18th International Conference
ENGINEERING MECHANICS 2012
Svratka, Czech Republic, May 14 – 17, 2012

pp. 489–498
Paper #90

NUMERICAL ANALYSIS OF THERMAL FIELDS IN THE INSULATED COVER OF TIRE CURING PRESSES

M. Hynek[*], P. Votapek[**]

Abstract: *Regardless what type of rubber curing chamber is used (steam dome curing press or mold with heated plates) it is reasonable to insulate the chamber cover properly. However, quite often the heat losses though the joints of the metal parts of the chamber cover are underestimated. Numerical analyses can help in an effort to estimate the thermal fields and the resulting heat losses. However, the main problem for such analyses is a realistic specification of thermal boundary conditions especially in case of heat convection. In this paper, an alternative way of setting of convection boundary condition is proposed. Instead of strict specification of convection parameters, which are very difficult to estimate, it is proposed to incorporate a simple thermal boundary layer based only on heat conduction. By this modification no boundary conditions need to be prescribed right on the surface of the analyzed structure and the resulting temperature distribution on the surface has more freedom to classify the actual design of the analyzed structure in particular locations.*

Keywords: ***curing press, heat losses, heat transfer coefficients, FEM thermal analysis***

1. Introduction

It is well known that motion of a fluid with respect to a surface with heat generation is accompanied with a specific form of heat transfer referred to as convection. If the motion of flow is generated by external forces (e.g. wind, air conditioning, fan or pump), it is called forced convection. If it is driven merely by gravity forces, it is called free (or natural) convection. In most cases, both types of convection affect the studied structure and the values of local heat transfer coefficient differ according to location.

Proper evaluation of local heat transfer coefficients based on experimental measurements is extremely complicated and in many situation even impossible. In laboratory conditions, however, there can be exploited several optical measurement methods, such as PLIF (Planar Laser-Induced Fluorescence). This method is based on the principle that atoms or molecules excited with laser spontaneously emit light (observed as fluorescence) which is affected by various parameters such as the concentration of a certain species in a fluid and the temperature. Signals from CCD cameras are then processed to provide 2-D spatial information on concentration, velocity or temperature maps in fluid flows with resolution under 1 mm. From these 2D maps it is possible to evaluate the temperature distribution also in the thermal boundary layer and subsequently the local heat transfer coefficients can be determined from the temperature gradient in the thermal boundary layer (see section 2).

Another method destined for nondestructive testing and inspection, which can also be used in heat transfer measurements, is holographic interferometry. The basic principle of holography lies in its ability to record two slightly different scenes and to display the difference between them. Holographic pictures in general can record motion, deformations, stress, temperature and other continual physical fields. Typically, optical lasers are used nowadays in holographic interferometry giving an accuracy of a half wavelength of the laser (i.e. up to 10^{-3} mm). Again, the detailed knowledge of the temperature field close to the surface allows us to determine the temperature gradient in the thermal boundary layer. Unfortunately, such measurements are hardly performable in real conditions of a plant.

[*] Doc. Ing. Martin Hynek, Ph.D.: Department of Machine Design, Faculty of Mechanical Engineering, University of West Bohemia; Univerzitní 22; 306 14, Pilsen; CZ, e-mail: hynek@kks.zcu.cz

[**] Ing. Petr Votápek: Department of Machine Design, Faculty of Mechanical Engineering, University of West Bohemia; Univerzitní 22; 306 14, Pilsen; CZ, e-mail: pvotapek@kks.zcu.cz

One could propose that running a numerical simulation in order to qualitatively compare heat passage through several variants of a structure might be conducted just with a unified, somehow estimated, heat transfer coefficient. Unfortunately, using a convection boundary condition by setting a specific value of heat convection coefficient and an approached temperature of the fluid (i.e. the temperature just outside the thermal boundary layer) will inevitably influence the temperature distribution in the analyzed structure. Instead, it is suggested here to incorporate a sandwich-like thermal boundary layer directly into the finite element model. As this thermal boundary layer consists of several sub-layers based on heat conduction only, there is no need to solve the problem as a coupled analysis of heat conduction and fluid flow in the boundary layer.

2. Determination of heat transfer coefficients from temperature gradient

It is a fact that the heat transfer coefficient for a specific location on the surface depend on many factor, such as the difference between the temperature of the fluid and the surface, shape, spatial orientation and roughness of the surface, fluid flow velocity, state of the velocity boundary layer and other factors.

Instead of taking all these factors into account it is advised to consider the following presumption. It is assumed that the molecules closest to the heated surface do not move relative to this surface and hence the thin sub-layer of the thermal boundary layer adjacent to the surface is subject to pure heat conduction. Written in an equation, it applies:

$$q_y = -\lambda_S \frac{dT(y)}{dy}, \tag{1}$$

where $\lambda_S[W.m^{-1}.K^{-1}]$ stands for thermal conductivity of the boundary sub-layer closest to the surface and q_y is the density of thermal flux in *y*-direction (see Fig. 1). Reported values of λ_S for dry air at temperatures 0 °C and 100 °C are 0.0237 $W.m^{-1}.K^{-1}$ and 0.0307 $W.m^{-1}.K^{-1}$, respectively.

Let us recall the well-known Newton's law for convective heating or cooling:

$$q_X = \alpha_X (T_{WX} - T_\infty), \tag{2}$$

where $a_X[W.m^{-2}.K^{-1}]$ stands for the local heat transfer coefficient at location *X*, T_{WX} denotes wall temperature at location *X*, T_∞ is the approached temperature of the fluid and q_X is the density of thermal flux at location *X* (again, in *y*-direction).

Admitting that all heat transmitted in convection from the surface to the surrounding fluid must be conducted through the stationary sub-layer of the thermal boundary layer leads to the equation:

$$\alpha_X (T_{WX} - T_\infty) = -\lambda_S \left(\frac{dT(y)}{dy}\right)_{WX}, \tag{3}$$

and hence it follows for the mean value of the heat transfer coefficient on an area A:

$$\alpha_X = \frac{1}{A} \int_A \frac{-\lambda_S}{(T_{WX} - T_\infty)} \left(\frac{dT(y)}{dy}\right)_{WX} dA, \tag{4}$$

where the location *X* takes the place of all points of the area *A* on the heated surface. The evaluation of thermal gradients $\left(\frac{dT(y)}{dy}\right)_{WX}$ can be accomplished from measurements of temperature distribution in thermal boundary layer by means of holographic interferometry or PLIF method. Further information about this approach can be found in Holman (1972) or in Pavelek et al. (2003) where Fig. 1 was taken from.

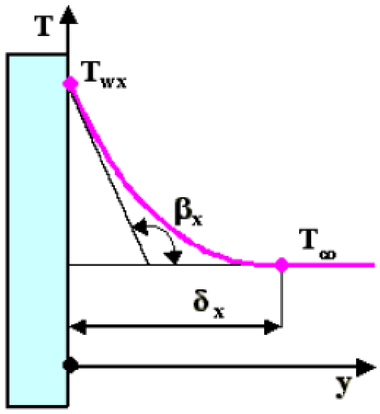

Fig. 1: Temperature profile in thermal boundary layer (δ_X denotes the thickness of the thermal boundary layer which is typically several millimeters)

3. Sandwich-like model of the thermal boundary layer

Applying the principle described in the previous section to the whole thickness of the thermal boundary layer turned out to be very useful for FEM thermal analyses. As mentioned earlier in the text, using a convection boundary condition by setting a specific value of heat convection coefficient and the approached temperature will inevitably influence the temperature distribution in the analyzed structure. To avoid this undesired effect, the thermal boundary layer can be divided into *N* sub-layers (each specified with a reasonable value of thermal conductivity) and included in the FEM model of analyzed structure. Setting the approached temperature outside the outer sub-layer provides more freedom in calculation of the resulting temperature distribution on the surface of the structure.

It should be stressed that pure heat conduction was assumed in individual sub-layers. However, in reality, only a thin sub-layer closest to the surface can be considered stationary. Therefore, as the distance of a particular sub-layer from the surface increases, its artificial value of heat conduction coefficient λ_T^i should increase accordingly to reflect the fact that heat is transferred more intensely due to increasing influence of both free and forced convection. The specific values of coefficients λ_T^i ($i = 1..N$) need to be set in such a manner that the resulting temperature distribution through the thermal boundary layer is in agreement with relevant experimental measurement or published results. For the purpose of thermal calculations presented in this paper a temperature profile published in Čížek (2005) was adopted (see Fig. 6).

In notation of the previous section the density of the artificial heat flux density is introduced as:

$$q_T = \alpha_T (T_{WX} - T_\infty) = -\lambda_{ML} \left(\frac{dT(y)}{dy} \right)_{WX} , \tag{5}$$

where $\lambda_{ML} [W.m^{-1}.K^{-1}]$ stands for heat conductivity of the whole thermal boundary layer evaluated according to the common theory of heat conduction in multilayer structures introducing terms of conduction resistance R_i for individual sub-layers with thicknesses t_i :

$$R = \frac{\delta_X}{\lambda_{ML} \, A} = \sum_i R_i = \sum_i \frac{t_i}{\lambda_T^i \, \mathrm{A}} , \qquad i = 1..N \tag{6}$$

Having known λ_{ML} together with the temperature profile in boundary layer and the wall temperature distribution makes it possible to easily evaluated heat flux through the area *A*:

$$Q_A = \alpha_T \, A \, (T_{WX} - T_\infty) = -\lambda_{ML} A \left(\frac{dT(y)}{dy} \right)_{WX} . \tag{7}$$

4. Application example - FEM thermal analysis of the insulated cover of a curing press

The approach described in the previous section was applied to the thermal analysis of the insulate cover of a large curing press for vulcanization of heavy machinery tires (see Fig. 2). The maximum press force of this particular steam dome press was 900 t, the production cycle was about 80 min., the temperature of the heating steam was 160 °C and the mean temperature of the surrounding air was about 40 °C (see Fig. 3). The main task was to propose cost-effective design changes in order to reduce the heat losses through the cover and to quantify the resulting power sawing.

Fig. 2: Analyzed curing press

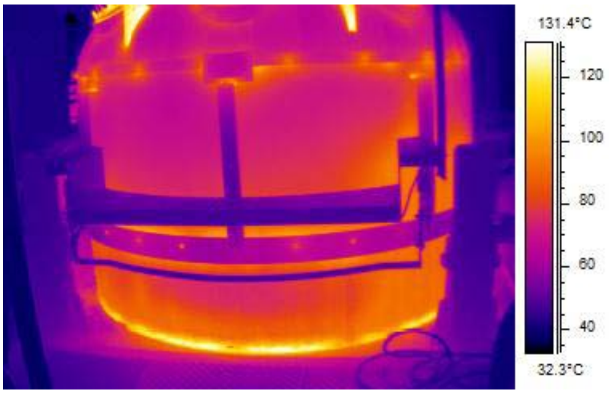

Fig. 3: Snapshot from a thermovision camera

The problem was solved as steady-state thanks to long production times. The geometry of the steam dome made it possible to analyze only an axisymmetrical FEM model. Special attention was paid to three areas where individual parts of the dome are joined together and the cover insulation is fixed to the steel structure of the dome and protected with sheetmetal. While the lower and the upper area represent welded joints, the middle area matches the place where the upper part of the dome is pressed against the circumferential sealing which is fixed in the lower part of the dome (see Fig. 4).

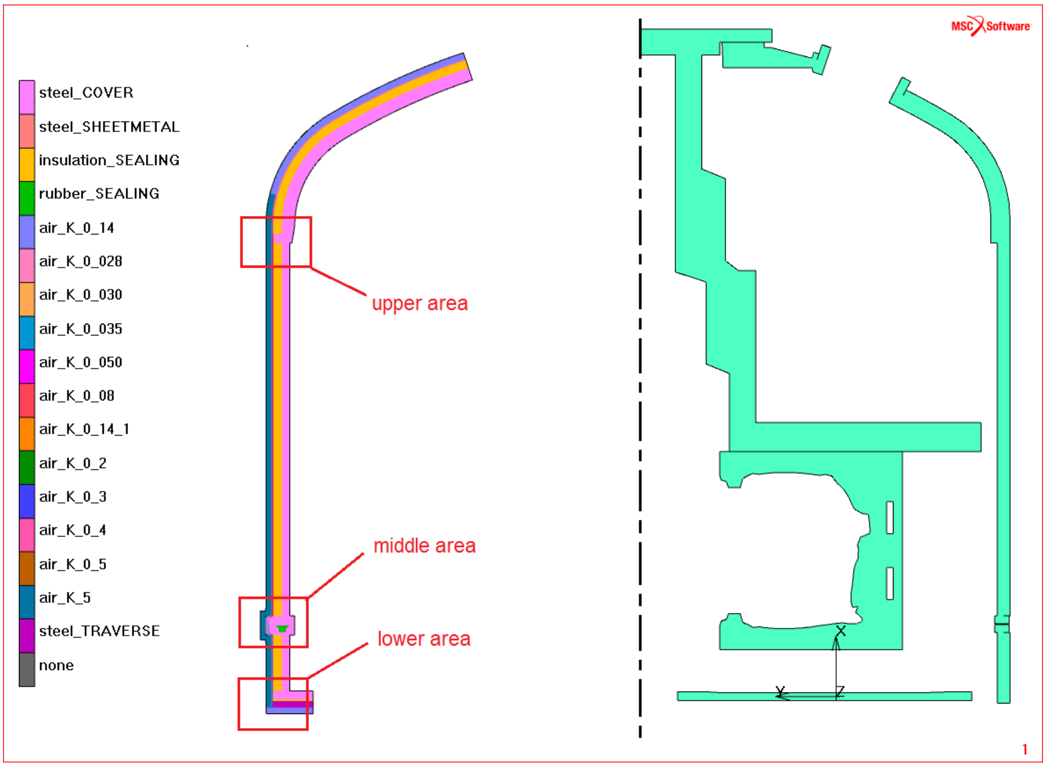

Fig. 4: Axisymmetrical FEM model (left) and the corresponding steam dome press geometry (right)

Critical for the analyzed FEM model was the implementation of the artificial multi-layer thermal boundary layer on the outer surface of the dome cover. The definition of heat conduction coefficients for individual sub-layers (e.g. “K_0_028” refers to $\lambda_T^i = 0.028\,W.m^{-1}.K^{-1}$) as well as the overall thickness of the artificial boundary layer (set to 4.5 mm) is shown in Fig. 5. These values of λ_T^i and the corresponding sub-layer thicknesses were determined by trial and error method until the calculated temperature distribution character in the boundary layer resembled the measured data from Fig. 6.

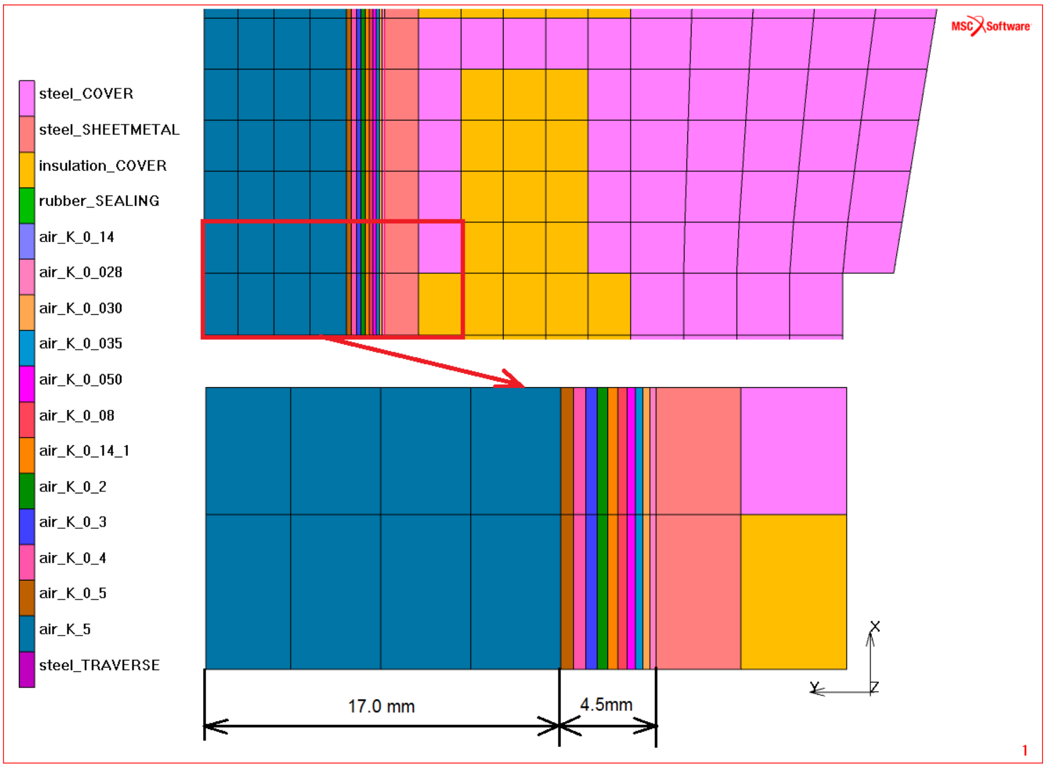

Fig. 5: A detailed view of the artificial thermal boundary layer

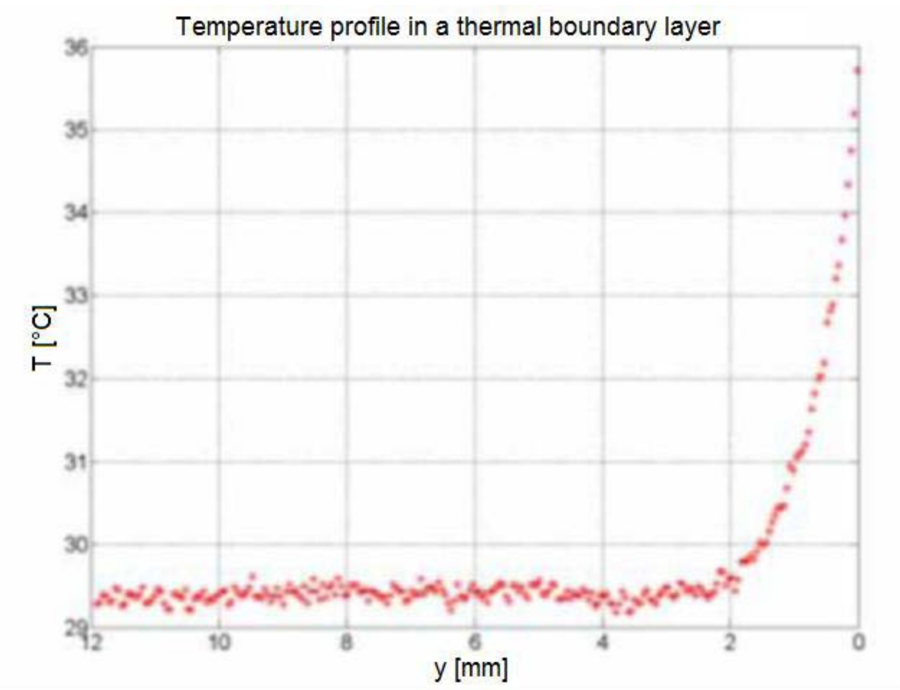

Fig. 6: A typical temperature profile in a thermal boundary layer measured with PLIF, published in Čížek (2005).

In Fig. 7 there are presented detailed views of the FEM model in all three monitored areas. From the material specified for individual FEM elements the design of the press cover can be deduced. On the left the original design and on the right the variant with proposed modifications are shown.

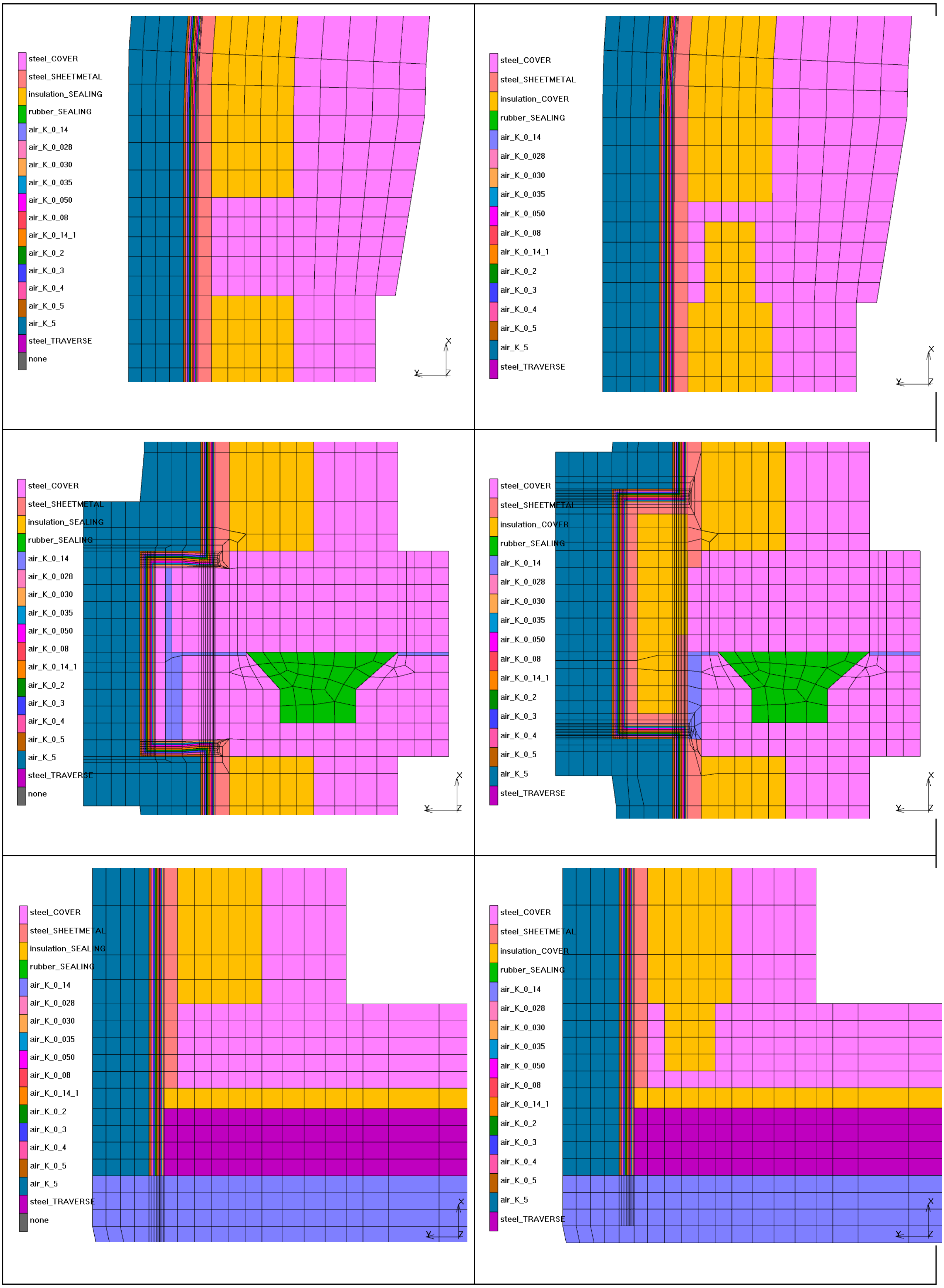

Fig. 7: Design changes between the original (left) and the modified (right) variants of the press cover

Fig. 8 presents the steady state temperature fields in the monitored areas. In each picture of Fig. 8 there is denoted a zone on the surface where heat fluxes will be evaluated and compared.

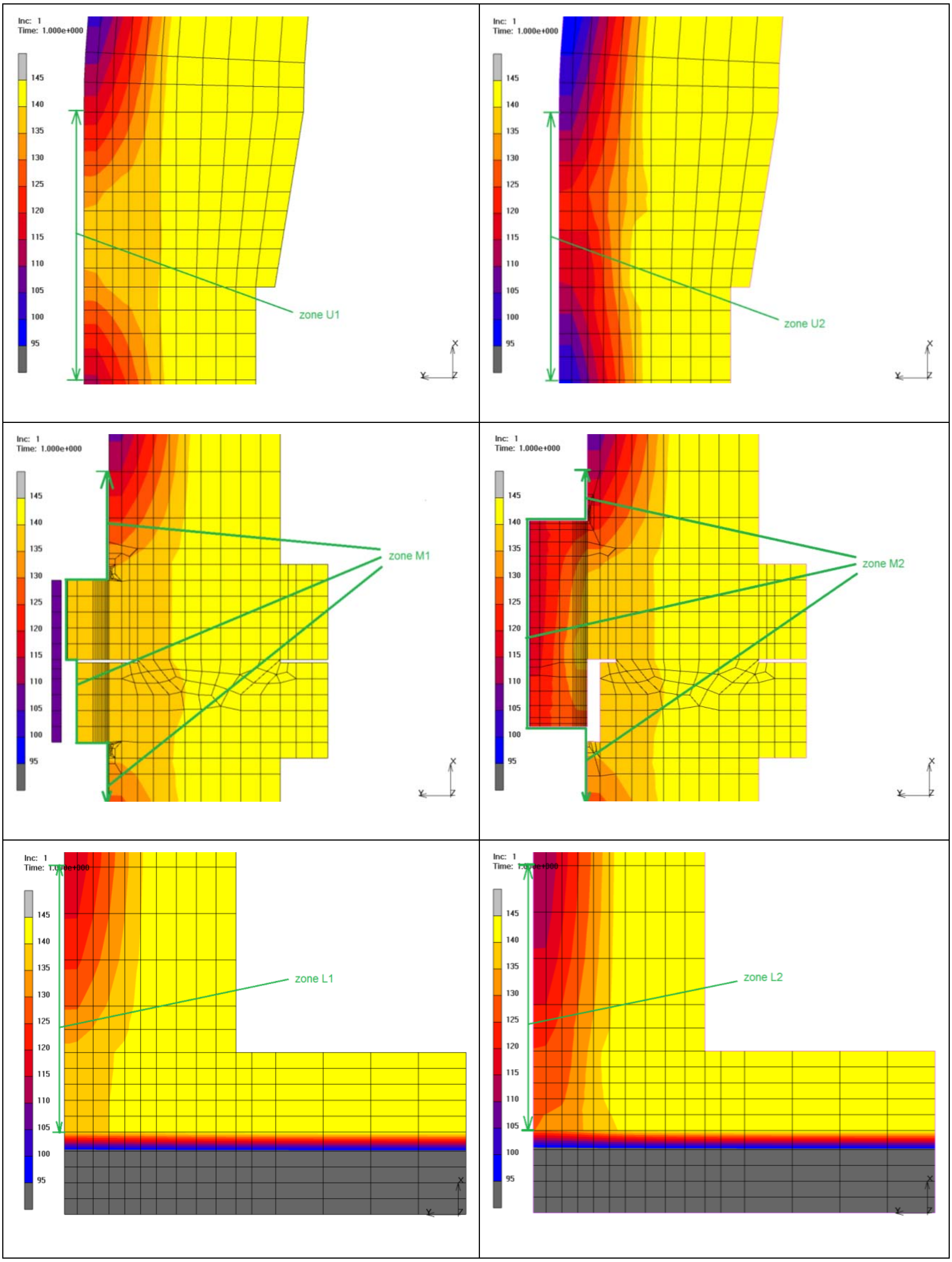

Fig. 8: Steady state temperature fields in the original (left) and the modified (right) variants

The curves plotted in Fig. 9 represent path plots of wall temperature on the surface of each zone, i.e. the position on *x*-axis of the plot corresponds to cumulative length of the edges of involved finite elements and the values on y-axis are nodal temperatures on the surface of the monitored zones.

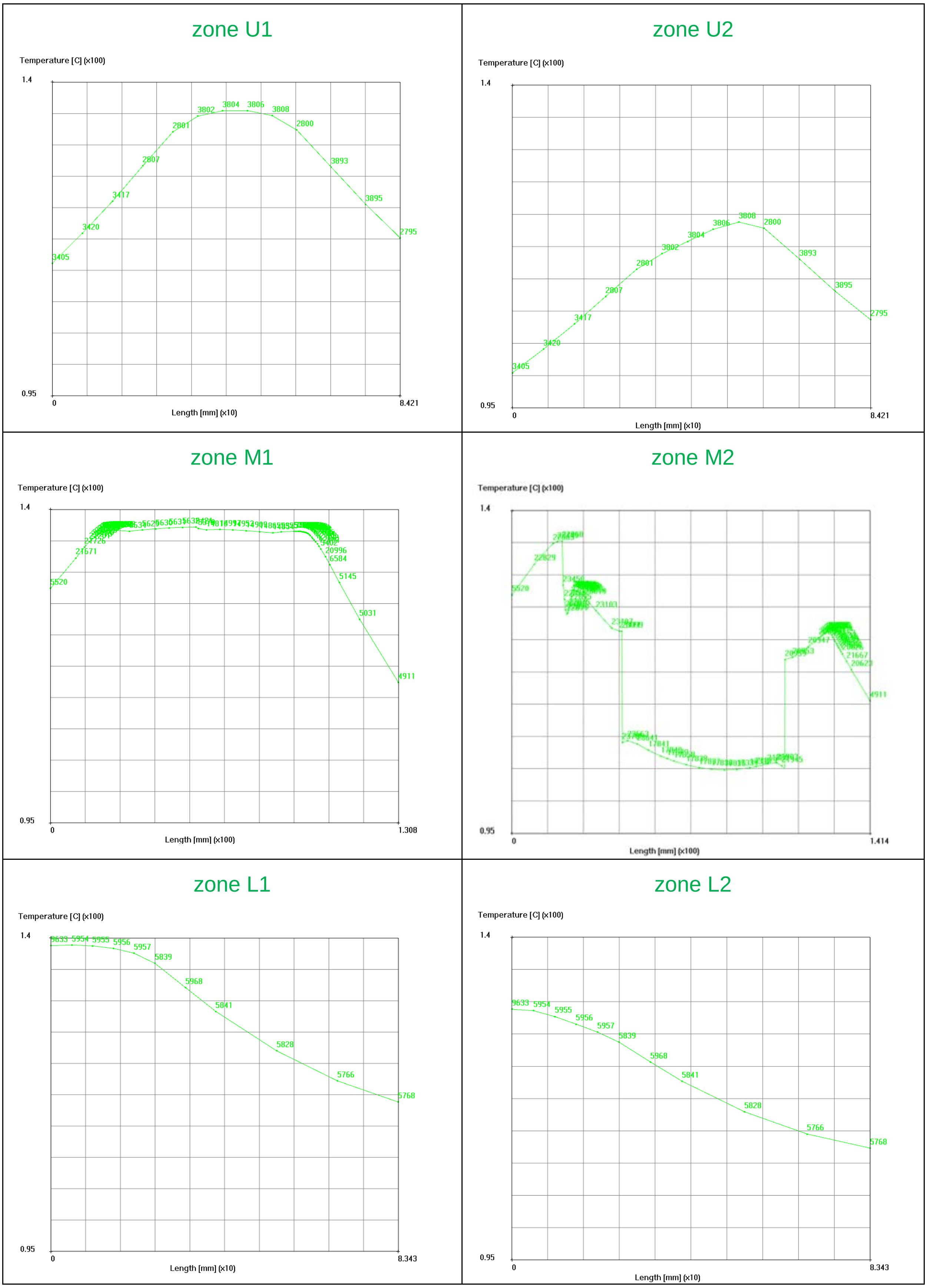

Fig. 9: Path plots of wall temperature for individual zones of interest

In Fig. 10, path plots of the whole cylindrical part of the press cover are compared for both original and modified design. Fig. 11 demonstrates that temperature profiles in the boundary layer depend strongly of the location on the surface of the cover. However, but their characteristic form is in agreement with the measured data from Fig. 6.

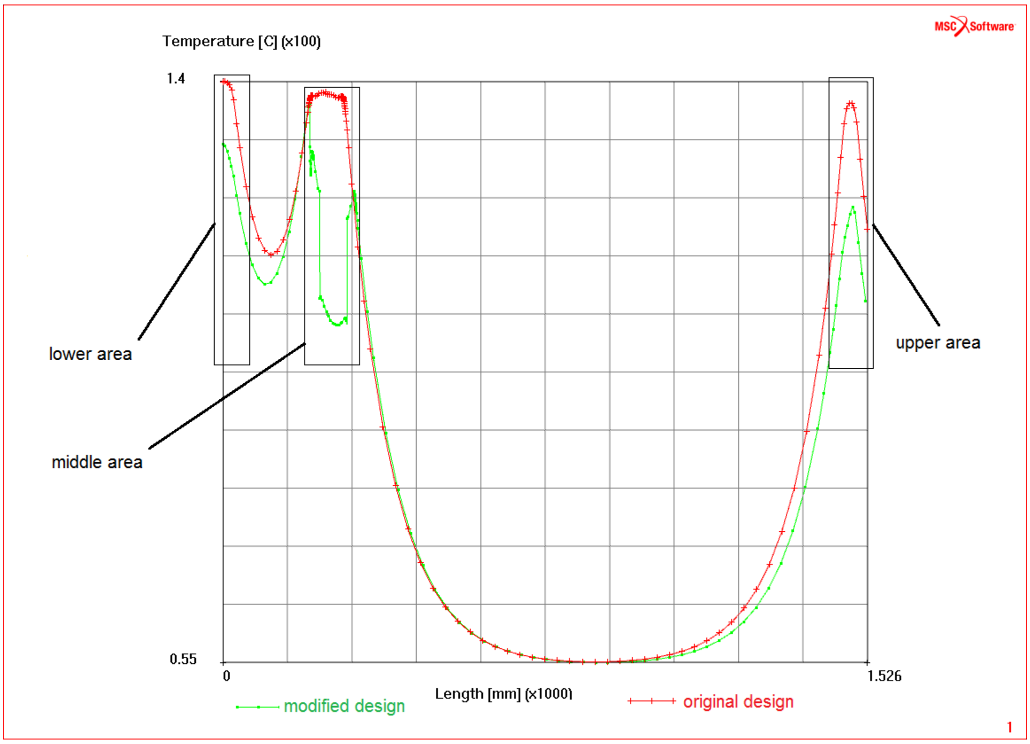

Fig. 10: Path plots of wall temperature for the whole cylindrical section of the press cover

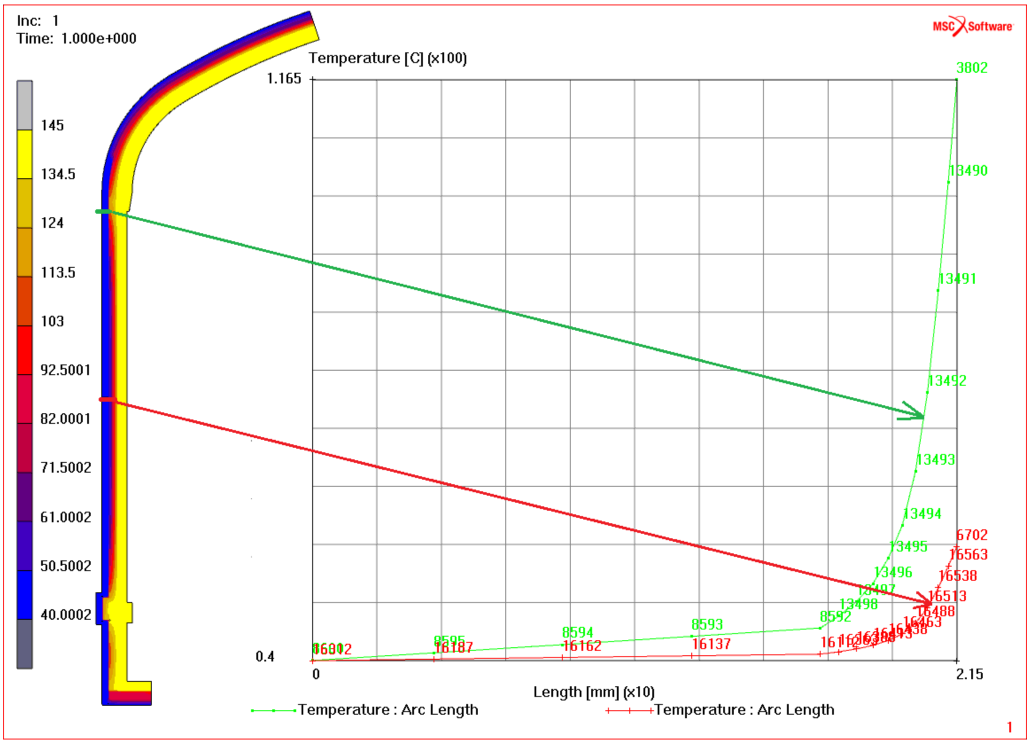

Fig. 11: Temperature profiles in the boundary layer plotted for two specific locations (the modified variant is visualized here): in the zone **U2** *the wall temperature is 116.5 °C, in the middle of the cylindrical section the wall temperature drops to 55 °C*

5. Conclusions

The proposed sandwich-like model of thermal boundary condition in a simplified FEM model of the curing press cove enabled a quantitative assessment of heat losses reduction resulting from the proposed design modifications. Applying the equation (7) to individual monitored zones as well as to the whole cylindrical section of the press cover estimates the values of heat fluxes (for summary, see Tab.1). The presented results confirm that even simple design changes helping to avoid thermal bridging are always worth to take into account.

Tab. 1: Heat fluxes through evaluated areas of the curing press cover evaluated on both original and modified variant

Monitored area	$Q_A[W.m^{-2}]$ **original design**	$Q_A[W.m^{-2}]$ **modified design**	**Power saving [%] modified on original**
upper	2.56	2.15	16
middle	4.11	2.59	37
lower	2.89	2.28	21
whole cylindrical surface	21.15	19.14	9.5

Acknowledgement

The research was performed as a part of the postgraduate study focused on the increasing energy efficiency of technological processes and therefore the authors are grateful to all consultants who contributed to this topic.

References

Bejan, A. (1995): Convection heat transfer, New York: Wiley. ISBN 0-471-57972-6

Čížek, J. (2005): Měření teplotních polí v tekutinách, http://stc.fs.cvut.cz/History/2005/Papers/D_P/cizek_12107.pdf

Holman, J. P. (1972): Heat transfer. Fourth edition. Tokyo, McGraw-Hill Kogakusha, Ltd.

Jícha, M. (1989): Přenos tepla a látky. Skripta VUT, Brno

Lédl,V., Vít,T., Psota,P., Doleček,R. (2011): Holografic Interferometry used for of Temperature Fiel in Fluid, Engineering and Technology 59, World Academy of Science www.waset.org/journals/waset/v59/v59-447.pdf

Meyer, K.E., Larsen, P.S., Guillard,F., Westergaard,C.H. (2002): Temperature and velocity fields in natural convection by PIV and LIF, Proceedings of 11th International Symposium on Applications of Laser Techniques to Fluid Mechanics, Paper 35.1., Lisbon

Pavelek, M. a kol. (2003): Termomechanika. Skripta VUT, Brno, http://ottp.fme.vutbr.cz/~pavelek/optika/1810.htm