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Thermal analysis of above-grade wall assembly with low emissivity materials and furred airspace

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Abstract

A 3D numerical model was developed to investigate the effect of foil emissivity on the effective thermal resistance of an above-grade wall assembly with foil bonded to wood fibreboard in a furred assembly having airspace next to the foil. This model solved simultaneously the energy equation in the various material layers, the surface-to-surface radiation equation in the furred airspace assembly, Navier-Stokes equation for the airspace, and Darcy and the Brinkman equations for the porous material layers. In this work, the furring was installed horizontally. In the first phase, the present model was benchmarked against the experimental data generated by a commercial laboratory for an above-grade wall assembly. The wall consists of a conventional wood frame structure sheathed with fibreboard and covered on the interior side with a low emissivity material bonded to wood fibreboard that is adjacent to a furred airspace assembly. The results showed that the predicted R-value was in good agreement with the measured one. After gaining confidence in the present model, it was used to predict the effective thermal resistance of the same above-mentioned wall but having Oriented Strand Board (OSB) sheathing in lieu of wood fibreboard sheathing. In the second phase, the model was used to quantify the contribution on the wall R-value by having a low foil emissivity. The results showed that a low foil emissivity of 0.04 can increase the R-value of this wall to as much as ~9%. This is on-going research. The present model is being used to investigate the transient thermal response of foundation wall systems with furring installed horizontally and vertically, and subjected to different Canadian climate conditions.

Keywords: Above-grade wall system, furred airspace, low emissivity material, thermal modelling, thermal resistance, R-value, flow in porous media, heat transfer by convection, conduction and radiation.

1. Introduction

A 2D and 3D hygrothermal model called “hygIRC-C” that uses COMSOL Multiphysics [1] was developed at the National Research Council of Canada’s Institute for Research in Construction (NRC-IRC). This model simultaneously solves the highly nonlinear 2D and 3D Heat, Air and Moisture (HAM) transport equations. These equations were discretized using the Finite Element Method (FEM). This model was benchmarked against the hygIRC-2D model that was previously developed at NRC-IRC, and test results in a number of projects.

In the case of accounting for heat, air and moisture transport, the 2D version of the present model was used to predict the drying rate of a number of full-scale wall assemblies subjected to different exterior and interior boundary conditions [2]. The results showed that the present model and the hygroIRC-2D model [3 and 4] as well as the experimental measurements were in good agreement in terms of the shapes of the drying and drying rate curves. Additionally, the predicted average moisture content of the different wall assemblies over the test periods were in good agreement, all being within $\pm 5\%$ of those measured [2].

In the case of accounting for heat and air transport (no moisture transport), the 3D version of the present model was used to conduct numerical simulations for different full-scale wall assemblies with and without penetration to represent a window in order to predict the effective thermal resistance (R-value) with and without air leakage [5]. These walls incorporated different types of insulation, specifically, spray polyurethane foams and glass fibre batts. The predicted R-values for these walls were in good agreement (within $\pm 5\%$) with the measured R-values in NRC-IRC's Guarded Hot Box (GHB) [6 and 7].

More recently, the present model was used to assess the dynamic heat transmission characteristics through two Insulating Concrete Form (ICF) wall specimens installed in the NRC-IRC's Field Exposure of Walls Facility (FEWF) [8]. In the early stage of this project, 3D numerical simulations were conducted using the present model to determine the needed thickness of thermal insulations to be used around and between the two ICF wall specimens in order to minimize the thermal interaction between the wall specimens and the rest of the FEWF test bay. After running the experiment, the present model was used to interpret the readings from the instrumentation and to improve the experimental design by repositioning the instrumentation at critical locations. Thereafter, the numerical results were compared with the measurements. The results showed that the present model predictions were in good agreement with experimental data [8].

For foundation wall systems, airspace can contribute in obtaining a higher R-value, if a reflective material such as aluminum foil is installed on one side or both sides of a furred airspace. However, for such assembly, the question was: what might be the increase in R-value as a function of the foil emissivity? To answer this question, the present model was used to conduct sensitivity analyses in order to investigate the effect of foil emissivity for foil laminated to expanded polystyrene foam when used within a furred airspace assembly (see Figure 1, [9]). In that work, furring strips made of spruce (19 mm x 38 mm) were installed horizontally. Unlike the work conducted in this paper using the 3D version of the present model (see the next section), there were no vertical studs in the wall assembly (i.e. no thermal bridging). As such, the 2D version of the present model was suitable for that study [9]. Replacing the horizontal furring shown in Figure 1 by vertical ones, however, requires 3D simulation to capture the thermal bridging due to vertical furring.

For the wall system shown in Figure 1 [9], the results showed that a foil with lower emissivity has two interactive and competing effects on the wall R-value, namely: (i) an increase in wall R-value due to lower net radiative heat flux in the furred airspace assembly, and (ii) a decrease in R-value due to stronger convection currents in the airspace. The former effect was found to outweigh the latter effect, resulting in a net increase in the effective R-value for a wall system with low foil emissivity. Additionally, the results showed that the modelled foundation wall system with foil of emissivity 0.05 increased the effective R-value by about $\sim 10\%$ in the case of the outdoor and indoor temperatures of $+20^{\circ}\text{C}$ and -20°C , respectively [9].

Figure 1 Wall system with multiple enclosures [9]

Six months later, two standard tests were conducted by a commercial laboratory [10] to determine the effective thermal resistance of 2" by 6" wood frame structure sheathed with a fibreboard and covered on the interior side with a low emissivity material bonded to a wood fibreboard that is adjacent to a furred airspace assembly (see Figure 2). The first test was conducted to measure the emissivity of the foil in accordance with ASTM C 1371 "Standard test method for determination of emittance of material near room temperature using portable emissometers [11]. The second test was conducted to determine the effective thermal resistance of the above-mentioned wall assembly in accordance with ASTM C 1363 "Standard Test Method for the Thermal Performance of Building Assemblies by Means of a Hot Box Apparatus" using full-scale wall specimen [12].

The objectives of this paper are to: (a) benchmark the present model against experimental data generated by a commercial laboratory [10], (b) quantify the contribution of low emissivity material to the wall R-value, (c) predict the effective thermal resistance of the above-mentioned wall having an OSB sheathing in lieu of a wood fiberboard, and (d) investigate the effect of a wide range of foil emissivity on the wall thermal resistance.

2. Problem Description

A schematic of a furred airspace within an above-grade 2" by 6" wood frame structure is shown in Figure 2. As shown in this figure, the stud cavities were filled with glass fibre batts. The heat

flow through this wall system involves a multi-dimensional heat exchange within the wall assembly and multi-dimensional airflow in the furred airspace. The multi-dimensional effect on both the heat transfer and airflow is due to the effect of thermal bridges that are created by the vertical wood studs (note that thermal conductivity of wood studs is about 2.3 times the thermal conductivity of glass fibre batts, see [7]). As such, there is a need to develop a 3D model to predict the thermal performance of wall systems such as the one shown in Figure 2 below.

Figure 2. An above-grade wall with foil bonded to wood fibreboard in a furred assembly

Figure 3 shows the modes of heat transfer in a furred airspace for the case of the indoor temperature greater than the outdoor temperature. As shown in this figure, there are three modes of heat transfer in the airspace, namely:

- (a) Radiation: A net radiative heat is transferred from the hot surface (gypsum board) to the cold surface (foil);

- (b) Conduction: A net conductive heat is transferred from the hot surface to the cold surface, and;
- (c) Convection: Since the gypsum board surface has a higher temperature than the foil surface, and taking into consideration the buoyancy effect, the air adjacent to the former moves upward, while the air adjacent to the latter moves downward, resulting in convection current that in turn increases the heat transfer in the airspace enclosure.

Case of indoor temperature > outdoor temperature

Figure 3. Modes of heat transfer in furred airspace assembly

As described in the next section, in order to model the heat transfer by radiation, convection and conduction in the airspace (Figure 2 and Figure 3), the energy equations in the solid material layers and airspace, surface-to-surface heat radiation equation in the furred airspace assembly, and the coupled mass balance and momentum equations of the air in the airspace and porous

materials were solved simultaneously. In the meantime, the opacity of the airspace must be considered as transparent so that the radiation rays from the surface boundaries of the airspace enclosure are not blocked. Since there is no radiation through the solid material layers, they are treated as opaque bodies.

3. Governing Equations

The present model solves the following governing equations:

3.1. Continuity and Momentum Equations

In the airspace, the air velocity field, $\vec{v}_a$, and pressure field, P_a , is calculated by solving the continuity equation and momentum equation (Navier-Stokes equation). These equations are respectively given as [13]:

$$\frac{\partial \rho_a}{\partial t} = -\nabla \cdot \rho_a \vec{v}_a, \text{ and} \quad (1)$$

$$\rho_a \frac{\partial \vec{v}_a}{\partial t} + \rho_a \vec{v}_a \cdot \nabla \vec{v}_a = -\nabla P_a + \nabla \cdot \left(\mu_a (\nabla \vec{v}_a + (\nabla \vec{v}_a)^T) - \frac{2}{3} \mu_a (\nabla \cdot \vec{v}_a) I \right) + \rho_a \vec{g}. \quad (2)$$

For airflow through porous material, the mass and momentum balance equations are given as [13]:

$$\text{Mass Balance: } \varepsilon_p \frac{\partial \rho_a}{\partial t} = -\nabla \cdot \rho_a \vec{v}_a, \text{ and} \quad (3)$$

$$\text{Darcy's law: } \vec{v}_a = -\frac{\kappa_a}{\mu_a} (\nabla P_a - \rho_a \vec{g}). \quad (4)$$

Where ε_p is the porosity (ratio of pore volume to total volume). The air superficial velocity, $\vec{v}_a$, sometimes called Darcy velocity, is the volume rate of air flow through a unit cross-sectional area of the solid plus fluid, which is averaged over a small region of space (small with respect to macroscopic dimensions in the flow system but large with respect to the pore size). The present model handles the air permeability, κ_a , as scalar quantity for isotropic materials and a tensor quantity for anisotropic porous materials. Moreover, the air density, ρ_a , and pressure, P_a , are averaged over a region available to airflow that is large with respect to the pore size.

An empirical modification of Darcy's law has been suggested by Brinkman as [13]:

$$0 = -\nabla P_a - \frac{\mu_a}{\kappa_a} \vec{v}_a + \mu_{a,eff} \nabla^2 \vec{v}_a + \rho_a \vec{g}. \quad (5)$$

In this modification, the term $\mu_{a,eff} \nabla^2 \vec{v}_a$ was introduced to account for the distortion of velocity profiles for which the momentum transport within the fluid due to shear stresses is of importance. The parameter $\mu_{a,eff}$ in Eq. (5) is the effective viscosity that theoretically takes into account the stress borne by the fluid as it flows through the porous media. However, experimental measurements of $\mu_{a,eff}$ is not a trivial matter, if not impossible [14]. In accordance with the published literature (e.g. see [15]), $\mu_{a,eff}$ is set to be equal to the fluid viscosity, μ_a .

For flow in a porous media, a parameter called Darcy number, D_N , is introduced to define the type of flow regime within the porous media. The D_N is a dimensionless parameter, which is defined as the ratio of medium permeability (κ_a) to the square of domain thickness, δ_{do} ($D_N = \kappa_a / \delta_{do}^2$). In the Darcy model, it is effectively assumed that all of the stress in the flow field is carried by the porous media and the fluid is not subjected to any strain because of the viscous stresses. This is a good assumption for flow through a low permeability porous media where the flow velocity is small (i.e. Darcy flow). However, this assumption cannot be regarded to be physically realistic for high permeability porous media where a part of the viscous stress is borne by the fluid itself, which is accounted for by the term $\mu_{a,eff} \nabla^2 \vec{v}_a$ in the Brinkman equation (see Eq. (5)). Therefore, the Brinkman equation which accounts for the transition from Darcy flow to viscous flow should be used for a flow through a high permeability porous media. This topic has been studied by many investigators [14, 15, 16, and 17].

The present model was used to conduct a simple numerical test in order to identify the different flow regimes in a porous media. In this test, a porous layer (140 mm thick) was sealed at the top and bottom and subjected to a pressure difference across it (see the insert in Figure 4). In order to cover a wide range of D_N , the numerical test was conducted using a wide range of air permeability. Figure 4 shows the dependence of the airflow ratio, Ψ_{Flow} , on the Darcy number, D_N . Ψ_{Flow} is defined as the ratio of net airflow rate obtained from solving the Brinkman equation (Eq. (5)) to that obtained from solving Darcy's law (Eq. (4)). As shown in this figure, the airflow rate obtained from solving Darcy's law and the Brinkman equation are equal (i.e. $\Psi_{Flow} = 1.0$) for flow through a porous media with low permeability ($D_N \leq 10^{-6}$, Darcy flow regime), where the effect of the viscous stress term in Brinkman equation is insignificant. However, for flow through a porous media with high permeability ($D_N > 10^{-6}$, viscous flow regime), Darcy's law over predicts the airflow rate (i.e. $\Psi_{Flow} < 1.0$). According to Parvazinia et al. [15] and the results shown in Figure 4, the flow can be described by Darcy's law for $D_N \leq 10^{-6}$. However, for $D_N > 10^{-6}$, the flow should be described by the Brinkman equation.

In the present model, for a given wall assembly with different types of porous materials, there are two options to solve for the airflow through these materials. Either solve the Brinkman equation for any porous material (since this equation is applicable for both Darcy flow and viscous flow regimes) or solve the Darcy's law for porous materials with $D_N \leq 10^{-6}$ and the Brinkman equation for porous materials with $D_N > 10^{-6}$. Note that the accurate prediction of the airflow through porous media is essential to accurately account for the moisture transport (not considered in this work) and energy transport by convection through these materials as explained next.

Figure 4. Dependence of the airflow ratio, Ψ_{Flow} on Darcy number

3.2. Energy Equation

The general energy equation in the airspace is given as [13]:

$$\rho_a C p_a \frac{\partial T}{\partial t} = \underbrace{-\rho_a C p_a (\vec{v}_a \cdot \nabla T)}_{\text{convection}} + \underbrace{\nabla \cdot (\lambda_a \nabla T)}_{\text{conduction}} + \underbrace{\eta_a \left((\nabla \vec{v}_a + (\nabla \vec{v}_a)^T) - \frac{2}{3} (\nabla \cdot \vec{v}_a) I \right) : \nabla \vec{v}_a}_{\text{viscous heating}} - \underbrace{\frac{T}{\rho_a} \frac{\partial \rho_a}{\partial T} \bigg|_p \left(\frac{\partial P_a}{\partial t} \right)}_{\text{Pressure work}} + q'''_{\text{source/sink}} \quad (6)$$

Where $q'''_{\text{source/sink}}$ represents the heat source/sink (e.g. due to moisture condensation/evaporation), which is neglected since no moisture transport was considered in this work. In typical building applications, the contribution due to pressure work and viscous

heating are much smaller than that due to convection and conduction. Neglecting these terms in Eq. (6), the resulting energy equation becomes:

$$\rho_a C p_a \frac{\partial T}{\partial t} = -\rho_a C p_a (\vec{v}_a \bullet \nabla T) + \nabla \bullet (\lambda_a \nabla T). \quad (7)$$

In porous materials, the energy equation after neglecting the heat source/sink ($q'''_{\text{source/sink}}$) is [6 and 7]:

$$\rho_o C p_{eff} \frac{\partial T}{\partial t} = -\rho_o C p_a (\vec{v}_a \bullet \nabla T) + \nabla \bullet (\lambda_{eff} \nabla T). \quad (8)$$

The surface-to-surface heat radiation equation for the furred airspace assembly (derived in the next section) coupled with Eqs. (1), (2) and (7) for airspace, and Eqs. (3), (4) for $D_N \leq 10^{-6}$, (5) for $D_N > 10^{-6}$ and (8) for porous material layers are solved simultaneously at the steady-state condition for pressure, P_a , velocity, $\vec{v}_a$, and temperature, T .

3.3. Radiative Heat Flux Equation in Enclosure

This section describes the theory behind the radiative heat transfer process in the building envelope. Thermal radiation denotes the stream of electromagnetic waves emitted from a body at a certain temperature. For opaque surfaces, no radiation is transmitted through the body. The incoming radiative heat flux at a location on the surface is called the irradiation, q_{irr} . A portion of the irradiation is reflected from the surface. This portion depends on the surface reflectivity, ρ_r . Additionally, the surface emits a thermal flux that depends on both its emissivity, ε , and temperature, T_s . The total outgoing radiative heat flux at this location is called the radiosity, q_{radi} . The radiosity is the sum of the reflected radiation and emitted radiation from the surface, which is given as:

$$q_{radi} = \rho_r q_{irr} + \varepsilon \sigma T_s^4. \quad (9)$$

The net inward radiative heat flux, q_{RAD} , is the difference between the irradiation and radiosity:

$$q_{RAD} = q_{irr} - q_{radi} = (1 - \rho_r) q_{irr} - \varepsilon \sigma T_s^4. \quad (10)$$

In the building envelope, most opaque bodies behave as gray bodies. As such, the long wave absorptivity, α , and emissivity are equal, and the reflectivity, ρ_r , is therefore given as:

$$\rho_r = 1 - \varepsilon = 1 - \alpha. \quad (11)$$

From Eqs. (10) and (11), the net inward radiative heat flux for gray bodies is given as:

$$q_{RAD} = q_{irr} - q_{radi} = \varepsilon (q_{irr} - \sigma T_s^4). \quad (12)$$

Surface-to-surface radiation is complex as mutual radiation from all adjacent surfaces must be taken into consideration when determining the net radiative flux at a given surface. An example of surface-to-surface is enclosed space with no openings (e.g. a furred airspace assembly in a wall system, see Figure 3). In the case of enclosure with opening(s) to the ambient surroundings, however, net radiative flux at a given surface is determined by taking into consideration radiation from both the ambient surroundings and from other surfaces.

To derive a general expression for surface-to-surface radiation, consider an enclosure with an opening to ambient surroundings as shown in Figure 5. This enclosure is formed from different surfaces with different materials. Consider an infinitesimal surface area dA of a

target located at arbitrary point h on the surface S_4 of material mat_4 . This point can see points on the other surfaces and the ambient surroundings as well. Each point on all surfaces has a local radiosity, $q'_{\text{radi}}(\vec{r}')$. Assume the ambient surroundings have a constant emissivity, ε_{amb} , and temperature, T_{amb} . The mutual irradiation, G_m , at an arbitrary point h is given by the following surface integrals as [18]:

$$G_m = \sum_{i=1}^m \int_{S_i} K_s(\vec{r}, \vec{r}') q'_{\text{radi}}(\vec{r}') dS, \quad m = \text{number of surfaces.} \quad (13)$$

Where $K_s(\vec{r}, \vec{r}')$ is the kernel function, which for a 3D problem is given as [18]:

$$K_s(\vec{r}, \vec{r}') = \frac{(-\vec{n}' \cdot (\vec{r} - \vec{r}'))(\vec{n} \cdot (\vec{r} - \vec{r}'))}{\pi |\vec{r} - \vec{r}'|^4} \chi. \quad (14)$$

Figure 5. A schematic of surface-to-surface radiation for enclosure with opening to ambient

In Eq. (14), the parameter χ has a value of 1.0 when the infinitesimal surface area dA at the target (point h) sees the infinitesimal surface area dA' at the source (point d) as illustrated by the ray connecting points h to d (i.e. $\vec{r} - \vec{r}'$), and otherwise it is 0 if the ray $\vec{r} - \vec{r}'$ is blocked (see Figure 5). Note that for a given surface the parameter χ can have a value of 1.0 on a portion of this surface and 0 on the other portion of the same surface. For example, for a target located at point h as shown in Figure 5, the parameter χ for the surface S_1 has a value of 1.0 on the portion of this surface from b to c , and a value of 0 on the portion from a to b .

In order to account for the mutual irradiation due to the ambient surroundings at a target located at point h , the ambient view factor, F_{amb} , needs to be determined as follows:

$$F_{amb} = 1 - \sum_{i=1}^m F_{S_i}, \text{ with } F_{S_i} = \int_{S_i} K_S(\vec{r}, \vec{r}') dS. \quad (15)$$

In the case of a 2D problem, the mutual irradiation at point h is given by the line integrals:

$$G_m = \sum_{i=1}^m \int_{L_i} K_L(\vec{r}, \vec{r}') q'_{radi}(\vec{r}') dL, \text{ m = number of line boundaries.} \quad (16)$$

Where the kernel function for a 2D problem is given as [18]:

$$K_L(\vec{r}, \vec{r}') = \frac{(-\vec{n}' \cdot (\vec{r} - \vec{r}'))(\vec{n} \cdot (\vec{r} - \vec{r}'))}{2 |\vec{r} - \vec{r}'|^3} \chi. \quad (17)$$

The ambient view factor in the case of 2D geometry is given as:

$$F_{amb} = 1 - \sum_{i=1}^m F_{L_i}, \text{ with } F_{L_i} = \int_{L_i} K_L(\vec{r}, \vec{r}') dL. \quad (18)$$

The general equation for the total irradiative heat flux arriving at a target at an arbitrary point on a surface (3D geometry) or line boundary (2D geometry) can now be obtained by summing up the contribution of the mutual irradiation from all surfaces (3D geometry) or all line boundaries (2D geometry), and the ambient surroundings as follows:

$$q_{irr} = G_m + F_{amb} \varepsilon_{amb} \sigma T_{amb}^4. \quad (19)$$

In the case of a 3D problem, G_m and F_{amb} are given by Eqs. (13) and (15), respectively.

Whereas, in the case of a 2D problem, G_m and F_{amb} are given by equations (16) and (18), respectively. For an enclosure with no openings to ambient surroundings, the last term on the RHS of Eq. (19) is zero (since $F_{amb} = 0$).

In the building envelope, a given problem may have more than one enclosure. Figure 2 shows an example of multiple airspace enclosures for a wall system with horizontal furring. Each airspace enclosure shown in this figure is bounded by furring (spruce), gypsum board and a low emissivity sheet of material.

After deriving the expression for the total irradiation arriving at an arbitrary point on a surface (3D geometry) or a line boundary (2D geometry), q_{irr} (Eq. (19)), the radiosity emitted from this point can now be calculated by substituting the value of q_{irr} from Eq. (19) into Eq. (12):

$$q_{radi} = (1 - \varepsilon) (G_m + F_{amb} \varepsilon_{amb} \sigma T_{amb}^4) + \varepsilon \sigma T_s^4. \quad (20)$$

The above equation is applicable to all points on the surface/line boundaries that participate in surface-to-surface radiation. This equation forms a system of equations in radiosity q_{radi} . This system of equations is solved simultaneously with the energy equation (Eqs. (7) and (8)) for the temperature, T and radiosity q_{radi} .

4. Description of Wall Specimen

In this study, the generalized equations presented in section 3 for heat and air momentum transport were applied to a specific 3D problem of interest to industry. Figure 2 and Figure 6 show a schematic of one module of the opaque wall specimen. Due to symmetry, this module includes one half of the vertical stud and one half of the cavity. The full-scale wall specimen of

2,438 mm x 2,438 mm was built using 2" by 6" wood frame construction, made of spruce, spaced at 16" (406 mm) center-to-center with double top plates and single bottom plate. The wall cavities were filled with friction-fit glass fibre batts insulation. An insulation board made of wood fibre having a foil with Low Emissivity Material (LEM) on one side that was covered with a Furred Airspace Assembly (FAA). The insulation board was installed on the interior side of the wood frame structure. The FAA consisted of 19 mm x 64 mm wood furring strips installed horizontally at 16" (406 mm) center-to-center and was closed with a gypsum board. The external layer of the wall assembly is wood fibreboard (11 mm thick). The present model was used to predict the R-value of this wall. After gaining confidence in the present model, it was used to predict the R-value of the same above-mentioned wall but having Oriented Strand Board (OSB) sheathing (11 mm thick) in lieu of wood fibreboard sheathing.

Figure 6. Horizontal cross section passing through the airspace of the wall specimen

5. Test Results

Two tests were conducted by a commercial laboratory [10]. The first test was conducted to measure the emissivity of the foil in accordance with ASTM C 1371 "Standard test method for determination of emittance of material near room temperature using portable emissometers [11]. The measured foil emissivity was 0.04. The second test was conducted to determine the effective thermal resistance of the wall specimen shown in Figure 2 and Figure 6. This test was conducted in accordance with ASTM C 1363 "Standard Test Method for the Thermal Performance of Building Assemblies by Means of a Hot Box Apparatus" [12]. The test conditions and results are listed in Table 1. The R-value derived from the test results shown in Table 1 was 4.24 m²K/W.

6. Boundary Conditions

The boundary conditions on the top and bottom surfaces are adiabatic and sealed (i.e. no heat and mass transfer). Since only one module of the wall was modeled (see Figure 2), the left and right surfaces are also adiabatic and sealed due to symmetry. The exterior surface of the fibreboard sheathing is subjected to convective boundary condition with air temperature and heat transfer coefficient of -34.07°C and $22.40 \text{ W/m}^2\text{K}$, respectively (same as in the test, Table 1). Similarly, the interior surface of the gypsum board is subjected to a convective boundary condition with air temperature and heat transfer coefficient of 20.50°C and $7.08 \text{ W/m}^2\text{K}$, respectively. The static pressure difference across the wall specimen is taken equal 1.62 Pa (Table 1). The benchmarking of the present model against the experimental data is discussed next.

Table 1. Test conditions and results of above-grade wall system shown in Figure 2

Net Heat through Specimen	73.29 W
Specimen Projected Area	5.95 m^2
Net Heat Flux through Specimen	12.32 W/m^2
Average Metering Room Air Temperature	20.50°C
Average Cold Side Air Temperature	-34.07°C
Area-Weighted Warm Side Surface Temperature	18.76°C
Area-Weighted Cold Side Surface Temperature	-33.52°C
Room Side Heat Transfer Coefficient	$7.08 \text{ W/m}^2\text{K}$
Weather Side Heat Transfer Coefficient	$22.40 \text{ W/m}^2\text{K}$
Maximum Static Pressure across the wall specimen	1.62 Pa
Derived Wall Thermal Resistance (R-value)	$4.24 \text{ m}^2\text{K/W}$

7. Results and Discussions

In the numerical simulation, the emissivity of all materials of the wall assembly except the foil was assumed equal to 0.9. For the foil, however, the emissivity was taken equal to the measured value (0.04). Additionally, all material layers were assumed to be in perfect contact (i.e. the interfacial thermal resistances between all material layers were neglected). As shown in Table 1, the measured R-value was $4.24 \text{ m}^2\text{K/W}$. The predicted R-value using the present model was $4.19 \text{ m}^2\text{K/W}$, which is slightly lower than the measured one. The predicted R-value is within the uncertainty of the guarded hot box (see [19]). As such, the predicted R-value is in good agreement with the measured R-value.

In many practical applications, above-grade walls have OSB sheathing. The present model was used to predict the R-value of the same above-mentioned wall but having OSB sheathing (11 mm thick) in lieu of fibreboard sheathing (11 mm thick). Because the thermal conductivity of OSB (0.09 W/mK) is greater than that for fibreboard (0.055 W/mK) [20], the predicted R-value

for wall with OSB sheathing ($4.12 \text{ m}^2\text{K/W}$) was lower than wall with fibreboard sheathing ($4.19 \text{ m}^2\text{K/W}$). Note that the predicted R-value for a wall specimen with a foil of measured emissivity of 0.04 represents the highest thermal resistance for a clean wall system (i.e. no moisture condensation and/or dust accumulation of the foil surface).

After gaining confidence in the present model prediction, it was then used to quantify the contribution of the low emissivity foil to the wall R-value. In practical applications, the accumulation of dust (e.g. due to construction) and/or moisture condensation on the foil surface can cause an increase in its emissivity. A numerical simulation was conducted for an above-grade wall with OSB sheathing but with a foil emissivity of 0.9 (same as all surfaces of the airspace enclosure of furring and gypsum board). The case of high foil emissivity (0.9) may represent the case of: (a) no foil installed in the wall system, or (b) a significant portion of the foil surface covered by dust and/or thin liquid film of water due to moisture condensation.

Two competing effects must be considered when assessing the effective R-value of a wall with foil installed on the surface of the wood fibre board:

- (a) The net radiative heat flux from the surfaces of the furring and gypsum board to the surface of foil is lower in the case of foil with low emissivity than in the case of foil with high emissivity, resulting in an increase in wall R-value with low foil emissivity. This is a positive effect on the wall R-value.
- (b) The temperature difference across the airspace is larger in the case of foil with low emissivity (due to the low net radiative heat flux mentioned above) than in the case of foil with high foil emissivity. As such, the convection current of the air inside the airspace becomes stronger in the wall with lower emissivity, resulting in an increase in the heat transfer in the airspace due to convection. This is a negative effect on the wall R-value.

The above two interactive and competing effects result in a net increase in the effective R-value of a wall having a foil of low emissivity as compared to a wall with a foil of high emissivity. As shown in Figure 7, in the case of foil emissivity of 0.9 (in other words no foil installed in the wall system), the predicted R-value was $3.78 \text{ m}^2\text{K/W}$ ($21.45 \text{ ft}^2 \text{ hr } ^\circ\text{F/BTU}$). As such, the increase in the R-value due to installing a foil with emissivity of 0.04 is $0.34 \text{ m}^2\text{K/W}$ ($1.92 \text{ ft}^2 \text{ hr } ^\circ\text{F/BTU}$). This represents an increase of the wall R-value by ~9%.

A number of numerical simulations were also conducted to quantify the reduction in R-value due to different heat transfer modes such as radiation heat transfer in the airspace only and convective heat transfer in the airspace only. Two walls with Furred-Airspace Assembly (FAA) have been assessed, one with a foil emissivity of 0.04 and another with a foil emissivity of 0.9. In the case of neglecting the effect of radiation heat transfer and convective heat transfer in the airspace (see the bars of “no Rad + no Conv” in Figure 7), the predicted R-value was $4.28 \text{ m}^2\text{K/W}$ for both walls. Neglecting the radiation heat transfer and accounting for convective heat transfer in the airspace (see the bars of “no Rad + Conv” in Figure 7) resulted in a decrease in the R-value by $0.13 \text{ m}^2\text{K/W}$ for both walls. On the other hand, accounting for the radiation heat transfer and neglecting the convective heat transfer in the airspace (see the bars of “Rad + no Conv” in Figure 7) resulted in a decrease in the R-value by $0.06 \text{ m}^2\text{K/W}$ and $0.41 \text{ m}^2\text{K/W}$ for the cases of foil emissivity of 0.04 and 0.9, respectively.

For the case of foil emissivity of 0.04, the reduction in the R-value due to accounting for radiation heat transfer in the airspace only (Rad + no Conv) is about $\frac{1}{2}$ the reduction in the R-

value due to accounting for convection heat transfer in the airspace only (no Rad + Conv). For the case of foil emissivity of 0.9, however, the reduction in the R-value due to accounting for radiation heat transfer in the airspace only (Rad + no Conv) is about 3 times the reduction in the R-value due to accounting for convection heat transfer in the airspace only (no Rad + Conv). Accounting simultaneously for both radiation heat transfer and convective heat transfer in the airspace (see the bars of “Rad + Conv” in Figure 7), resulted in a decrease in R-value by 0.16 $\text{m}^2\text{K/W}$ and 0.50 $\text{m}^2\text{K/W}$ for the cases of foil emissivity of 0.04 and 0.9, respectively.

In previous studies [5, 6, 7], the present model was used to predict the R-values of wall systems with and without penetration (to represent a window), and with and without air leakage. The stud cavities of 2” by 6” wood frame construction of those walls were filled by different types of insulation, specifically, light density spray polyurethane foams and glass fibre batts. In the case of no air leakage, the wall with glass fibre insulation and without penetration is similar to the wall system considered in this study but without furred airspace assembly (FAA). The predicted R-value for that wall was 3.20 $\text{m}^2\text{K/W}$, which was in good agreement with measured R-value in the NRC-IRC’s GHB (3.25 $\text{m}^2\text{K/W}$) (see references [5, 6, 7] for more details). As shown in Figure 7, a FAA with foil emissivity of 0.04 resulted in an increase of R-value by 0.92 $\text{m}^2\text{K/W}$. While a FAA with foil emissivity of 0.9 resulted in an increase of R-value by 0.58 $\text{m}^2\text{K/W}$.

Figure 7. Contribution of Furred Airspace Assembly (FAA) with foil emissivity of 0.04 and 0.9 to the R-value of a foundation wall system (case of wall having OSB sheathing board)

In a previous study by the authors [9], the 2D version of the present model was used to conduct parametric study in order to investigate the effect of air temperature difference between the outdoor and indoor, ΔT_{amb} , (40°C and 60°C) on the effective wall R-value for a wide range of foil emissivity of 0 – 0.8 (see Figure 1 for the wall description). It was shown that in the range of foil emissivity of 0 – 0.4, the wall R-value decreases considerably as the foil emissivity increases. For foil emissivity > 0.4, however, the wall R-value still decreases as the foil emissivity increases but at a reduced rate. For example, a wall with foil emissivity of 0.05 has an increase in R-value by ~10% and ~9% for ΔT_{amb} of 40°C and 60°C, respectively, compared to wall with higher foil emissivity of 0.8 (see [9] for more details). Furthermore, in this study, ΔT_{amb} of 54.57°C (between -34.07°C and 20.50°C, see Table 1) results in an increase of wall R-value by ~9% (see the next section for more details), which is in a good agreement with the previous study [9]. Accordingly, for airspace size of 19 mm thick and given ΔT_{amb} , the percentage increase in the R-value due to having foil with different emissivity could be obtained using either 2D simulation or 3D simulation. Therefore, 2D simulation and 3D simulation could be conducted to predict the R-value in the case of no foil for walls with no thermal bridges (e.g. see Figure 1) and with thermal bridges (e.g. see Figure 2), respectively. After that the apparent R-value (i.e. wall with foil) can be calculated by adding up the contribution of the foil for a given emissivity. Because the previous study [9] and the present study were conducted in the case of only one size of airspace (19 mm thick), the question is: does the percentage increase in the R-value obtained from 2D simulation and 3D simulation due to having foil with different emissivity will be the same for the same airspace of different sizes? This is on-going research and a number of numerical simulations will be conducted to address this question.

A close examination for the different modes of heat transfer and airflow in the airspace and porous material layers in the wall specimen shown in Figure 2 are discussed. Figure 8 shows the temperature contours through two main thermal paths: section A-A (passing through the middle of wood stud) and section C-C (passing through the middle of stud cavity). In order to look closely at the temperature distribution around the furred airspace assembly of the wall specimen, only the temperature isotherms between 5°C and 19°C is shown in this figure. Since the temperature of the gypsum board is higher than the temperature of the foil surface (see Figure 8), the air adjacent to the former travels upward while the air adjacent to the latter travels downward, forming convection current (loop) inside the airspace. This convection current is shown in Figure 9. Note that the effective thermal conductance of the airspace increases with higher air velocity.

Because of the thermal bridging caused by the vertical wood stud, the isotherm line of 5°C approximately touches the exterior surface of the wood fibreboard in the slice passing through the middle of the stud (Figure 8(a)) and moves further inside the glass fibre in the slice passing through the middle of the stud cavity (Figure 8(b)), resulting in a higher temperature gradient in the former than that in the latter. As such, the convection current of the air inside the airspace is stronger in the vertical slice passing through the middle of the stud (Figure 9(a)) than that in the slice passing through the middle of the stud cavity (Figure 9(b)). For example, the maximum downward velocity (close to the surface of the foil) of the air in slice passing through the middle of stud was ~71 mm/s, which is ~twice the air velocity in the slice passing through the middle of the stud cavity (~35 mm/s).

The results shown in Figure 8 and Figure 9 clearly show the multi-dimensional effects of thermal bridging caused by the vertical stud on both the energy transport through the wall system and momentum transport of the air inside the furred airspace assembly. These multi-dimensional effects were successfully captured using the 3D version of the present model. However, as indicated earlier, for a wall system with furred airspace assembly (horizontal furring only) and no vertical studs (i.e. no thermal bridging, [9]), the 2D version of the present model was suitable for predicting the thermal performance of the wall system shown in Figure 1 [9].

Figure 8. Temperature contours (in °C) in vertical slices through sections A-A and C-C shown in Figure 6 (foil emissivity = 0.04)

Figure 9. Vertical velocity contours (in mm/s) in vertical slices through sections A-A and C-C shown in Figure 6 (foil emissivity = 0.04).

Figure 10 shows the contours of the vertical velocity in three slices located at sections A-A, B-B, and C-C (see in Figure 6). In order to look closely to the flow in porous media and given that the air velocity in the airspace is much higher than that in the porous materials of the wall specimen, this figure shows the contours of the vertical velocity in all wall components except the airspace. As indicated earlier the pressure difference across the wall specimen was 1.62 Pa. Provided that there is no deficiencies in the wall specimen (e.g. no cracks), this pressure difference is very small and cannot cause airflow across the wall specimen due to the low air permeability of both the OSB ($3.12\text{E-}14 \text{ m}^2$) and the gypsum board ($8.40\text{E-}14 \text{ m}^2$) (i.e. no infiltration or exfiltration). However, the existing air in the pores of the glass fibre batts can

Figure 10. Contours of the vertical velocity (in mm/s) at sections A-A, B-B and C-C shown in Figure 6

move by natural convection due to its high air permeability ($5.83\text{E-}10 \text{ m}^2$). As shown in Figure 10(a), the vertical velocity in the slice passing through the middle of the stud (section A-A) is zero. Due to the effect of thermal bridging caused by the stud, the temperature gradient in the portion of glass fibre near to the stud (e.g. see section B-B in Figure 6) is higher than that in the portion of the glass fibre far away from the stud (e.g. see section C-C in Figure 6). As such, the convection current of the air is stronger in the vertical slice passing through section B-B (Figure 10(b)) than that in the vertical slice passing through section C-C (middle of stud cavity, Figure

10(c)). For example, the maximum downward velocity of the air in section B-B is 0.038 mm/s, which is 70% higher than that in section C-C (0.022 mm/s).

Although the convection current of the air through the porous layers of the wall specimen considered in this study (no deficiencies, no cracks, etc) has insignificant effect on the effective R-value, it can play an important role on the moisture transport (not considered in this study) through the wall system [21]. On the other hand, the convection current of the air in porous media can cause a significant effect on the R-value of a wall system with deficiencies and/or 3 mm air gap at the mid height of the OSB sheathing and electrical outlet box in the gypsum board, when the wall is subjected to pressure difference across it. This effect depends on the air leakage rate through the wall system (see references [5, 6 and 7] for more details).

Since the objective of this paper is to predict the wall R-value, the numerical simulations were conducted only at the steady-state condition. On the other hand, the present model was recently used to conduct numerical simulations in order to investigate the transient thermal response of full-scale foundation wall systems (including the above-grade and below-grade portions of the wall and taking into consideration the temperature distribution in the soil) with low emissivity material and furred airspace assembly and without furred airspace assembly. These walls were subjected to climate conditions of Ottawa, Canada [22].

8. Dependence of R-Value on Foil Emissivity

Figure 11 shows the effect of foil emissivity on the effective R-values of wall foundation system when the difference between the indoor and outdoor temperatures, ΔT_{amb} , is 54.57°C (see Table 1). As shown in this figure, the wall R-value decreases considerably as the foil emissivity increases from 0 to 0.45. In the range of foil emissivity of 0.45 – 0.9, however, the wall R-value slowly decreases as the foil emissivity increases. For example, the wall R-value decreases by 0.31 m²K/W, and 0.08 m²K/W as the foil emissivity increases from 0 to 0.45, and from 0.45 to 0.9, respectively.

As shown in Figure 8, there is large temperature difference between the gypsum board and foil surface (~13 K). Also, this figure shows that the foil surface temperature changes locally with the height in between two furrings. As such, there is a risk for local moisture condensation on the foil surface. This risk depends on both the relative humidity of the air inside the furred airspace assembly and the local surface temperature of the foil. As indicated earlier, a local moisture condensation on the foil would result in an increase in its emissivity. Although we intended to predict the thermal performance only (no moisture transport is accounted for), the present model was used to predict the wall thermal resistance for a wide range of foil emissivity from 0 (pure reflective foil) to 0.9 (full surface coverage by moisture condensation on the foil or no foil installed in the wall system). In order to accurately predict the thermal performance of a wall with furred airspace assembly and subjected to different climate conditions, it is highly recommended to account for moisture transport. This will be the subject of future work.

Figure 11 Effect of foil emissivity on the effective R-value of wall with OSB sheathing board

9. Summary and Conclusions

The present model was used to conduct 3D numerical simulation to determine the effective thermal resistance of a foundation wall system having a furred airspace assembly and incorporating low emissivity foil material. This model accounts for surface-to-surface radiation between the surfaces of the furring, gypsum board and foil. In order to account for the conductive, convective and radiative heat transfer in the furred airspace assembly, the energy equations of porous media and airspace, the surface-to-surface radiation equation in the furred airspace assembly, and the mass and momentum balance equations of the air in the airspace and porous media were solved simultaneously at steady-state condition. Results showed that the predicted R-value of wall specimen with fibreboard sheathing ($4.19 m^2K/W$) was in good agreement with the measured one ($4.24 m^2K/W$). After benchmarking the present model, it was used to predict the R-value of the same wall specimen but having OSB sheathing in lieu of fibreboard sheathing. Finally, the present model was used to quantify the contribution on the wall R-value by having a low foil emissivity. The results showed that a wall system with foil emissivity of 0.04 can increase the wall R-value by $0.34 m^2K/W$ ($\sim 2 ft^2 hr ^\circ F/BTU$) compared to the case of no foil installed in the wall system. Currently, the present model is being used to investigate the hygrothermal performance (the moisture transport is taken into consideration) of

different wall systems. These systems are subjected to different climate conditions of North America. The results of this effort will be reported at a later date.

Nomenclatures

C_p	Specific heat (J/(kg K))
$C_{p_{eff}}$	effective specific heat of porous media, (J/(kg K)), $C_{p_{eff}} = C_{p_o} + C_{p_f}(\rho_f / \rho_o)$
$\vec{g}$	vector of gravitational acceleration (m ² /s)
G_m	mutual radiation heat flux (W/m ²)
I	3x3 unit matrix
$\vec{n}$	normal vector at the target (point h)
$\vec{n}'$	normal vector at the sources (point d)
P	pressure (Pa)
q_{irr}	irradiation heat flux (W/m ²)
q_{radi}	radiosity (W/m ²)
q_{RAD}	radiative heat flux (W/m ²)
$\vec{r}$	vector points from the origin to the target (point h)
$\vec{r}'$	vector points from the origin to the source (point d)
$\vec{r} - \vec{r}'$	vector points from the source (point d) to the target (point h)
$\vec{v}$	velocity vector (m/s)
T	temperature (K)
t	time (s)

Greek Symbols

α	material absorptivity
ε_p	porosity (ratio of pore volume to the total volume)
ε	surface emissivity
ρ	density (kg/m ³)
ρ_r	surface reflectivity
σ	Stefan-Boltzmann constant, 5.6704E-8 W/(M ² K ⁴)
μ	dynamic viscosity (Pa·s)
κ	permeability (m ²)
λ	thermal conductivity (W/(m·K))
λ_{eff}	effective thermal conductivity of porous media (W/(m·K)), $\lambda_{eff} = \varepsilon\lambda_f + (1 - \varepsilon)\lambda_o$
χ	parameter equal to 0 or 1.0

Subscripts

a	air
f	fluid
o	solid matrix
s	surface

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