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

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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. 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.

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 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:

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

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).

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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:

$$\alpha_X(T_{WX} - T_\infty) = -\lambda_S \left(\frac{dT(y)}{dy}\right)_{WX}, \tag{2}$$

where $\alpha_X[W.m^{-2}.K^{-1}]$ stands for the local heat transfer coefficient at location X, T_{WX} denotes wall temperature at location X and T_∞ is the approached temperature of the fluid.

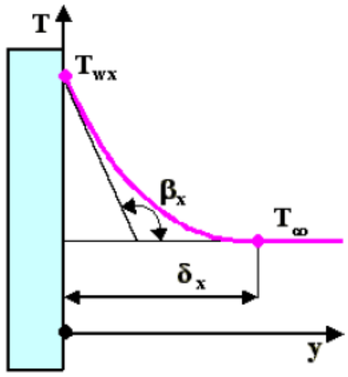

Fig. 1: Temperature profile in thermal boundary layer (δ_X denotes the thickness of the thermal boundary layer which is typically several millimeters)

3. Sandwich-like model of the thermal boundary layer

Applying the principle described in the previous paragraph to the whole thickness of the thermal boundary layer turned out to be very useful for FEM thermal analyses. The thermal boundary layer was be divided into N sub-layers (each specified with a reasonable value of thermal conductivity) and included in the FEM model of analyzed structure. 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 values of coefficients λ_T^i ($i = 1..N$) need to be set in such a manner that the resulting temperature distribution in the thermal boundary layer is in agreement with experimental measurements. In notation of the previous section the density of the artificial heat flux density is now introduced as:

$$q_T = \alpha_T(T_{WX} - T_\infty) = -\lambda_{ML} \left(\frac{dT(y)}{dy}\right)_{WX}, \tag{3}$$

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 :

$$R = \frac{\delta_X}{\lambda_{ML}\, A} = \sum_i R_i = \sum_i \frac{t_i}{\lambda_T^i\, A}, \qquad i = 1..N \tag{4}$$

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 on the analyzed cover of the curing press:

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