Prediction of the Average Surface Heat Transfer Coefficient for Model Foodstuffs in a Vertical Display Cabinet

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Abstract

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Calculations of transient temperatures of food products after they are transferred from a warm environment into a display cabinet, require data on the surface heat transfer coefficient (SHTC). There is no forced air flow in an ordinary display cabinet, so the energy transfer is achieved mainly by free convection, conduction to a supporting plate, and radiation. Theoretical analysis of the heat transfer to a cylindrical sample demonstrates the relative influences of these mechanisms. This work investigates the apparent surface transfer coefficients with metal models. Heated models were placed individually (bare) in containers with and without lids. Each model was surrounded by identical containers filled with water. These were initially at the same temperature as the model or at the mean cabinet temperature. There were one, two, or three layers of these water containers. From the measured time-temperature histories of the model and the air surrounding the model, the SHTCs were calculated as functions of time and transformed into the dependencies between SHTC and temperature difference. The highest SHTCs were observed when the model was placed directly on the metal shelf of the display cabinet. The models surrounded by cool water containers showed lower SHTC values. The lowest SHTC values were found with the models placed in the middle of three layers of warm water containers. Placing the model on an insulating base leads to a lower SHTC. This effect confirms that the heat conduction through the substrate increases the heat transfer from the model and thus increases the average value of the apparent SHTC.

Keywords: SHTC; vertical display cabinet; correlations; food safety

The mathematical modelling of the temperature history of foods in the distribution chain together with the modelling of micro-organisms potential growth can contribute to the enhancement of the methods for the quantitative microbial risk assessment and to improvements on food safety.

In the commercial practice, it happens very often that the chilled foods are not placed into the display cabinets pre-cooled but in a heated state either from the transport or because of having been stored out of a chilled space for some time. In order to be able to determine the time course of the temperature of foods put into the display cabinet in this way and to evaluate the degree of the microbiological risk, it is also necessary to know, among other parameters, the surface

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heat transfer coefficient (SHTC) from the foods into the chilled air in the cabinet. In the available literature, practically no data are given on the SHTCs of foods put into the chilled air of the display cabinet under the conditions of negligible forced convection.

There exists literature data about forced convection. Recently, Becker and Fricke (2004) collected and evaluated a great number of equations for cooling and freezing the foods under the conditions of forced convection. The problem of cooling the containers with foods in a thick layer on the pallet under the conditions of forced air convection at different velocities is also dealt with Carniol *et al.* (1998). The authors specified the Nusselt criterion values as functions of the container layer order and the air velocity.

Our case (placing a warm food into a cool air space with a low velocity air flow) is most similar to the solved case of the time-varying SHTC when storing the frozen foods (Scott & Beck 1992). The authors used the "Karlsruhe test substance" in the form of a flat block as a model of frozen foods. They studied the transfer of frozen foods from the surroundings at –6°C into an environment of about –35°C and back. The free convection around the model can be characterised by the time-varying SHTC within the range from 6 to 12 W/(m^{2}K).

Recently Anderson *et al.* (2004) dealt with thawing and freezing selected kinds of meat products in home refrigerators under the conditions of free convection. They specified the mean values of the SHTC within the range of 8–15 W/(m^{2}K) for freezing and 5–7 W/(m^{2}K) for thawing, respectively.

As the SHTC from the foods into the environment inside the display cabinet is influenced by the shape of the foods, the kind of packaging, location, the temperature difference, and the mechanisms of the heat transfer, it is not possible to use the values valid for completely different geometries and conditions. The objective of this work is therefore to specify the time courses of SHTCs and the correlations of SHTC with temperature difference for a typical food object in a specific display cabinet.

Theoretical analysis

The simplest way how to determine the mean SHTC experimentally is to substitute a sample of food with a geometrically identical metallic model and to record the time temperature profile of this model. The SHTC can be evaluated by using only two recorded temperatures: the ambient temperature and the uniform temperature of the model. The procedure can be illustrated by analysis of the cooling of a warm metallic container (height H, initial temperature T_{c0}) that is placed upon a steel plate (thickness h) at a temperature T_0. The container is fully enclosed by the cooling cabinet walls and by air having the same initial temperature T_0. As long as the thermal conductivity of the contents of the container is very high, it can be assumed that its temperature is uniform – the time necessary to attain a uniform temperature is quite short for the typical size of metallic models ($\sim H^2/a\pi \sim 1$ s). The rate of temperature changes is controlled by three basic mechanisms: natural convection, radiation, and conduction, assuming a perfect contact with the supporting plate.

With the aim to estimate the relative contributions of these heat transfer mechanisms, we shall consider a simplified case, a cylindrical sample of the radius R and height H (ρ_c, c_c, $\lambda_c \to \infty$) placed upon an infinitely large plate of the thickness h (ρ, c, λ, thermal diffusivity $a = \lambda/\rho c$). It is always useful to describe the problem in terms of dimensionless quantities, for example dimensionless temperature θ (plate), θ_c (cylinder), and dimensionless radius

$$\theta = \frac{T - T_0}{T_{c0} - T_0}\ , \quad \theta_c = \frac{T_c - T_0}{T_{c0} - T_0}, \quad r = \frac{r^*}{R} \tag{1}$$

It could be possible to introduce also the dimensionless time (Fourier number), however, it is probably better to preserve the dimensional time due to the fact that the characteristic time has quite different meaning and different values in the different heat transfer mechanisms mentioned above.

Free convection. The heat transfer coefficient α corresponding to the free convection is usually correlated with the Rayleigh number. As soon as a uniform α on the sample surface can be assumed, and assuming a thermally insulated bottom, the rate of the recorded temperature decrease of the metallic cylinder is described by a simple equation:

$$\frac{d\theta_c}{dt} = \frac{\alpha}{\rho_c c_c H}\left(1 + \frac{2H}{R}\right)\theta_c \tag{2}$$

For a constant heat transfer coefficient, the solution of Eq. (2) is an exponential time temperature profile, however, a more realistic solution corresponds to the varying heat transfer according to

$$Nu = \frac{\alpha H}{\lambda_a} = C_R \mathrm{Ra}^m \tag{3}$$

where:

Ra – Rayleigh number

m – exponent being usually 1/4 for laminar and 1/3 for turbulent flows respectively

Using correlation (3), the solution of (2) can be expressed as follows

$$\theta_c(t) = \frac{1}{(1 + t/\tau)^m}$$

where:

$$\tau = \frac{H^2 \rho_c c_c}{C_R \mathrm{Ra}_0^m m \, \lambda_a (1 + 2H/R)} \tag{4}$$

where: Rayleigh number was evaluated at the initial temperature T_0

Radiation. Radiation heat transfer can be estimated for sources and sinks that are not very distant from each other using the Boltzmann law. Temperatures T_0 (cool cabinet walls acting as heat sinks) and T_c (cylinder, θ_c – dimensionless temperature of cylinder) in the case of a negligible ratio of the sample surface to the surface of the surrounding walls are related by the following equation:

$$\frac{d\theta_c}{dt} = \frac{\sigma (T_0 + T_c)(T_0^2 + T_c^2)}{\rho_c c_c H} \left(1 + \frac{2H}{R}\right) \theta_c \tag{5}$$

assuming relative emissivity of the sample surface = 1. This expression corresponds, in the case of a small difference between T_0 and T_c to the equivalent heat transfer coefficient $\alpha = 4\sigma T_0^3$, where σ is the Stefan-Boltzmann constant. The value α = 5 W/(m^{2}K), corresponding to the typical mean temperature T_0 = 280 K, is quite comparable with the values predicted by free convection.

Conduction. A warm metallic sample is cooled down also by the direct thermal contact with the supporting plate (shelf) representing an additional heat capacity – the plate acts as a firmly attached thin fin. Assuming an insulated thin stainless steel plate, the temperature in the plate depends only upon the dimensionless radial coordinate r as described by the Fourier equation

$$\frac{\partial \theta}{\partial t} = \frac{\alpha}{R^2 r} \frac{\partial}{\partial r}\left(r \frac{\partial \theta}{\partial r}\right) \tag{6}$$

together with the boundary and initial conditions

$$\theta(t, r = 1) = \theta_c(t),\ \theta = (t, r \to \infty) = 0,\ \theta(t = 0, r) = 0 \tag{7}$$

The temperature in the insulated metallic cylinder is uniform and depends only upon time

$$\frac{d\theta_c}{dt} = \frac{2a\phi}{R^2} \left(\frac{\partial \theta}{\partial r}\right)_{r=1} \tag{8}$$

$$\phi = \frac{h\rho c}{H \rho_c c_c} \tag{9}$$

where the dimensionless criterion ϕ, the relative heat capacity of the plate and sample characterises the influence of the heat conduction.

The system (Eqs 6–9) of partial and ordinary differential equations for the temperature of the cylinder and that of the plate can by solved analytically by using Laplace transform (Carslaw & Jaeger 1986). Transforming time to the Laplace variable p, we can easily derive the Laplace transform of solutions, expressed in terms of Bessel functions of the second kind K_0 for the temperature field in the plate

$$\tilde{\theta}(p,r) = \tilde{\theta}(p) \frac{K_0(qr)}{K_0(q)},\quad q = \sqrt{\frac{R^2 p}{a}} \tag{10}$$

and for the Laplace transform of the cylinder temperature

$$\tilde{\theta}_c(p) = \frac{R^2 K_0(q)}{aq(qK_0(q) + 2\phi K_1(q))} \tag{11}$$

The time profile of the cylinder temperature can be obtained from Eq. (11) by using the inversion theorem and by calculating residuals of Eq. (11), giving the result in the following integral form:

$$\theta_c(t) = \frac{1}{2\pi i} \int_{\gamma - i\infty}^{\gamma + i\infty} e^{ut} \tilde{\theta}_c(u) du = \frac{8\varphi}{\pi^2} \int_0^{\infty} e^{-atu^2/R^2} \frac{du}{u \Delta u} \tag{12}$$

where:

$$\Delta u = (uJ_0(u) - 2\,\phi J_1(u))^2 + (uY_0(u) - 2\,\phi Y_1(u))^2 \tag{13}$$

is expressed by means of Bessel functions of the first kind J_n and Y_n. The rate of temperature changes follows immediately from Eq. (12) and can be related to the rate of temperature changes for free convection

$$\frac{\left(\frac{d\theta_c}{dt}\right)_{conduction}}{\left(\frac{d\theta_c}{dt}\right)_{convection}} = \frac{\frac{8\varphi a}{\pi^2 R^2} \int_0^{\infty} e^{-atu^2/R^2} \frac{u du}{\Delta u}}{\frac{\alpha}{\varrho_c c_c H}\left(1 + \frac{2H}{R}\right) e^{\frac{\alpha}{\varrho_c c_c H}\left(1 + \frac{2H}{R}\right)t}} \tag{14}$$

Remark: The integrals in Eqs. (12) and (14) have to be evaluated numerically.

Combined effect of heat transfer mechanisms. Previous paragraphs analysed the possible heat transfer mechanisms separately, giving the results (temperature profiles and cooling rates corresponding to the instantaneous change in the ambient temperature) in an analytical form. With the aim to analyse a complex case, it is simpler to carry out numerical analysis, for example by using the finite difference method. The problem was formulated by means of the previously introduced correlations for the radiation, Eq. (5) and free convection Eq. (3), where $C_R = 0.59$ and $m = 1/4$ are the values recommended for laminar flow by Šesták and Rieger (1974). The effect of the heat conduction was solved not only according to section 2.3 (perfectly thermally insulated plate), but also assuming a constant heat transfer coefficient at the plate surface. The following cases were evaluated (Figure 1). A cylindrical sample with an insulated bottom (free convection and radiation), the same sample cooled down also from bottom, further on a completely insulated sample connected to a circular insulated fin, and finally the most realistic case when all heat transfer mechanisms are considered. The geometry and properties were selected as close as possible to real cases: $R = 0.05$ m, $H = 0.1$ m, $h = 0.002$ m, $\beta = 0.0037\ K^{-1}$, $\nu_a = 13\times10^{-6}\ m^2/s$, $a_a = 19\times10^{-6}\ m^2/s$, (air), $\rho_c = 2700\ kg/m^3$, $c_c = 900$ J/(kgK), $\lambda_c = 200$ W/(mK), (cylinder), $\rho = 7800\ kg/m^3$, $c = 460$ J/kg K, $\lambda = 15$ W/(mK), (plate), ambient temperature $T_0 = 0°C$, the initial temperature of the cylinder $T_{c0} = 20°C$.

The results shown in Figure 1 demonstrate that the effect of radiation is significant (even more important than the free convection in this particular case), and that the effect of the heat conduction in the supporting plate cannot be neglected either, namely at the very beginning of the cooling process (Figure 2).

There are other effects that have not been taken into account, for example the velocity field disturbance after the placement of the sample into the cooling cabinet and the effects of the air-curtain. Generally speaking, there are several mechanisms, characterised by different time constants: the shortest one is the evolution of hydrodynamic and thermal boundary layer on the sample surface (analysis indicates that this time constant is of the order of seconds), followed by the heat conduction into the supporting plate (time constant about a minute). The convection and radiation are characterised by much longer time constants (up to an hour), however, not exactly the same because the free convection heat transfer coefficient is a decreasing function of the temperature difference while the equivalent heat transfer coefficient for radiation is almost constant. The relative magnitude of the heat flow mechanisms analysed is of the same order and depends upon the sample size, temperature differences, relative heat capacities of the sample and shelf ϕ, emissivity of the sample surface and the cabinet walls. The procedure described above can be applied only for the simplest cases i.e. simple geometry (cylinder without wrapping) and a stand alone sample. For a geometrically more complicated sample and first of all for a group of mutually interacting samples, other factors start to be important: the wrapping of the sample surface, the relative distance between the samples, the effects of the lids, etc.

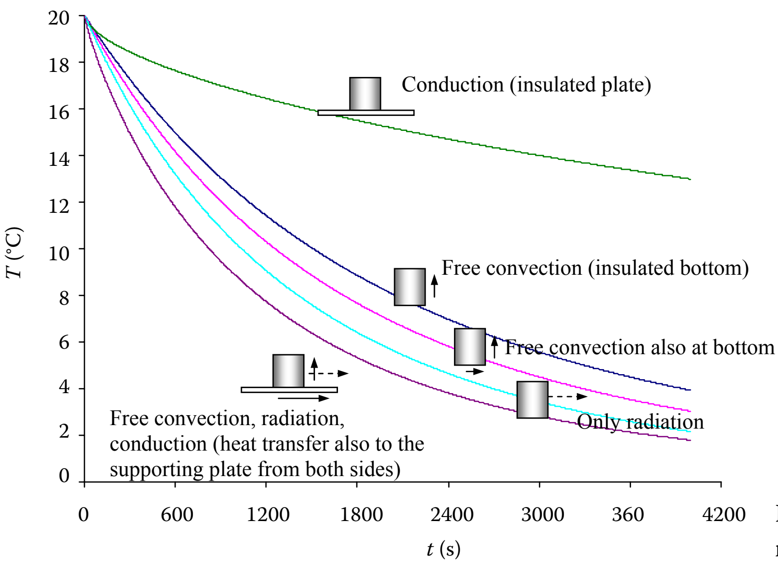

Figure 1. Time temperature dependencies of a metallic cylinder ($R = 0.05$ m, $H = 0.1$ m)

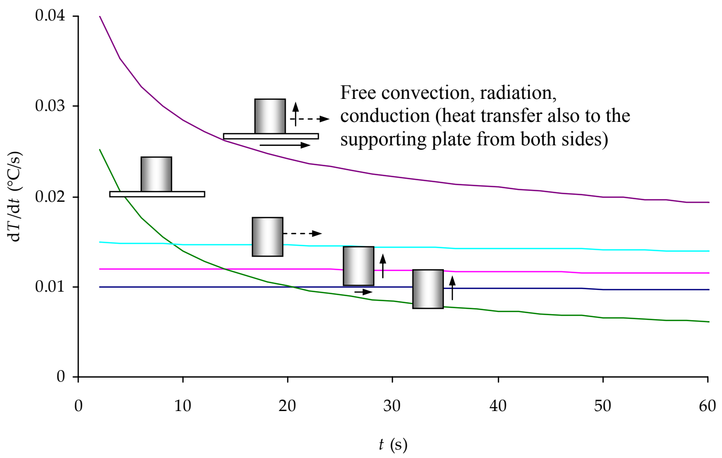

Figure 2. Rate of temperature decrease at the beginning of cooling (the values are practically proportional to the SHTC)

The conventional characterisation using a general engineering correlation between dimensionless numbers starts to fail for such a complex assembly of containers. This is because the complex geometry necessitates using an unmanageably large number of the dimensionless descriptors. In view of these complications, we suggest an alternative approach of specifying an extended set of experimentally obtained SHTC values. We discuss this alternative approach in the following sections.

MATERIAL AND METHODS

The measurement of the temperatures of the models during cooling was carried out in an industrially manufactured refrigerated display cabinet, which is intended for sale of chilled perishable foods in the retail network.

Description of the refrigerated display cabinet. The display cabinet used was Optimer 1946 (Linde Chladicí technika Ltd., Beroun-Závodí, Czech Republic) with the following dimensions: length 1875 mm, breadth with lighting 895 mm, height 1980 mm. The cabinet has five horizontal shelves for chilled products. The control system uses the on/off principle within an adjustable temperature range of the air of 0.5–4.5°C. A steady load was placed into the cabinet for simulating the real conditions. The steady load was created by identical containers filled with water and closed with lids. The containers with the water content of 0.5 l in the number of 20 pieces were evenly placed on sides of the shelves 1–4. On shelf 5, the number of containers was 30. For the measurement itself, a free space of the breadth of 380 mm in the middle of every shelf was left for placing the models.

Description of the measurement. The models heated up to the room temperature were placed either individually or in a group with other samples into the display cabinet in specific positions on the individual shelves. The positions of the models placed in the display cabinet are schematically shown in Figure 3. Both the model temperature and the temperature of the air surrounding the model on its shelf were measured.

Experimental set-up. The temperatures of the models and the surrounding air were measured by Cu-Co thermocouples from the company Omega (USA). The thermocouples measuring the tem-

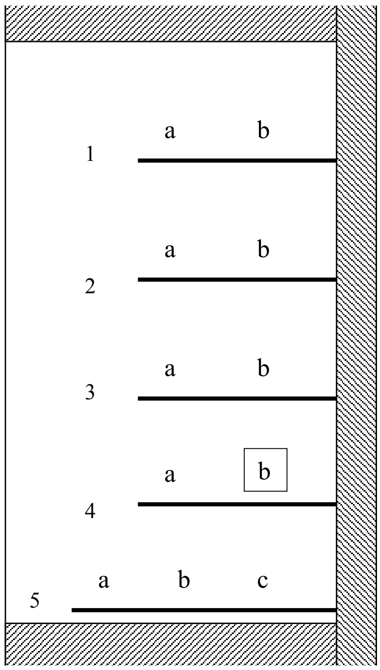

Figure 3. A schematic drawing of the display cabinet, indication of the placement of the model in cabinet by letters a, b, c on five shelves numbered 1–5, e.g. position 4b means shelf 4, position b

perature of the metal models were placed into the middle of the metal models. For measuring the thermoelectric voltages, the multimeter HP-34970A was used, from which the measured data were transmitted into a personal computer at regular intervals (5 s) by means of the program Datalogger (Hewlett & Packard Co., USA). The thermocouples were calibrated before use with the help of the stirred water ultrathermostat Prüfgeräte Medingen (Germany) by means of the calibrated digital thermometer Therm 2230-1 Ahlborn Messtechnik (Germany). The maximum temperature measurement error was ± 0.1°C.

Description of models. Three metal models were made for three sizes of plastic containers. Figure 4 shows the shapes of the models in the containers covered withy lids. The sizes of the containers were: the big container: base 58 × 83 mm, top part 82 × 108 mm, height 98 mm; the medium container: base 63 × 88 mm, top part 82 × 108 mm, height 48 mm; and the small container: base 66 × 91 mm, top part 82 × 108 mm, height 28 mm. Identical lids 90 × 116 mm, height 8 mm, fitted all three types of container. The containers and their lids are made of polypropylene, the side walls are corrugated, the thickness of the wall and the bottom is 0.25 mm. The influence of corrugation on the size of the heat transfer surface S was not taken into consideration (planar faces were assumed). The metal models were made of an alloy of tin and lead in equal proportions (1:1) with the estimated heat conductivity of 50.6 W/(mK), specific heat capacity of 0.177 kJ/(kgK), and density of 9325 kg/m^3, based on the thermal properties of the pure components, as stated by Raznjevic (1984). The metal models for the individual sizes of the containers were made in accordance with

Figure 4. Metal models placed into containers and covered with lids used in the experiments

the standard metal casting procedure. The height of the model was adjusted so that an air gap between the model and the lid was created, which corresponds to the filling of a real food into the packaging. The heights of the big, medium, and small metal models were 85, 36, and 22 mm, respectively. The outer surfaces and the masses of the metal models were as follow: the big model: S = 0.0387 m^2, m = 4.760 kg; the medium model: S = 0.0254 m^2, m = 2.256 kg, and the small model: S = 0.0214 m^2, m = 1.433 kg. Several drops of oil were put into the containers before placing the metal model in them in order to improve the thermal contact. The metal model was placed among geometrically identical containers filled with water to the same level as the upper edge of the metal model and closed with a lid.

Arrangement of models in the display cabinets. The measurements were carried out for different positions and arrangements of the sample models and surrounding containers on the shelves of the display cabinet. At first, the influence was studied of the packaging on the heat transfer during the cooling of the metal models of all three sizes. The temperature drop of the metal model and the temperature of the air inside the cooling cabinet on the same shelf near the model (distance about 200 mm) were recorded simultaneously. The metal model without packaging, in the containers with and without the lid, was tested. The air temperature was measured outside the layers. That temperature represents the mean air temperature on the respective shelf. In the following tests, aimed at studying the effect of different configurations of the metal models and surrounding samples (containers filled with water and covered with lids), the metal model was always in the container covered with the lid. The samples with water were initially heated up to the room temperature or cooled down to the temperature in the display cabinet. The following arrangements were tested:

Configuration 3 × 3, small, medium, and big models. Single layer with the metal model placed in the middle of eight samples.

Configuration 3 × 3 × 3, small and medium models. Three layers with the metal model in the middle of the central layer among 26 containers filled by water.

Configuration 3 × 3 × 2, big models. Two layers with the big samples (9 samples in each layer).

Procedure of prediction of the SHTC **α.** On the basis of the measured time-temperature course of the metal model and the air temperature in the display cabinet measured near the metal model, the SHTC α was computed according to

$$mc_p \frac{dT_m}{dt} = -\alpha S(T_m - T_p) \tag{15}$$

With the aim to obtain the time-course of α in an analytical form, it was necessary to fit the function into the time course of the temperatures. The time course of the metal model temperature was approximated by double exponential regression function (*a, b, c, d* and *e*-fitted parameters)

$$T_m = ae^{-t/d} + be^{-t/e} + c \tag{16}$$

by using the DataFit (Oakdale Engineering, USA) program. The temperature of the air T_p in the display cabinet was approximated by linear or quadratic polynomials and together with Eq. (16) was used for the evaluation of the SHTC from Eq. (15). The measurement was terminated when the temperature change of the model became small. Only that part of the measurement where the temperature changes of the model were significant enough was used for the calculation (we have used the part of data characterised by maximum relative error of the temperature difference lower than 15%). Those parts of the measured temperature courses which were influenced by the regime of defrosting the display cabinet (a sudden rise and drop of the air temperature in the display cabinet) were also excluded.

Processing of data. Preliminary measurements of the air velocity in the display cabinet indicated that the effect of forced convection is negligible and the heat transfer is controlled by natural convection, radiation, and conduction, as discussed in the previous section and by Hoke *et al.* (2006). While these theoretical models seem to be applicable for a single container situation, different arrangements of many containers of complex shapes are difficult to generalise: the correlations including the terms describing all the heat transfer mechanisms would involve too many parameters. Therefore, the measured SHTC values have been approximated quite empirically by means of a simple power-law function of the temperature difference between the metal model and the air in the display cabinet atmosphere

$$\alpha = p \times \Delta T^n \tag{17}$$

where:

p, n – empirical constants predicted by means of the computer program DataFit (Oakdale Engineering, USA)

One correlation relation was applied to several data sets measured at different positions of the experimental set-up (different shelves, different positions on each shelf). Therefore, the regression equation is the "mean" of the results of several repetitions of the experiment (minimum two, maximum five), e.g. Figure 6.

RESULTS AND DISCUSSION

The typical time dependencies ofthe model and air temperatures measured after placing the metal model on shelf 3, position "a" (medium size model in packaging placed into the middle of 8 heated samples) are given in Figure 5. This shows that the measured oscillating temperature of the air on the shelf of the cabinet T_p was "smoothed" by the regression equation. The model temperature T_m was fitted by Eq. (16) with high accuracy.

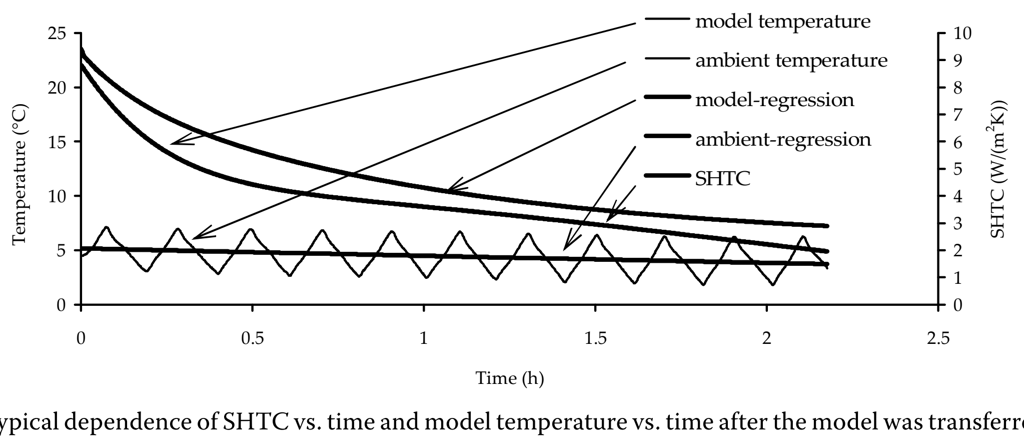

Figure 5. Typical dependence of SHTC vs. time and model temperature vs. time after the model was transferred into the display cabinet. Medium size model in packaging among 8 heated samples in the middle of one layer (position 3a)

Table 1. Metal model without packaging, in a container without lid and in a container with lid (in packaging)

Model	Equation	Validity range (°C)	SHTC at ΔT = 10°C (W//m^{2}K))
Without packaging			
Small	$\alpha = 0.0620\ \Delta T^{2.37}$	$11.2 < \Delta T < 24.2$	
	$\alpha = 1.551\ \Delta T^{1.038}$	$5.6 < \Delta T < 11.2$	16.9
Medium	$\alpha = 0.00175\ \Delta T^{3.36}$	$15.3 < \Delta T < 25.0$	
	$\alpha = 8.370\ \Delta T^{0.252}$	$2.8 < \Delta T < 15.3$	14.9
Big	$\alpha = 9.202\ \Delta T^{0.146}$	$3.3 < \Delta T < 22.9$	12.9
In a container without lid			
Small	$\alpha = 2.219\ \Delta T^{0.843}$	$2.4 < \Delta T < 23.3$	15.5
Medium	$\alpha = 0.02611\ \Delta T^{2.38}$	$15.1 < \Delta T < 24.1$	
	$\alpha = 14.85\ \Delta T^{0.04}$	$2.8 < \Delta T < 15.1$	16.3
Big	$\alpha = 6.477\ \Delta T^{0.305}$	$3.6 < \Delta T < 22.2$	13.1
In a container with lid (in packaging)			
Small	$\alpha = 2.985\ \Delta T^{0.616}$	$4.4 < \Delta T < 25.3$	12.3
Medium	$\alpha = 6.436\ \Delta T^{0.351}$	$5.0 < \Delta T < 26.0$	14.4
Big	$\alpha = 5.291\ \Delta T^{0.358}$	$4.2 < \Delta T < 25.6$	12.1

Also presented is the calculated apparent SHTC as a function of time. The calculated SHTC decreased from the maximal value of 8.8 W/(m^{2}K) at the beginning of the experiment to the value of 2 W/(m^{2}K) at the end of the experiment.

Figure 6 shows the calculated dependencies of SHTC vs. temperature difference for the same experimental arrangement as shown in Figure 5. There are several curves representing the experiments made at different positions in the cabinet. It is apparent that there are small differences between different shelves and positions. We have omitted these differences and predicted the parameters of regression Eq. (17) valid for all positions in the cabinet.

The regression equations obtained for different model sizes and experimental arrangements are given in Tables 1–4. In some cases, it was necessary to predict the regression equation for two separate ranges of temperature difference (Tables 1 and 3). The sharp decay of the apparent SHTC was found at the initial stage of the experiments, probably due to the enhanced conduction of heat from the model to the shelf. This effect was observed for the small and middle size models only, when the relative heat capacity of the shelf ϕ is high (0.05 or 0.1). The big

Table 2. Model in packaging among 8 samples which were heated up to the room temperature and cooled down to the temperature in the display cabinet (one layer) – the model in the middle of the layer

Model	Equation	Validity range (°C)	SHTC at ΔT = 10°C (W/(m^{2}K))
Heated up to the room temperature			
Small	$\alpha = 0.747\ \Delta T^{0.986}$	$2.2 < \Delta T < 22.4$	7.23
Medium	$\alpha = 1.092\ \Delta T^{0.697}$	$2.5 < \Delta T < 18.3$	5.44
Big	$\alpha = 0.861\ \Delta T^{0.699}$	$4.2 < \Delta T < 24.8$	4.31
Cooled down to the temperature in the display cabinet			
Small	$\alpha = 4.58\ \Delta T^{0.53}$	$3.1 < \Delta T < 24.8$	15.5
Medium	$\alpha = 5.18\ \Delta T^{0.42}$	$3.0 < \Delta T < 23.5$	13.6
Big	$\alpha = 4.88\ \Delta T^{0.355}$	$4.0 < \Delta T < 24.5$	11.0

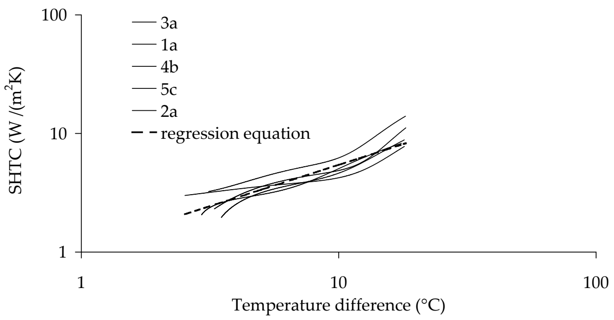

Figure 6. Medium size model in packaging in the middle of 8 heated samples (in one layer) for various positions in the cabinet (see Figure 3 for definition of positions)

model temperature decay is probably not much affected by the heat conduction to the shelf (ϕ = 0.02). We also studied how this effect is influenced by a layer of foam polystyrene insulation at the sample bottom (Figure 10). It is obvious that the placement of the model on the insulation decreases the apparent SHTC values, however, the sharp decay of SHTC values persists. The exact explanation has not been found, the cause may be the initial disturbance of the velocity field or a short term influence of the small heat capacity of the insulation.

The comparison of the influence of different geometry arrangements for different sizes of the models is made in Figures 7–9. The marks on the curves do not represent the experimental data. They are used only for the individual regression curves identification.

Figure 7 is devoted to the small model. It can be seen that the metal model without packaging exhibits the largest apparent SHTC values. If the model is placed into the packaging (plastic container fitting tightly on the model), the SHTC is slightly smaller. Also, covering the plastic container with a lid affects the SHTC values. If the model is placed in one layer of chilled samples (the model is placed on the metal shelf), the SHTC values do not differ very much from the case valid for the individual model. If the model is placed into the centre of three layers of chilled samples, the SHTC values are much lower. Similar values were with the found with the model placed into one layer of heated samples. The lowest SHTC values were found with the metal model placed into the centre of three layers of heated samples (the same containers filled with water).

Figure 8 is devoted to the medium model. The arrangements of the model without packaging and

Table 3. Model in packaging among 26 samples heated up to the room temperature and cooled down to the temperature in the display cabinet (three layers)-the model in the middle of the central layer

Model	Equation	Validity range (°C)	SHTC at ΔT = 10°C (W/(m^{2}K))
Heated up to the room temperature			
Small	$\alpha = 1.072\ \Delta T^{0.183}$	$4.2 < \Delta T < 20.0$	1.63
Medium	$\alpha = 1.009\ \Delta T^{0.15}$	$4.2 < \Delta T < 18.6$	1.42
Cooled down to the temperature in the display cabinet			
Small	$\alpha = 1.339\ \Delta T^{0.81}$	$2.7 < \Delta T < 20.5$	8.64
Medium	$\alpha = 0.0143\ \Delta T^{2.32}$	$14.7 < \Delta T < 20.1$	
	$\alpha = 2.46\ \Delta T^{0.41}$	$2.7 < \Delta T < 14.7$	6.32

Table 4. Model in packaging among 17 samples heated up to the room temperature and cooled down to the temperature of the display cabinet (two layers) – the model in the middle of the upper layer

Model	Equation	Validity range (°C)	SHTC at ΔT = 10°C (W/m^2 K)
Heated up to the room temperature			
Big	$\alpha = 1.454\ \Delta T^{0.097}$	$3.9 < \Delta T < 18.9$	1.82
Cooled down to the temperature of the display cabinet			
Big	$\alpha = 3.333\ \Delta T^{0.331}$	$3.1 < \Delta T < 20.5$	7.14

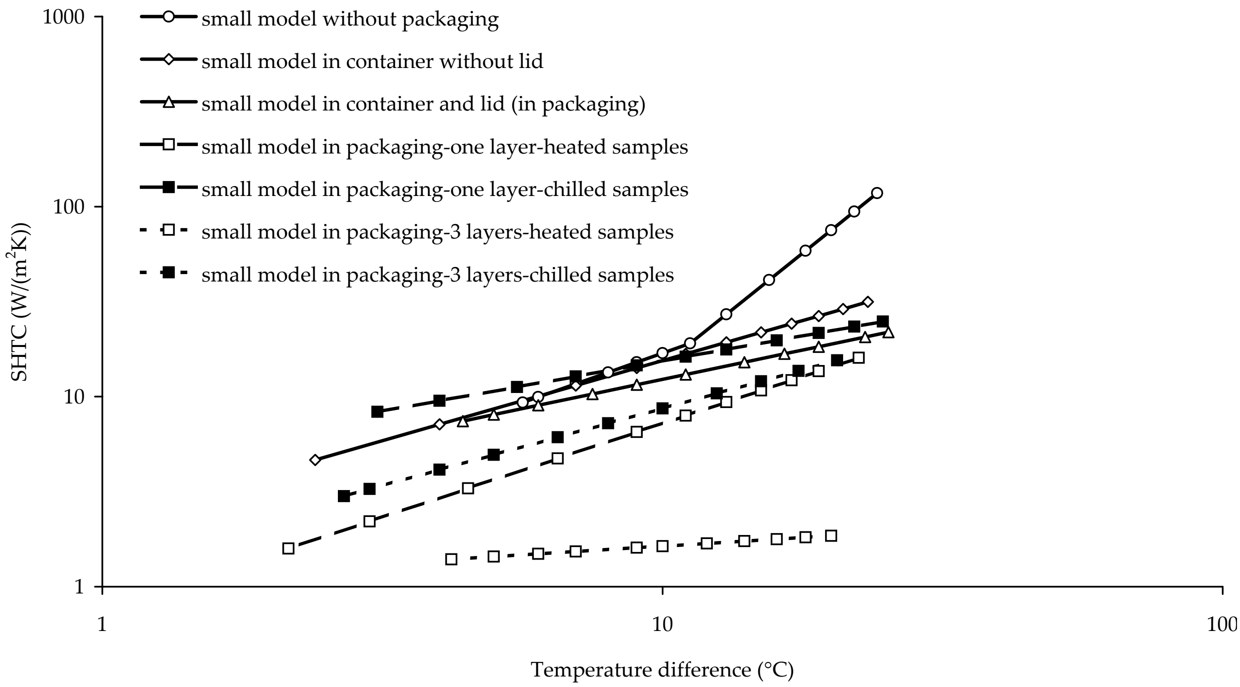

Figure 7. SHTC versus temperature difference for the small model in various arrangements

the model in packaging without the lid exhibit the largest values of SHTC. The curves for these two cases cross over at the temperature difference of about 15°C. Similar but slightly lower values were exhibited by the arrangements of model in packaging with the lid and the model placed in one layer between chilled samples. Much lower values of SHTC were predicted for the model placed in one layer of heated samples and that placed into the centre of three layers of chilled samples. The lowest values of SHTC, nearly independent of temperature difference, were found for the model placed into the centre of three layers of heated samples.

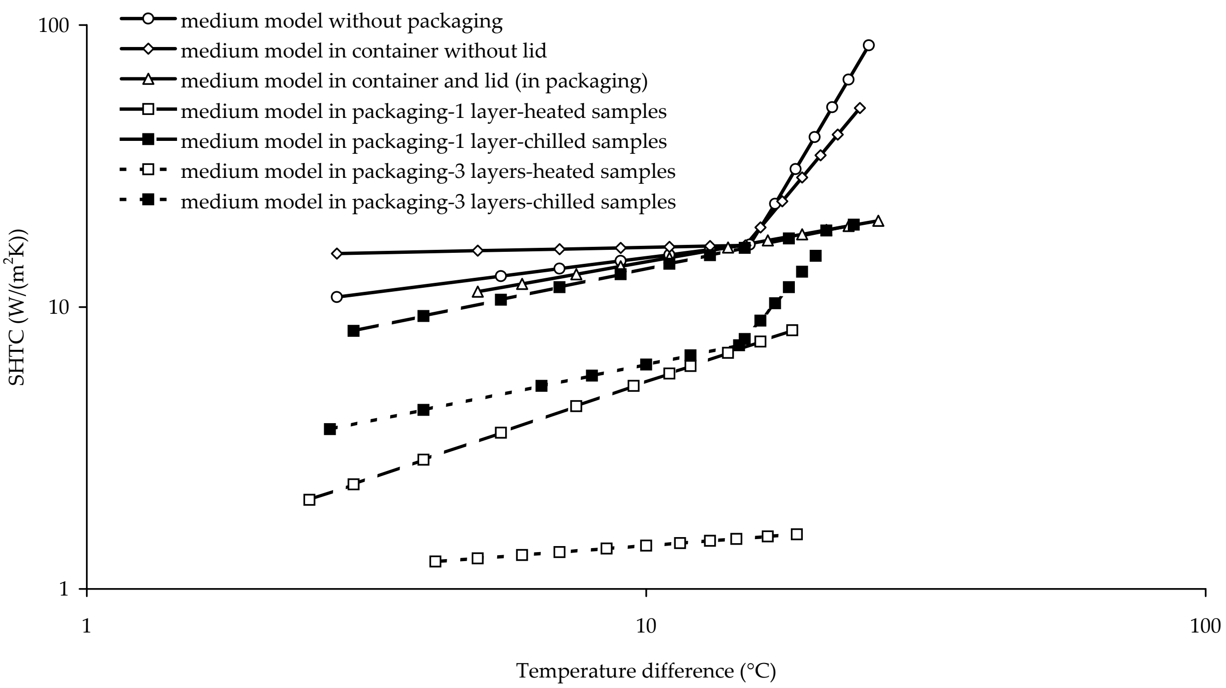

Figure 8. SHTC versus temperature difference for the medium size model in various arrangements

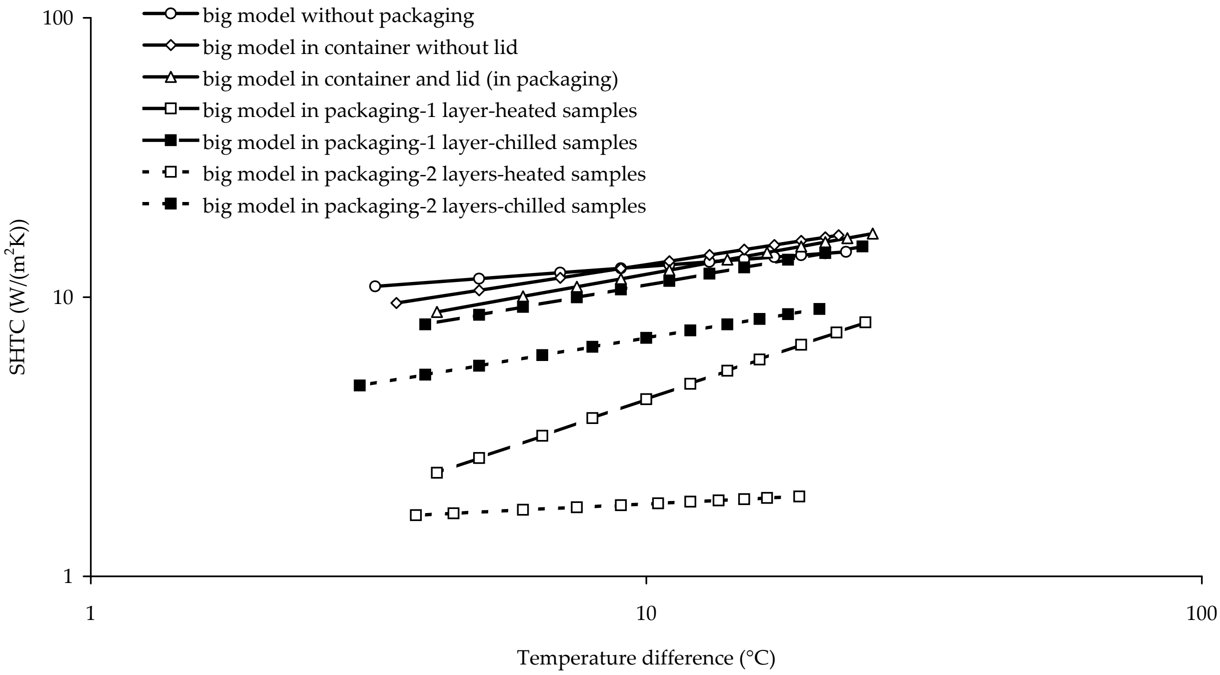

Figure 9. SHTC versus temperature difference for the big model in various arrangements

Figure 9 is given to the big model. The tendencies are similar to those with the medium and small models. The highest but very similar values of SHTC have been predicted for the model without packaging, in the containers without and with the lid. Similar values were found for the model in packaging placed into one layer of chilled samples. Much lower values of SHTC were found for the model placed between two layers of chilled samples (the model was placed in the second layer and not on the metal shelf of the cabinet). Lower values than these were found with the model placed into one layer of heated samples. The lowest values of SHTC were found with the model placed into two layers of heated samples (the model was placed in the second layer, i.c. not on the metal shelf of the cabinet). In this case, the apparent SHTC was nearly independent of the temperature difference (Table 4).

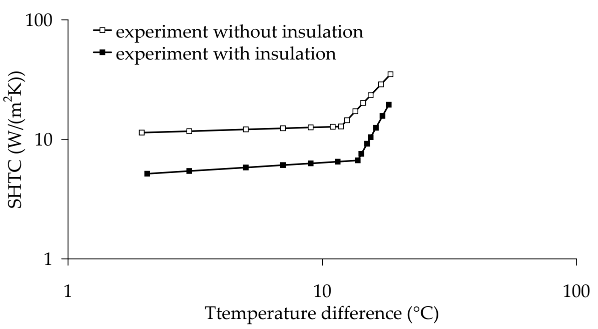

Figure 10. Results with the medium size model in a container without lid – showing the influence of positioning on the SHTC

CONCLUSIONS

The apparent SHTC value α is dependent on the time elapsed after placing a model into the chilled air of a display cabinet. The radiation, free convection, and conduction into the shelves are dominant heat transfer mechanisms in the cases studied.

An intensive unsteady heat transfer was observed with small and medium size models at the initial stages of the cooling. This effect was caused probably by the conduction from the model to the shelf. This effect was observed for temperature differences greater than 15°C.

The highest values of the apparent SHTC were received with the model without packaging placed individually on the display cabinet shelf.

The apparent SHTC is lowered only slightly by packaging. The metal models in packaging with or without the lid exhibited nearly the same dependencies on the temperature difference as the models without packaging or the models placed in one layer between chilled samples.

Surrounding the model with other containers lowered the SHTC substantially, both in a single layer and in the cases when the small and medium size models were placed in the centre of three layers of containers.

The lowest values of the apparent SHTC were found with small and medium size metal models placed in the centre of three layers of samples. The values of SHTC approach 1 W/(m^{2}K), regardless of the temperature difference.

Therefore, to load several layers of heated products into the display cabinet can be considered as a very bad practice. In such a case, the cooling of the central piece can last even for several hours. Obviously, the placement of heated products directly into the display cabinet cannot be recommended at all.

Nomenclature

a – thermal diffusivity of plate (m^2/s)
a, b, c, d, e – coefficients of Eq. (16)
c_p – specific heat of the metal model (J/(kgK))
h – thickness of plate (m)
H – height of sample (m)
m – mass of the metal model (kg)
$Nu = \alpha H/\lambda$ – Nusselt number
p – Laplace transform variable
$q = \sqrt{\frac{R^2 p}{a}}$ – auxiliary variable
p, n – empirical constants in Eq. (17)
R – cylinder radius (m)
r^* – radial coordinate (m)
r – dimensionless radial coordinate
$\mathrm{Ra} = g\beta H^3 \Delta T/(a\nu)$ – Rayleigh number
S – surface of the metal model (m^2)
t – time (s)
T_m – temperature of the metal model (°C)
T_0 – cool cabinet walls (°C)
T_p – temperature of air surrounding the model or experimental arrangement (°C)
T_s – surface temperature (°C)
α – surface heat transfer coefficient SHTC (W/(m^{2}K))
β – coefficient of thermal expansion (1/K)
γ – a real value which is greater than the real part of all poles (inversion theorem)
φ – relative heat capacity of the plate and sample
λ – thermal conductivity (W/(mK))
ν – kinematic viscosity (m^2/s)
θ – dimensionless temperature of plate
θ_c – dimensionless temperature of cylinder
ρ – density (kg/m^3)
σ – Stefan-Boltzmann constant (= 5.6697×10^{-8} W/(m K^4))
τ – time constant (s)

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