial direction. A correction was later introduced by including heat transfer in the stator too [11]. However, this model can be used only for short length rotors.

Although simple and easy to use, all the above-mentioned models include some restrictive assumptions regarding the shape of the ring, or the variation of heat input and convection coefficients on the boundaries, which should be assessed.

It is the main objective of the work presented in this paper to evaluate the differences between two basic analytical models (1-D fin model, respectively, 2-D Buck's model) and a more realistic 3D FEM model. The comparative analysis is focused on the evaluation of the ring shape simplification and on the influence of variable convection coefficient, two of the most important limitations of the analytical models. The analysis is made on two typical balanced MFS from vendors catalogues.

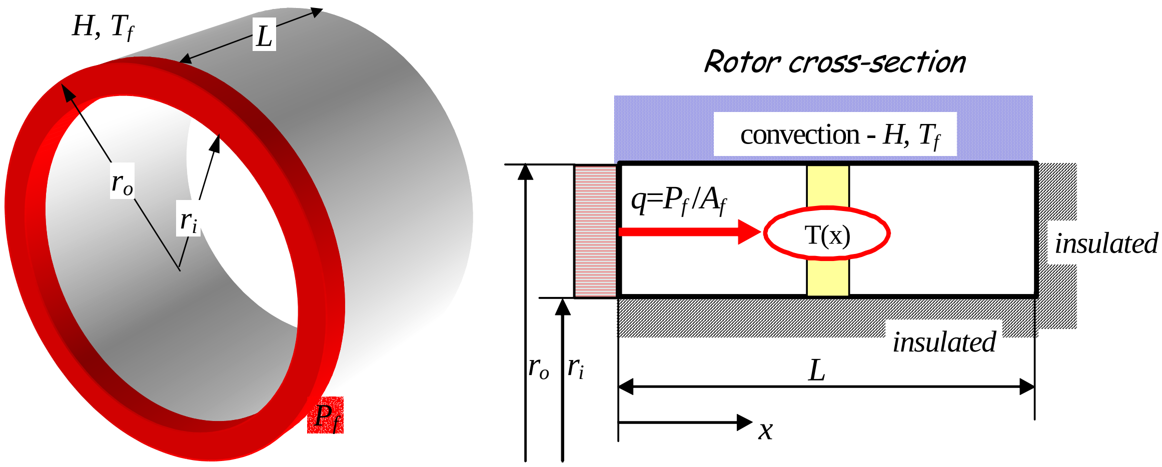

Fig. 1: *Simplified rotor thermal model*

2. THE MODEL

The best way to evaluate the simplified analytical models is to apply them for typical MFS available on the industrial market. Two such seals have been selected, both being balanced seals, in order to assure a fluid film operation. For the sake of clarity, the two simplified analytical models will be resumed at the beginning of this chapter.

2.1. 1-D FIN MODEL

As shown previously (chapter 1) several authors used the one-dimensional heat transfer model for extended surfaces (also known as "fin model") to calculate temperature distribution in the seal rings both for steady-state and transient conditions, as typically the rotor length, L, is several times greater than the seal width, w. In addition, seal width is normally a few millimeters so that one can neglect seal curvature and make the analyzis in Cartesian coordinates.

According to this model [2,5,7], the rotor is considered to be one dimensional fin (the temperature varies only along the length of the rotor), insulated on the inner surface (at radius r_i) and on the end-side opposite to the seal face (at x=L). The heat is rejected through the outer cylindrical surface (at radius r_o) by convection with the sealed fluid assumed to be at constant temperature, T_f. The convection heat coefficient, H, is assumed constant. The rotor is heated on the front face with a constant and uniform heat flux, q, calculated assuming that the heat produced in the seal face and rejected by the rotor is uniformly distributed along the seal gap (Fig. 1.):

$$q=\frac{P_f}{A_f}=\frac{P_f}{\pi\left(r_o^2-r_i^2\right)} \qquad (1)$$

where the power loss, P_f is calculated assuming constant film thickness, h:

$$P_f=\frac{2\pi^3\eta n^2}{h}\left(r_o^4-r_i^4\right) \qquad (2)$$

Finally, the temperature distribution along the rotor yields:

$$T(x)=T_f+\frac{q\cdot L}{k\cdot m\cdot\sinh(m)}\cosh\left[m\cdot\left(1-\frac{x}{L}\right)\right] \qquad (3)$$

where $m=\sqrt{\frac{2r_o H L^2}{k\left(r_o^2-r_i^2\right)}}=\sqrt{\frac{2\pi r_o H L^2}{k A_f}}$ is a dimensionless complex parameter.

2.2. 2-D BUCK'S MODEL

Based on the classical technique of separation of variables the Laplace equation for steady-state heat conduction can be analytically solved on rectangular domains for homogenous boundary conditions. Such a solution has been proposed in 1989 by Buck [4] who assumed the same heat transfer boundary conditions as for 1-D fin model. According with Buck's model the temperature distribution is (see Fig. 2):

$$T(x)=T_f+\sum_{n=1}^{\infty}B_n\cosh(\lambda_n x)\cdot\cos(\lambda_n y) \qquad (4)$$

where

$$B_n=\frac{2\cdot q\cdot\sin(\lambda_n w)}{k\lambda_n\sinh(\lambda_n l)\left[\lambda_n w+\sin(\lambda_n w)\cos(\lambda_n w)\right]} \qquad (5)$$

and λ_n are the solutions of nonlinear characteristic equation $\lambda_n w\tan(\lambda_n w)=\frac{Hw}{k}$.

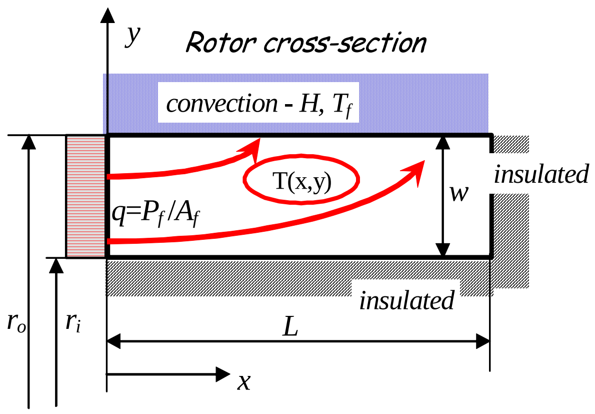

Fig. 2: *Simplified rotor 2-D thermal model*

3-D FEM MODEL

In order to evaluate the accuracy of analytical solutions a 3-D finite elements model for heat transfer and thermo-elastic deformations of seal rings has been developed using I-deas commercial software [12].

The analysis was made for steady-state operating conditions, assuming a complete fluid film separating the faces. However, the model and consequently the conclusions obtained herein can be easily and straightforwardly extended to contacting MFS.

Table 1. Main parameters of the seals

	BH140	LB500
Material	carbon	carbon
Thermal conductivity	45 W/m·K	45 W/m·K
Inner radius	15	14
Outer radius	17	17
Balance ratio	0.515	0.68

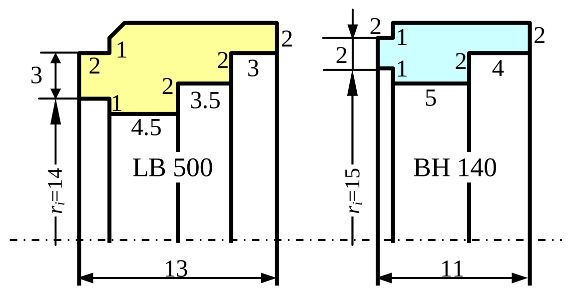

Fig. 3. *Main dimensions of analyzed MFS*

In order to minimize the great number of parameters involved in the comparison the power produced by friction drag in the seal gap was assumed constant and uniformly distributed on the seal face. In addition, only the rotor temperature distribution was analyzed as most of the heat produced is rejected by the rotor [4,13,14]. A second reason is that the rotor face rotation is greater due to both increased temperature gradients and length.

Two commercially available seals have been analyzed [15]: LB500 respectively BH140 seals (see Fig. 3 and Table 1).

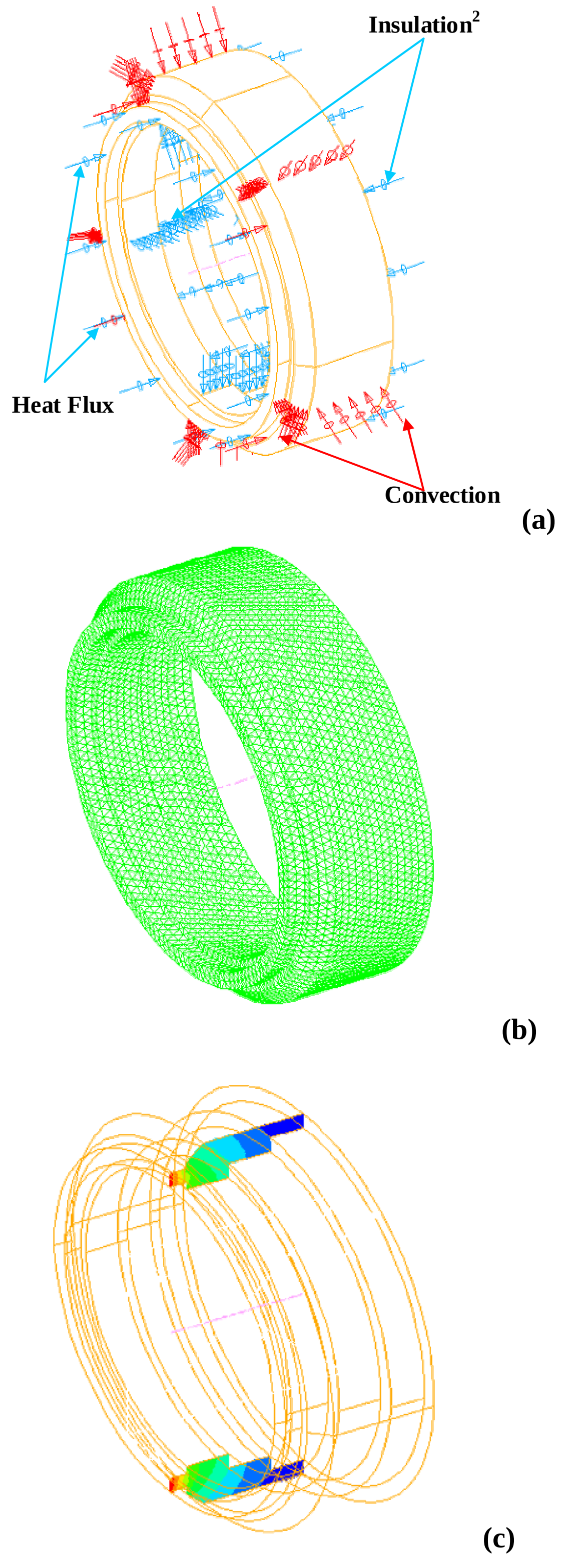

Fig. 4. *FEM model: Boundary conditions (a), meshing (b) and Temperature distribution (c)*

The LB500 rotor was modeled with 25192 solid parabolic tetrahedron elements. The BH140 rotor was modeled with 12255 solid parabolic tetrahedron elements. Contact between two contacting surfaces of the rotor and rotor case was assumed. Axisymmetric boundary conditions have been considered. (Fig. 4)

Several typical convection coefficients according with well-known analytical [13] and experimental studies [14,16] have been used.

3. RESULTS AND DISCUSSION

Two important simplifying assumptions of the analytical models have been primary assessed during our work:

i) the influence of the shape "stylization" that is the way in which a typical complex ring shape is simplified to standard rectangular domains according to analytical modeling;

ii) the influence of the assumption of constant heat convection coefficient.

The comparison was made in terms of two important thermal parameters:

a) **maximum face temperature** which is an important limiting factor for MFS successful functioning

b) **temperature gradient along the seal ring** (axial gradient) considered as the difference between the face temperature and the opposite side (end-side) temperature; this is a measure of ring thermo-elastic deflection (face rotation [17]) item considered in the second phase of the present project.

In a first stage of numerical modeling several 3-D models including all the design details (non-axisymmetric fixtures, anti-rotational pins, etc.) have been considered. A comparative analysis led to the conclusion that for heat transfer analysis, a simpler axisymmetric model is sufficiently accurate.

Four standard simplified geometric models have been considered (Fig. 5): two long models (the length of the simplified model the same with that of the rotor) and two short models (the length of the simplified model does not take into consideration the length of outer rotor fixture). For each of these two categories, two values of seal face width ("narrow" respectively "wide" models) have been considered. A summary of these cases is presented in Table 2.

As an example of the comparisons performed, Fig. 5 shows the axial temperature variation along the seal rotor in the case of BH140 seal. 1-D and 2-D analytical predicted temperatures are plotted for the same thermal conditions.

One can remark that short models (narrow and wide, for both 1-D and 2-D) greatly overestimate the maximum temperature on the seal face. That is due to the design of the ring which is somewhat short that is a shortened model will change significantly the cooling length. At the same time, all the other four possible models could be considered sufficiently accurate for design purposes. However, narrow models should be preferred for conservative design. On the other hand, thermal gradient is overestimated by narrow models while wide models underestimate it. However, the differences, either positive or negative are within acceptable limits.

A similar plot is presented in Fig. 6 for the LB500 seal. As this seal has a longer rotor, the differences between numerical results and all analytical models are smaller. However, one can remark, as in the previous case, that 2-D models predict greater maximum temperatures but are more accurate in terms of temperature gradient.

For the numerical results presented previously, the 3-D model was run with a constant convection coefficient along the entire length of the seal ring, a solution which corresponds to the case of seal cooled with a quenching fluid.

Table 2. Main characteristics of the simplified analytical models

		L		***w***		***m***	
		BH140	**LB500**	**BH140**	**LB500**	**BH140**	**LB500**
Case (a)	short & narrow	7	11	2	3	0.761	0.991
Case (b)	short & wide	7	11	4	5	0.553	0.790
Case (c)	long & wide	11	13	4	5	0.870	0.934
Case (d)	long & narrow	11	13	2	3	1.195	1.172

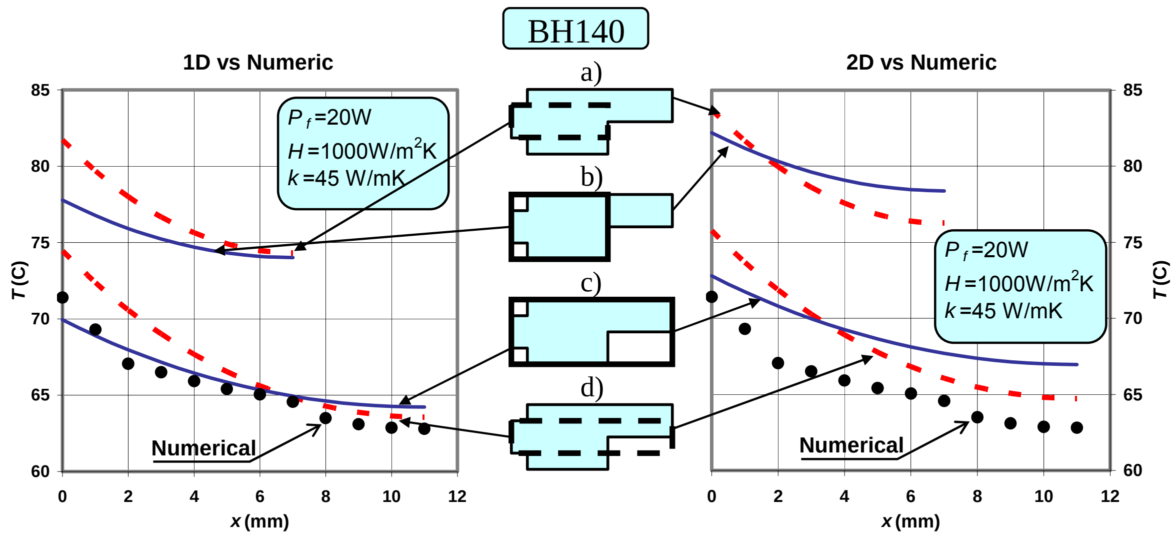

Fig. 5. *Analytical-Numerical comparison for BH140 seal model*

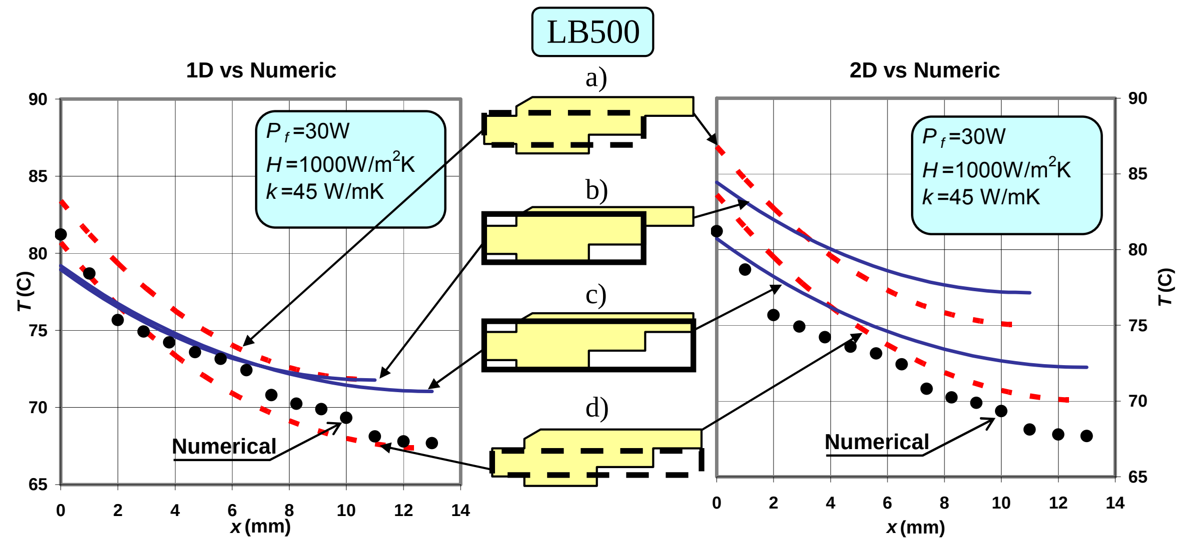

Fig. 6. *Analytical-Numerical comparison for LB500 seal model*

However, in some design solutions, the sealed fluid does not assure the same cooling effect along the entire rotor so that a variable convection coefficient is more realistic. Fig. 7 shows one of the studied cases, where convection coefficient is reduced four times along the rotor length, from a maximum value of H_{max}=1000W/m^2·K downto H_{min}=250W/m^2·K (for the reference the convection coefficient is considered constant and equal to H_{max}). One can see that for both seals the influence is quite important, an increase of about 10ºC being counted for the maximum temperature. At the same time the temperature difference increases also for variable convection coefficient with about 15%. The effect is greater for the longer seal, LB500.

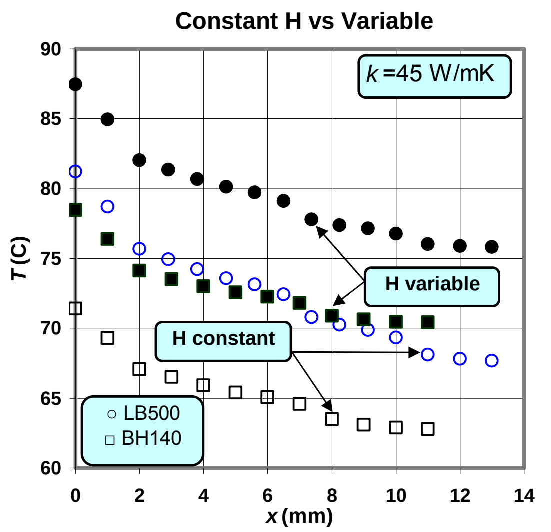

Fig. 7. *The influence of convection coefficient variation*

4. CONCLUSIONS

A comparative analysis of two of the analytical models extensively used in practical design for predicting temperature distribution in seal rings has been performed based on 3-D FEM modeling.

The comparison shows that 1-D models can be used with acceptable accuracy to predict both maximum temperature and the axial temperature gradient, subsequently used to evaluate thermo-elastic deformations of the seal rings. However, for more accurate calculations of seal ring thermal deflections, 2-D models must be considered.

On the other hand, the length assumed for analytical prediction of the maximum temperature must be equal with the entire length of the seal ring, including the dead end section where heat transfer is attenuated in reality.

It can be also concluded that variation of convection coefficients is very important for an accurate prediction of seal temperature and consequently, for seal deformations.

The model and consequently the conclusions obtained herein can be used for both contacting and noncontacting MFS.

The 3-D solid models of the MFS are presently used for the prediction of thermo-elastic deformations of the ring and comparison with published analytical solutions. The results will be published in a future paper.

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