Hot-pressing process modeling for medium density fiberboard (MDF)

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Abstract

In this paper we present a numerical solution for the mathematical modeling of the hot-pressing process applied to medium density fiberboard. The model is based in the work of Humphrey [1982], Humphrey and Bolton [1989] and Carvalho and Costa [1998], with some modifications and extensions in order to take into account mainly the convective effects on the phase change term and also a conservative numerical treatment of the resulting system of partial differential equations.

Contents

Classification

65M60 Finite elements, Rayleigh-Ritz and Galerkin methods, finite methods

82C70 Transport processes

76S05 Flows in porous media; filtration; seepage

76T30 Three or more component flows

1   Hot-pressing mathematical model

Hot pressing is the process in which a mattress composed of wood fibers and resin is cured by applying heat and pressure in a press (see figure 1). Continuum and batch presses do exist, and one of the main issues in reducing the cost of the final product is to reduce the press cycle time. In order to improve the heat transfer between the press platens and the inner layers, some amount of water is added to the mat. Another issue is to adjust the parameters and the temperature history of the cycle in order to obtain a given density profile in the board. Normally, it is desirable to have lower densities at the center of the board in order to increase the mechanical rigidity for a given total mass per unit area. Predicting the influence of these parameters, namely water content, press cycle duration and history (pressure and temperature) is one of the main concerns of numerical models.

Many numerical models have been reported to help in predicting the influence of the process parameters in the final product. Among the most complete, we can find the finite difference 2D (axisymmetrical) model presented by Humphrey [1982] and, more recently the 3D model of Carvalho and Costa [1998]. Both consider conduction, phase change of water from the adsorbed to the vapor state and convection. The stress development and the determination of the density profile are not included in these models.

In this paper we present a numerical model which includes all these features and makes some correction to the energy balance equation, as

presented in Carvalho and Costa [1998]. The model is based on the finite element model, so that it allows for a more versatile definition of geometry, dimensionality and, eventually, coupling with other packages. It also will allow the use of adaptive refinement, which may be an important issue at the lateral borders, where the hot steam flows from the board to the ambient. However, this issue is not considered in this paper.
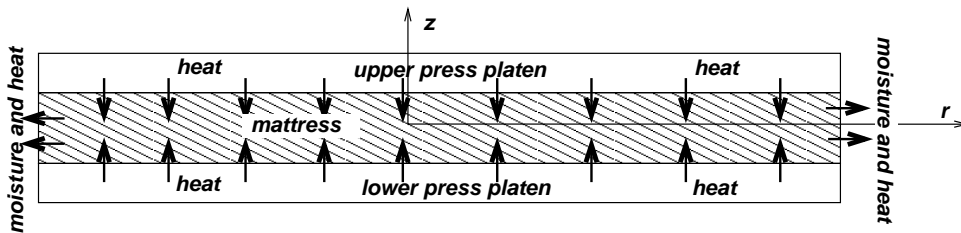

Figure 1: Hot pressing process description

1.1 Multiphase model

In order to avoid modeling the material down to the scale of the microstructure (the fibers in this case), non homogeneous materials are solved via *"averaged equations"* so that the intricate microstructure results in a continuum with averaged properties. The averaged equations and properties can be deduced in a rigorous way through the theory of mixtures and averaging operators [Whitaker, 1980].

1.2 Energy balance

We will not enter in the details of all the derivations but only for the averaged energy balance equation, which can be found in Appendix B. The referred equation is

$$\rho_s C_p \frac{\partial T}{\partial t} = \nabla \cdot (k \nabla T - \rho_v \mathbf{V}_g (C_{pv} T + \lambda + Q_l)) - \dot{m}(\lambda + Q_l), \quad (1)$$

where T is temperature, t time, C_p specific heat, k thermal conductivity, ∇ the gradient operator, ρ_s the density of the dry board (solid phase), ρ_v vapor density, $\mathbf{V}_g$ the volume averaged gas velocity, i.e.

$$\mathbf{V}_g = \epsilon \mathbf{v}_g \quad (2)$$

where $\mathbf{v}_g$ is the velocity averaged on the phase, see Appendix B), C_{pv} specific heat of vapor, $\dot{m}$ evaporation rate, λ latent heat of vaporization of free water, and Q_l adsorption heat. The main difference between this equation and that presented by both Carvalho and Costa [1998] and Humphrey [1982] is in the addition of the water evaporation heat term in the convection term instead of considering the phase change effect only on the temporal term. This term should be included because both phases, the solid material and the vapor are in relative motion

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and we think that its influence is not negligible in a high temperature process.

1.3 Steam mass balance

Carvalho and Costa [1998] proposed the following steam mass conservation equation

$$\dot{m} = \frac{\mathrm{MM_w}}{R}\,\epsilon\nabla\cdot\left[-D_v\nabla(\frac{P_v}{T}) + \frac{1}{\epsilon}\mathbf{V}_g\frac{P_v}{T}\right],\qquad(3)$$

where $\mathrm{MM_w}$ is the molecular weight of water, R the gas constant, ϵ board porosity, D_v diffusivity of water vapor in the air/vapor mixture and P_v vapor partial pressure. Considering that the steam is treated as an ideal gas, then

$$\frac{P_v}{T} = \mathrm{MM_w}^{-1}R\rho_v,\qquad(4)$$

so it may be written, assuming ϵ constant, as

$$\dot{m} = \nabla\cdot\left[-\epsilon D_v\nabla\rho_v + \mathbf{V}_g\rho_v\right].\qquad(5)$$

This expression is preferable to (3) because it is written in a conservative form that is more agreeable for a numerical treatment. The left hand side term represents the mass interfacial transport and those in the right hand side take into account the mass diffusion and the mass convection. However, it should be noted that this last expression does not have a temporal term as every consistent balance equation does. For example, if evaporation is not considered, then (5) is valid only for a steady situation, which is not in general the case. Then, we rewrite the steam mass balance as

$$\epsilon\frac{\partial\rho_v}{\partial t} = \nabla\left[\epsilon D_v\nabla\rho_v - \mathbf{V}_g\rho_v\right] + \dot{m}.\qquad(6)$$

This is another difference between our model and that proposed by Carvalho and Costa [1998].

1.4 Gas mixture mass balance

Finally, because the gas phase is composed of two main constituents, steam and air, we may use an additional equation for the mass transport of the whole gas phase. Carvalho and Costa [1998] considered

$$\frac{\partial P}{\partial t} = -\frac{1}{\epsilon}\nabla\cdot\left(-\frac{\mathbf{K}_g}{\mu}\frac{P}{T}\nabla P\right)T + \frac{\dot{m}}{\epsilon\mathrm{MM_a}}TR + \frac{P}{T}\frac{\partial T}{\partial t},\qquad(7)$$

where P is the pressure of the gas phase and K_g the board permeability tensor. Again, assuming ideal gas law as the state equation for this phase,

$$\frac{\partial}{\partial t}\left(\frac{P}{T}\right) = -\frac{R}{\epsilon\mathrm{MM_a}}\nabla\cdot\left(-\frac{K_g}{\mu}\rho_g\nabla P\right) + \frac{\dot{m}R}{\epsilon\mathrm{MM_a}},\qquad(8)$$

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we arrive to

$$\epsilon \frac{\partial \rho_g}{\partial t} = -\nabla \cdot (\rho_g \mathbf{V}_g) + \dot{m}. \tag{9}$$

In order to close the system of equations we need to introduce a relationship between $\dot{m}$ and $(\partial P_v / \partial t)$. Consider the steam mass balance (6) and the relation

$$\rho_s \mathrm{H} = \epsilon \rho_v + \rho_L, \tag{10}$$

that represents the fact that the board moisture content H is composed of vapor and bound water ρ_L. If we assume that no liquid phase is considered, then bound water may be transferred to the gas phase only (solid to steam). So

$$\dot{m} = -\frac{\partial \rho_L}{\partial t}, \tag{11}$$

and then

$$\begin{aligned} \rho_s \frac{\partial \mathrm{H}}{\partial t} &= \epsilon \frac{\partial \rho_v}{\partial t} + \frac{\partial \rho_L}{\partial t} \\ &= \nabla \cdot [\epsilon D_v \nabla \rho_v - \mathbf{V}_g \rho_v] . \end{aligned} \tag{12}$$

The air mass balance equation can be obtained by substracting (6) from (9)

$$\epsilon \frac{\partial \rho_a}{\partial t} = \nabla \cdot [-\epsilon D_v \nabla \rho_v - \mathbf{V}_g \rho_a] . \tag{13}$$

Due to the fact that the mean macroscopic diffusive fluxes should be null

$$D_v \nabla \rho_v + D_a \nabla \rho_a = 0, \tag{14}$$

the air mass balance equation is transformed in the following expression

$$\epsilon \frac{\partial \rho_a}{\partial t} = \nabla \cdot [\epsilon D_a \nabla \rho_a - \mathbf{V}_g \rho_a] , \tag{15}$$

which is very similar to (6) but here valid for the air. Obviously, the air transport equation has no evaporation term.

2 Summary of equations and boundary conditions

In order to clarify the mathematical model that is finally used for the simulation of hot-pressing process we present the following brief summary of partial differential equations.

- Energy balance equation

$$\rho_s C_p \frac{\partial T}{\partial t} = \nabla \cdot (k \nabla T - \rho_v \mathbf{V}_g (C_{pv} T + \lambda + Q_l)) - \dot{m}(\lambda + Q_l). \tag{16}$$

- Water content balance equation

$$\rho_s \frac{\partial \mathrm{H}}{\partial t} = \nabla \cdot \left[\epsilon D_v \nabla \rho_v - \mathbf{V}_g \rho_v \right]. \tag{17}$$

- Air mass balance equation

$$\epsilon \frac{\partial \rho_a}{\partial t} = \nabla \cdot \left[\epsilon D_a \nabla \rho_a - \mathbf{V}_g \rho_a \right]. \tag{18}$$

The boundary conditions are the following:

- At the press platen:

$T = T_{\mathrm{platen}}(t)$, air/water mixture in equilibrium with platen temperature

$\mathbf{V}_g \cdot \hat{\mathbf{n}} = 0$, no mass flow across the platen

$\dfrac{\partial \rho_a}{\partial z} = 0$, no air diffusion across the platen

$\dfrac{\partial \rho_v}{\partial z} = 0$, no vapor diffusion across the platen.

$$\tag{19}$$

- At the center line ($r = 0$), axial symmetry for all variables

$$\frac{\partial T}{\partial r} = 0, \quad \frac{\partial H}{\partial r} = 0, \quad \frac{\partial \rho_a}{\partial r} = 0. \tag{20}$$

- At the mid plane ($z = 0$), symmetry for all variables

$$\frac{\partial T}{\partial z} = 0, \quad \frac{\partial H}{\partial z} = 0, \quad \frac{\partial \rho_a}{\partial z} = 0. \tag{21}$$

- At the exit boundary ($r = R_{\mathrm{ext}}$),

$\dfrac{\partial T}{\partial r} = 0$, null diffusive heat flux

$P_v = P_{v,\mathrm{atm}}$, equil. with external air/water mixture,

$P_a = P_{a,\mathrm{atm}}$, equil. with external air/water mixture.

$$\tag{22}$$

3 Numerical method

The above system of equations contains three main unknowns, the temperature, the moisture content and the air density representing the dependent variables of the problem also called the state variable. In this work we have used as independent variables the time and two spatial coordinates (3D problems may be computed much in the same way). Due to the physical and geometrical inherent complexity of this problem this may be computed only by numerical methods. For the spatial discretization we have employed finite elements with multilinear elements for all the unknowns. Due to the high convective effects the numerical scheme was stabilized with the SUPG (for *"Streamline Up-wind - Petrov Galerkin"*) formulation (see Brooks and Hughes [1982]),

otherwise spurious oscillations arise. Once the spatial discretization is performed the partial differential system of equations is transformed into an ordinary differential system of equations like

$$\dot{U} = R(U), \tag{23}$$

where U is the state vector containing the three unknowns in each node of the whole mesh. So, the system dimension is $3N$ where N is the number of nodes in the mesh. The numerical procedure is as follow: Knowing the state vector at the current time (t^n), i.e. $U_j(t^n) = [T_j, H_j, \rho_{a,j}](t^n)$ where j represent an specific node in the mesh. To get the residual, right hand side of (23) the following steps should be done:

- Obtain air pressure from the gas state equation $((T, \rho_a) \to P_a)$.
- Obtain relative humidity from sorption isotherms $((T, H) \to HR)$.
- Compute saturated vapor pressure from Kirchoff expression (38) $(T-> P_{sat})$.
- Compute vapor pressure $P_v = HR P_{sat}$.
- Obtain vapor density in air from vapor state equation $(T, P_v) \to \rho_v$.
- Compute coefficients from additional constitutive laws $(P, H, T) \to (D, K_g, k_{x,y}, C_p)$.
- Compute gradients of T, P from nodal values at the Gauss points using the finite element interpolation.
- Assemble the element residual contributions in a global vector.

Once the whole residual vector at $t = t^n$ is computed the unknowns variables at the next time step is updated with

$$U^{n+1} = U^n + \Delta t R(U^n). \tag{24}$$

This kind of scheme, called *explicit integration in time* is very simple to be implemented but it has two major drawbacks, one is the limitation of the time step to ensure numerical stability and the other is the bad convergence rate for ill-conditioned system of equations. In this application the last disadvantage is very restrictive because the characteristic times of each equation are very different. To circumvent this drawback we have implemented an implicit numerical scheme

$$U^{n+1} - \Delta t R(U^{n+1}) = U^n, \tag{25}$$

where the residue is computed at the new state variable $U(t^{n+1})$ instead of using the current value $U(t^n)$. The non-linearities and the time dependency of the state vector makes this implementation more difficult and time consuming but more stable. This non-linear equation in U^{n+1} is solved by the Newton method, which requires the computation of the Jacobian

$$J = \frac{\partial R}{\partial U}. \tag{26}$$

In order to avoid the explicit computation of the Jacobian, we compute an approximate one by finite differences.

$$J \approx J_{num} = \frac{R(U + \delta U) - R(U)}{\delta U}. \qquad (27)$$

This can be done element-wise, so the cost involved is proportional to the number of elements in the mesh. In our application and despite of the ill-condition of the problem we have found a good convergence at each time step, in average 4 iterations per time step to reduce 10 orders of magnitude in the global residue.

4 Physical and transport properties

4.1 Thermal conductivity

Following Humphrey [1982] the influence of board density, moisture content and temperature on the thermal conductivity is considered as an independent correction factor obtained experimentally

$$\kappa_z = F_{\kappa,H} \, F_{\kappa,T} \, (1.172 \times 10^{-2} + 1.319 \times 10^{-4} \, \rho_s),$$
$$F_{\kappa,H} = 1 + 9.77 \times 10^{-3} \, (H - 12), \qquad (28)$$
$$F_{\kappa,T} = (T - 20) \times 1.077 \times 10^{3} + 1,$$

where κ_z is the thermal conductivity in the pressing direction in W/m$^\circ$K and ρ_s is the oven dry density of the material in Kg/m^3. The moisture correction factor $F_{\kappa,H}$ (Kollmann and Malmquist [1956] and Humphrey [1982]) assumes H, the moisture content of the board material, in %, and the temperature correction factor $F_{\kappa,T}$ (Kuhlmann [1962] and Humphrey [1982]) assumes T in $^\circ$C.

4.1.1 Heat flux direction correction

By far the greater part of conductive heat translation takes place in the vertical plane. However the energy lost from the mattress is largely the result of radial vapor migration from the center toward the atmosphere. The associated horizontal relative humidity gradient lead to a horizontal temperature gradient. Even though this gradient is always lower than the vertical one its influence should be taken into account if multidimensional analysis is required. Ward and Skaar [1963] made experimental measurements and they observed that at a first glance a factor of approximately 1.5 may be a good initial guess before doing some extra measurements. Then

$$\kappa_{xy} = 1.5\kappa_z. \qquad (29)$$

4.2 Permeability

The evaporation and condensation of water changes the vapor density and consequently its partial pressure in the voids within the composite.

A vertical pressure gradient leads to the flow of water vapor from the press platens toward the central plane of the board. At the same time an horizontal vapor flow is set up in response to the pressure gradient established in the same direction. The relation between the pressure gradient and the flow features may be assigned to the material permeability. Permeability is a measure of the ease with which a fluid may flow through a porous medium under the influence of a given pressure gradient. Different mechanisms may be involved in this flow, a viscous laminar flow, a turbulent flow and a slip or Knudsen flow. In this study only the first type is included with the assumption that Darcy's law is obeyed. This may be written as

$$\mathbf{V}_g = -\frac{\mathbf{K}_g}{\mu_g}\nabla p \tag{30}$$

where ∇p is the driving force and $\mathbf{v}_g$ is the flow variable. $\mathbf{K}_g/\mu_g$ is called the superficial permeability with μ_g the dynamic viscosity of the gas phase and $\mathbf{K}_g$ the specific permeability.

The kinetic theory of gases suggests that at normal pressure the viscosity is independent of pressure and it varies as the square root of the absolute temperature,

$$\mu \propto \sqrt{T} \qquad 10^3 < p < 10^6. \tag{31}$$

Corrections with the absolute temperature are often considered by Sutherland law

$$\mu = \frac{B \; T^{3/2}}{T + C}, \tag{32}$$

with B and C characteristic constants of the gas or vapor with μ in $Kg \, m^{-1} \, s^{-1}$. Values for B and C are available from Keenan and Keyes [1966]. For this application the following expression was adopted,

$$\mu = 1.112 \times 10^{-5} \times \frac{(T + 273.15)^{1.5}}{(T + 3211.0)}, \tag{33}$$

with T in $^{\circ}$C.

4.2.1 Variation of vertical permeability with board material density

Even though we consider that the press is closed and the density profile is set up and fixed it is included here some conclusions from Humphrey [1982] with results obtained by Denisov et al. [1975] for 19 mm boards and others from Sokunbi [1978]. The data to be fitted are the following

4.2.2 Horizontal permeability

Sokunbi measures included in figure 2.7 of Humphrey [1982] shows the relation between the board thickness in mm with horizontal permeability. For approximately 15mm board thickness and $\rho_s = 586 Kg/m^3$ the horizontal permeability is 59 times the vertical value, in agreement with the values assumed by Carvalho and Costa [1998].

Mean density $[\mathrm{Kg/m^3}]$	Mean vertical permeability $[\mathrm{m^2 \times 10^{15}}]$
425	64
475	40
525	24
575	16
625	11
675	7
725	5
775	3
825	2
875	2

Table 1: Table I: Vertical permeability density correction data

4.3 Steam in air diffusivity

The interdiffusion coefficient of steam in air can be calculated from the following semi-empirical equation [Stanish et al., 1986],

$$D_a = 2.20 \times 10^{-5} \left(\frac{101325}{P} \right) \left(\frac{T}{273.15} \right) \tag{34}$$

where the diffusivity is in $\mathrm{m^2/sec}$, pressure in $\mathrm{N/m^2}$ and T in $^\circ$K.

4.4 Vapor Density

For the pressure range likely to occur during hot pressing (between 10^3 and $3 \times 10^5 \mathrm{N/m^2}$) a linear relationship between saturated vapor pressure and vapor density may be assumed. Fitting experimental data Humphrey [1982] proposed the following expression

$$\rho_v = P_{sat}\, 6.0 \times 10^{-8}\, HR, \tag{35}$$

with ρ_v in $\mathrm{Kg/m^3}$, P_{sat} in $\mathrm{N/m^2}$ and the relative humidity HR in %. This can be deduced from the relative humidity definition

$$HR = P_v/P_{sat}, \tag{36}$$

and applying ideal gas law for gaseous phase

$$\frac{P_v}{\rho_v} = \bar{R}/\mathrm{MM_w}T. \tag{37}$$

Taking $\bar{R} = 8314 \mathrm{J/Kmol/K}$ and $\mathrm{MM_w} = 18 \mathrm{Kg/Kmol}$ with $T \approx 360^\circ \mathrm{K}$ we obtain (35).

4.5 Saturated vapor pressure

Following the Kirchoff expression with data presented by Keenan and Keyes [1966] we include here the following equation,

$$\log_{10} P_{sat} = 10.745 - (2141.0/(T + 273.15)), \tag{38}$$

with P_{sat} in N/m^2 and T in $^\circ$C.

4.6 Latent heat of evaporation and heat of wetting

Using Clausius-Clapeyron equation in differential form and after some simplification the latent heat of vaporization of free water may be written as

$$\lambda = 2.511 \times 10^6 - 2.48 \times 10^3 \, T, \tag{39}$$

with λ in J/Kg and T in $^\circ$C.

For the differential heat of sorption we follow Humphrey [1982] that used (see Bramhall [1979])

$$Q_l = 1.176 \times 10^6 \, e^{-0.15H}, \tag{40}$$

with Q_l in J/Kg and H in %.

4.7 Specific heat of mattress material

It is computed by adding the specific heat of dry wood and that of water according to the material porosity and assuming full saturation. The specific heat of dry mattress material is taken as 1357J/Kg/K and the specific heat of water has been taken to be 4190J/Kg/K. From Siau [1984] the expression for specific heat of moist wood is

$$C_p = 4180 \frac{0.268 + 0.0011(T - 273.15) + H}{1 + H}, \tag{41}$$

T in $^\circ$K and H in %.

4.8 Porosity

According to Humphrey [1982] the volume of voids within the region may be computed with

$$\epsilon = \frac{V_{\text{voids}}}{V} = (1 - \frac{\rho}{\rho_s}), \tag{42}$$

where ϵ is the porosity, ρ is the density of the region and ρ_s is the dry density of the board material.

In Carvalho and Costa [1998] they included the expression from Suzuki and Kato [1989]

$$\epsilon = 1 - \rho_s \frac{1/\rho_f + y_r/\rho_r}{1 + y_r}, \tag{43}$$

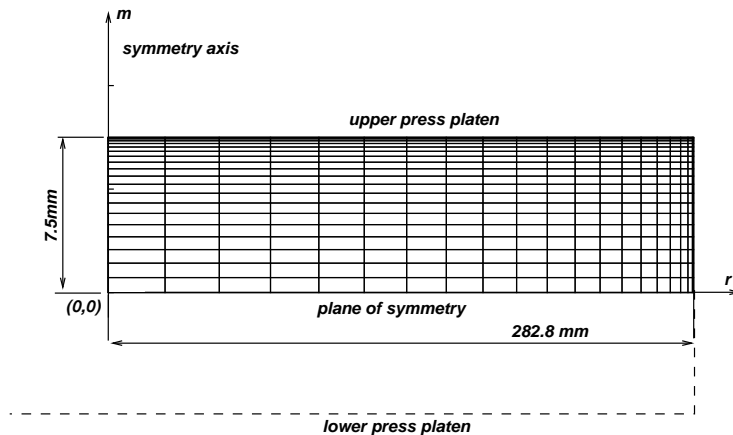Figure 2: Finite element mesh

where ρ_r is the cure resin density, ρ_f is the oven dry fiber density and y_r is the resin content (resin weight/board weight). In Carvalho and Costa [1998], the authors used $y_r = 8.5\%$ with $\rho_f = 900\text{Kg/m}^3$ and $\rho_r = 1100\text{Kg/m}^3$.

5 Numerical results

In this section we present some results that can be compared with those reported in Humphrey [1982]. This numerical experiment allows the validation of the mathematical model and its numerical implementation for future applications to hot-pressing process simulation and control. This experiment consists of a round fiberboard of 15 mm of thickness and 0.2828 m of radius that according to its axisymmetrical geometry needs as spatial coordinates only the radius r and the axial coordinate z assuming no variation in the circumferential coordinate. The axisymmetrical domain is discretized in 20×20 elements in each direction with a grading toward the press platen and the external radius as may be visualized in figure 2. In order to follow the same assumptions as in that work we fixed the air density to a very low value ($\rho_a \approx 10^{-6}$), as if the press was close with no air inside the fiberboard. The boundary conditions are as in section 2.

For the press platen temperature we have applied a ramp from 30°C at $t = 0$ to 160°C at $t = 72$ seconds with a least square fitting from data in Humphrey [1982]. As initial conditions we have assumed a uniform temperature of $T(t = 0) = 30^{\circ}$C in the whole solid material with a uniform moisture content of $H = 11\%$. The external atmosphere was considered to be at $T_{\text{atm}} = 30^{\circ}$C, $\text{HR}_{\text{atm}} = 65\%$, in such a way that the internal moisture content is, at the initial state, in equilibrium with the external atmosphere.

In the next section we include the results obtained by using the model above cited to the original problem presented in Humphrey

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[1982]. Next we present some other experiments showing some phenomena that deserve more attention for futue studies.

5.1 Original numerical experiment

Figure 3 shows the temperature distribution in time and with the axial coordinate at $r = 0$ (centerline). We can note the penetration in the axial direction of the temperature profile in time for $t = 1, 10, 50, 100, 200, 300$ and 400 seconds. Figure 4 shows the same kind of plot for moisture content. For r not to close to the external radius, the problem is almost one dimensional in the z direction. As the thermal front penetrates into the board water evaporates. This vapor advances to lower pressure regions near the symmetry plane and, as it encounters lower temperatures, it condenses releasing heat. This process can be clearly seen from the wave in moisture content (figure 4) exceeding the initial water content of 11% and results in an improvement in the heat transfer with respect to the pure conduction case. Also this phenomenon is responsible of the change in curvature of thee temperature curves, mainly at $t = 10, 50$, and 100 sec (see figure 3). The total water content in the board at a given instant can be found by integrating the bound water content and the water in vapor phase. However, this last is negligible. We can see in figure 4 that the depression in water content near the board (for instance at $t = 400$sec), is larger that the water enrichment in near the center plane. This is due to water migration from the center of the board to the external radius, where it flows to the external atmosphere. The following figures show similar plots but at different locations,

- Figure 5: temperature at external radius
- Figure 6: moisture content at external radius
- Figure 7: temperature at axial centerline ($z = 0$)
- Figure 8: moisture content at axial centerline ($z = 0$)
- Figure 9: moisture content at press platen

Figure 10 shows several isotherms at $t = 200$ seconds distributed in the r, z plane.

Figure 11 shows a three-dimensional view for moisture content represented by the third coordinate axis at $t = 200$ seconds as a function of r, z.

These results are in good agreement with those presented by Humphrey [1982]. However the cited author did not present his results at some locations that in our opinion should be treated with some care, for example at the external radius.

5.2 Further numerical experiments

In Humphrey [1982], results for moisture and temperature in the vertical and radial directions at both central planes, $r = 0$ and $z = 0$ respectively are included. No mention about the vertical distribution at $r = R = 0.2828$m or about the moisture content at press platen.
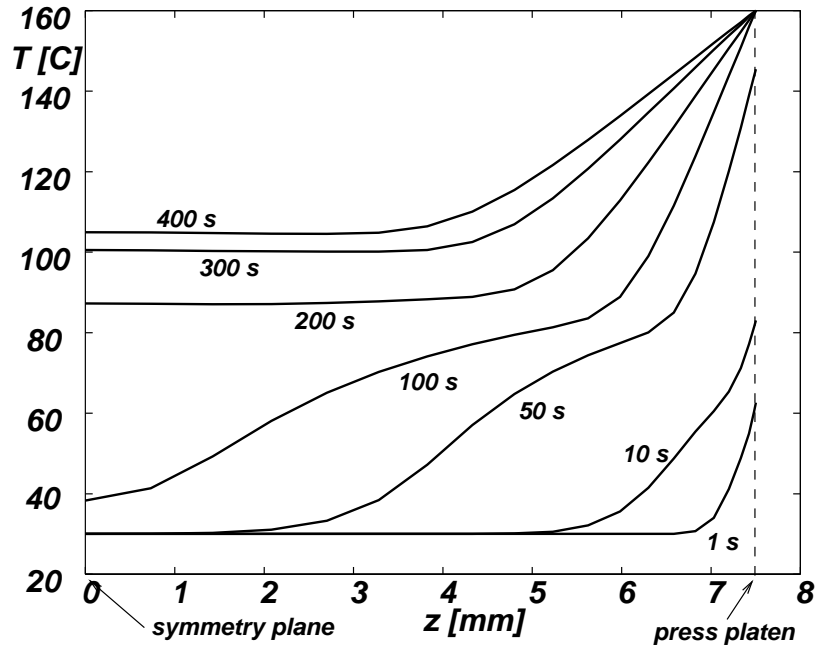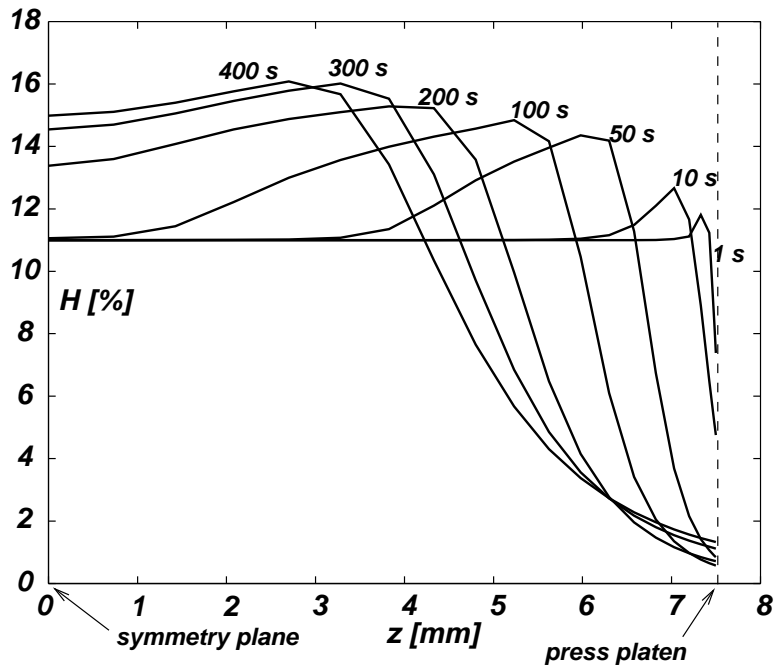
Figure 3: Axial temperature profile at $r = 0$
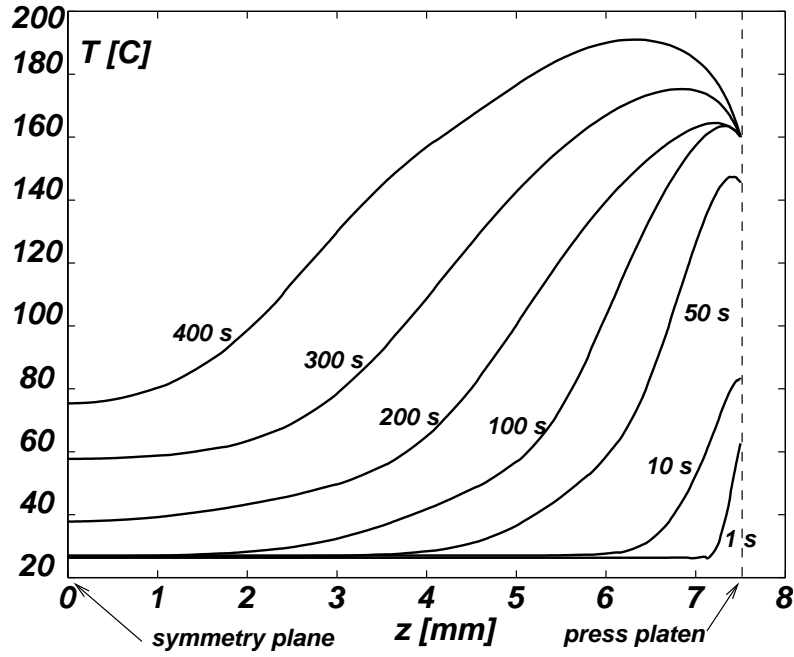

Figure 4: Axial moisture content profile at $r = 0$

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Figure 5: Axial temperature profile at $r = R = 0.2828$m
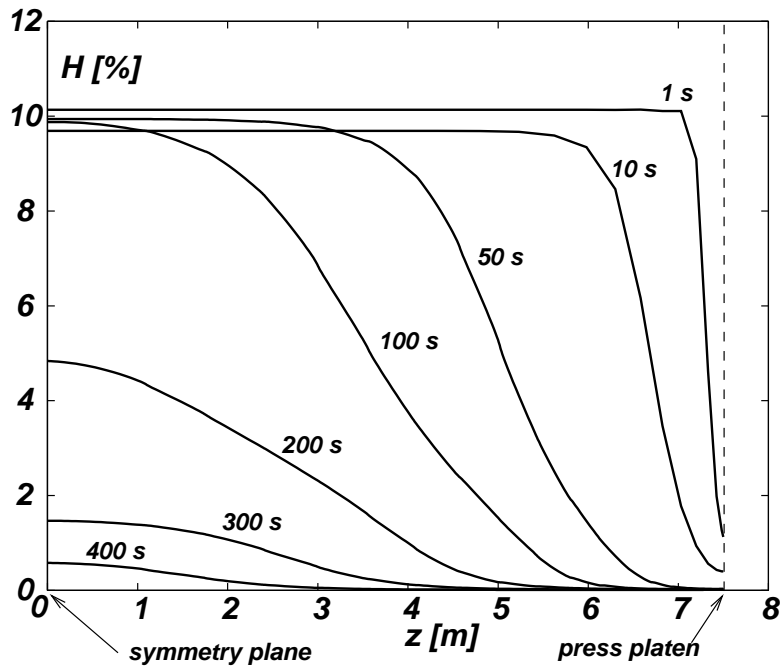

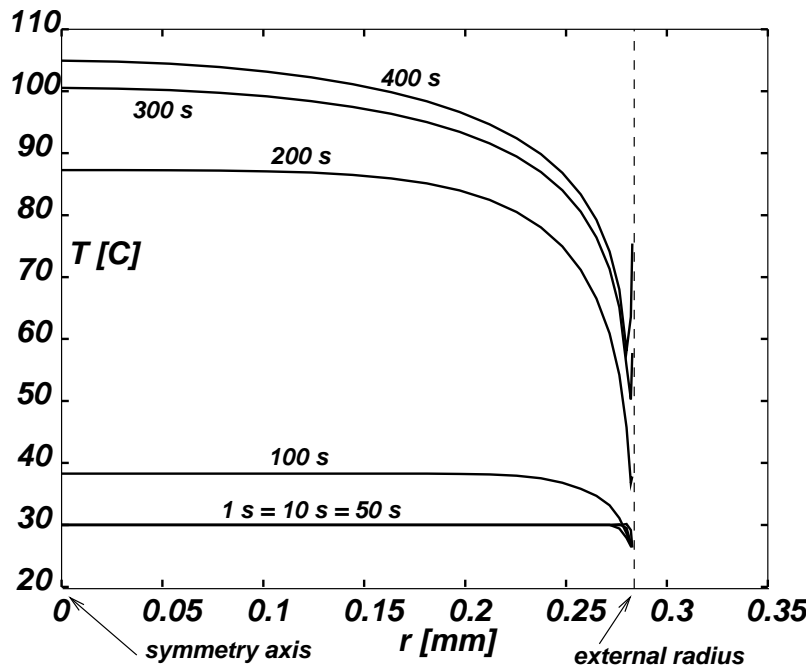Figure 6: Axial moisture content profile at $r = R = 0.2828$m
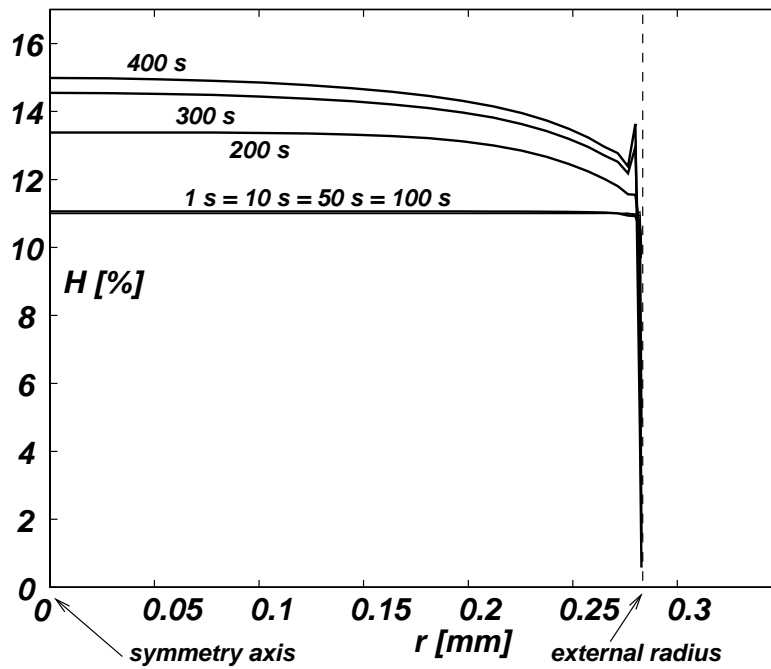
Figure 7: Radial temperature profile at $z = 0$

Figure 8: Radial moisture content profile at $z = 0$

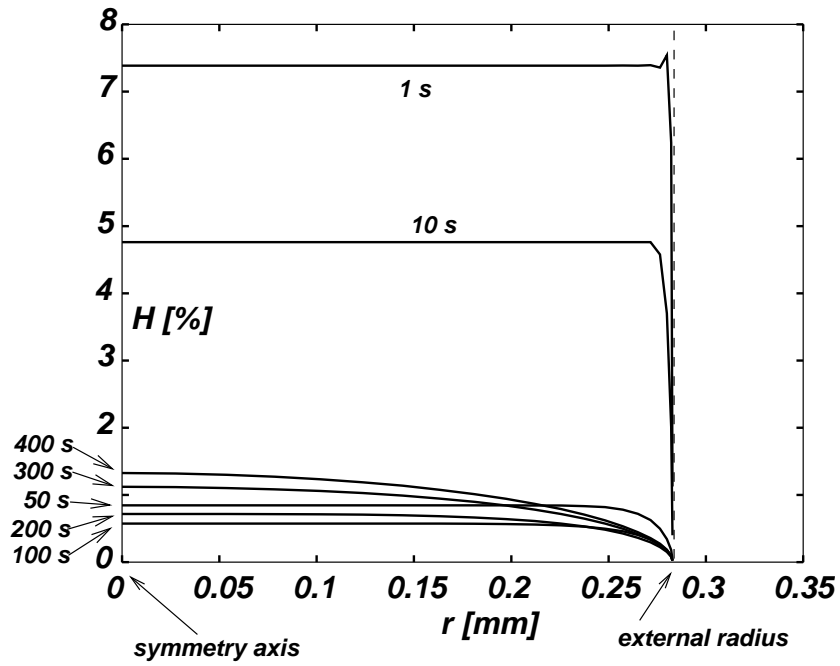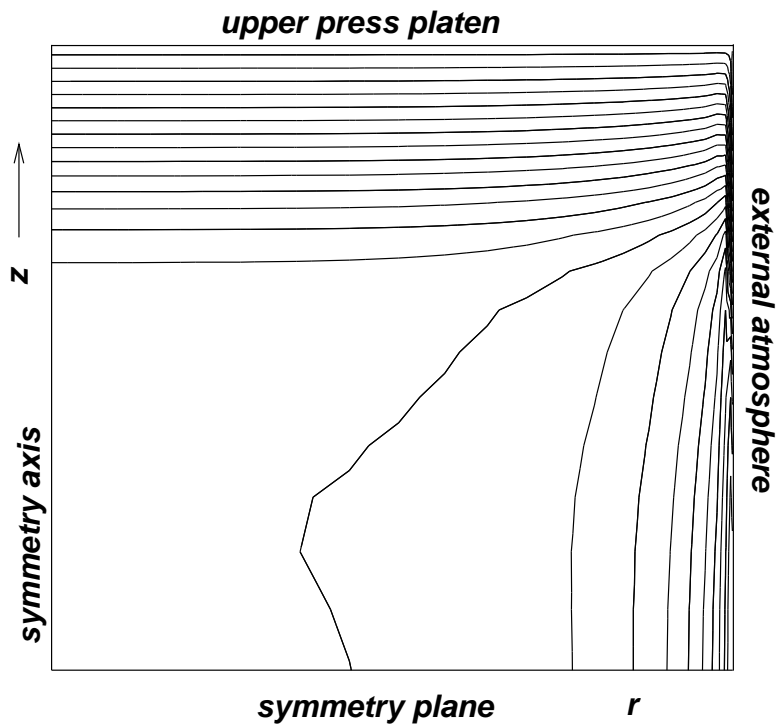Figure 9: Radial moisture content profile at press platen

Figure 10: Isotherms at t=200 seconds (z and r axis not to scale.)
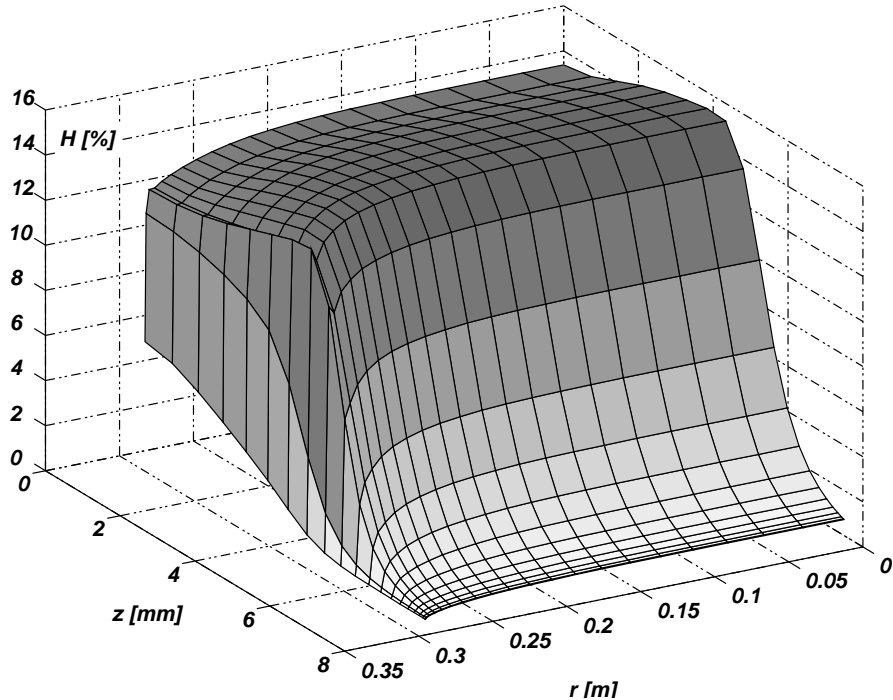
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Figure 11: 3D view for moisture content at t=200 seconds

Moreover, he had used a uniform mesh of 10×10 elements without showing what happen at the last annuli of elements corresponding to the external radius.

Our results present some overshooting in the temperature profile very close to the radial exit contour and at the first moment we though about a spurious numerical problem, but it is due to large variations of the magnitude of vapor pressure and density at the boundary. In typical runs, vapor pressure varies from near 2 atm in the center of the board to 0.01 atm at the external radius. We think that this problem will be fixed if we solve for the air density also, but then a very fine grid will be necessary at the exit boundary, since large variations of the vapor molar fraction are expected. (see figure 12). Molar fraction varies from nearly 1 at the interior of the mat to a 2% at the external atmosphere. This variation is produced in a thin layer of width δ proportional to the diffusivity of vapor in air which is very small.

6   Conclusion

We presented a numerical model for the heat and mass transfer in the hot-pressing model of a MDF fiberboard. The model includes convective effects on the phase change term and also a conservative numerical treatment of the resulting system of partial differential equations. Convective effects are responsible of an increase in heat transfer from the
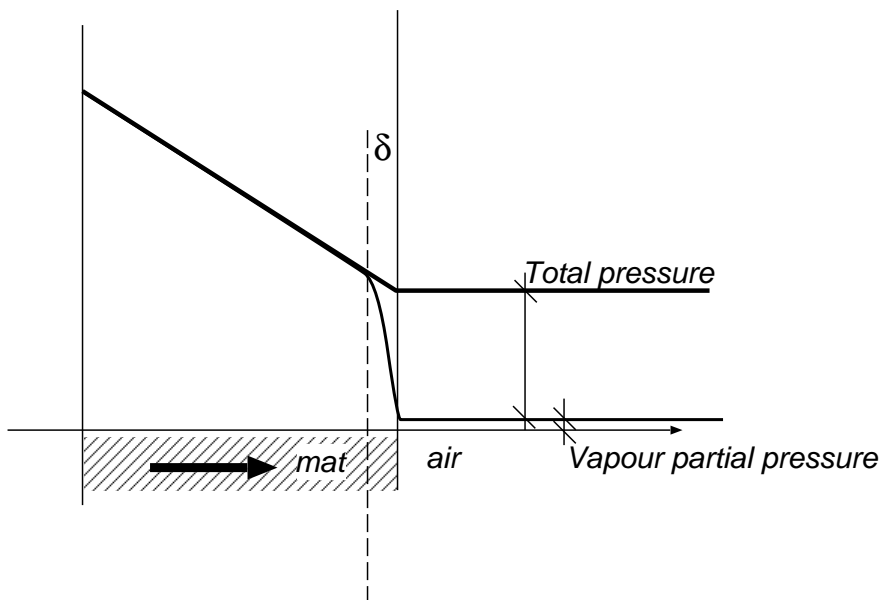
Figure 12: Boundary layer in vapor partial pressure at external radius

platen to the center of the board due to water vapor evaporation and condensation. Two-dimensional simulations allow to estimate border effects.

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A   List of symbols

A.1   Physical constants and quantities

D: diffusivity [m^2/s]

Δt: time step [s]

ϵ: material porosity (dimensionless)

F_k: correction factor for thermal conductivity (dimensionless)

Fo: Fourier number (dimensionless)

h: enthalpy

H: water content in % weight

HR: relative humidity [%]

J: Jacobian matrix

k: thermal conductivity [W/m$^\circ$K]

$\mathbf{K}_g$: specific permeability [m^2]

λ: latent heat of vaporization of free water [J/Kg]

$\dot{m}$: evaporation rate [Kg/m^{3}s]

MM: air molecular weight [Kg/Kmol]

μ: dynamic viscosity [Kg/ms]

N: number of nodes in the finite element mesh

P: pressure [N/m^2]

Q_l: adsorption heat [J/Kg]

R: gas constant for air (8314KJ/Kmol$^\circ$K

R: vector of residuals for the discrete model

r: radial coordinate [mm]

ρ: density [Kg/m^3]

T: temperature [$^\circ$K]

U: state vector for the discrete model

V: volume [m^3]

$\mathbf{v}$: gas mixture velocity [m/s]

V_{voids}: volume of voids [m^3]

y_r: resin content [weight %]

z: coordinate normal to the plate [mm]

A.2   Indices

eff: effective quantities (averaged for the gas/solid mixture)

f: fiber

g: gas phase, (air/water mixture)

j: nodal index

L, l: bound water

platen: quantity evaluated at the press platen

r: resin

ref: reference state

s: solid phase

sat: saturated atmosphere

v: water vapor

w: water

A.3 Mathematical symbols

∇: gradient (nabla) operator

$\dot{(\ )}$: temporal derivative

B Derivation of the averaged energy balance equation

The microscopic energy balance equation in the gas phase is

$$\frac{\partial}{\partial t}(\rho h) + \nabla \cdot (\rho \mathbf{v} h) = -\nabla \cdot (k \nabla T). \tag{44}$$

where h is enthalpy. For the other phases (solid and bound water) a similar expression holds, but neglecting the advective term. Applying the volume average operator [Whitaker, 1980], we arrive to the following equation averaged on the gas phase

$$\frac{\partial}{\partial t}(\epsilon_g \langle \rho h \rangle^g) + \nabla \cdot (\epsilon_g \langle \rho h \mathbf{v} \rangle^g) = \nabla \cdot (\epsilon_g \langle k \nabla T \rangle^g) + Q_g, \tag{45}$$

ϵ_g is the volumetric fraction of phase g (i.e. gas) and $\langle X \rangle^g$ is the average of quantity X on the volume occupied by phase g

$$\langle X \rangle^g = \frac{1}{\Omega_g} \int_{\Omega_g} X \, \mathrm{d}\Omega. \tag{46}$$

The term Q_g is the total enthalpy flux through the solid-gas interface Γ

$$Q_g = \int_\Gamma (\rho h)_g (\mathbf{v} - \mathbf{w}) \cdot \hat{\mathbf{n}} \, \mathrm{d}\Gamma, \tag{47}$$

where $(X)_g$ is the value of property X on the g side of the interface and $\mathbf{w}$ is the velocity of the interface. Assuming that h_g is constant on all Ω_g (for a certain volume control) then

$$\begin{aligned} Q_g &= \langle h \rangle^g \int_\Gamma (\rho)_g (\mathbf{v} - \mathbf{w}) \cdot \hat{\mathbf{n}} \, \mathrm{d}\Gamma, \\ &= \langle h \rangle^g \, \dot{m}, \end{aligned} \tag{48}$$

where $\dot{m}$ is the rate of mass of water being evaporated. A common drawback of averaged equations is that, when products of variables like ρh appear in the microscopic equation, the average of the product $\langle \rho h \rangle^g$ is obtained in the averaged equation. Now, it is not true that

$$\langle \rho h \rangle^g = \langle \rho \rangle^g \, \langle h \rangle^g, \tag{49}$$

so that the averaged equation contains more unknowns than the original equation. A common assumption is that no correlation exists between variables and so (49) is approximmately valid. This can be

justified, for instance, if the variations of each quantity around the mean is small.

Then, applying the volume average operator over the gas, solid, and bound water phases and assuming no correlations between variables we obtain the following averaged equations for the three phases

$$\frac{\partial}{\partial t}(\epsilon_g \rho_g h_g) + \nabla \cdot (\epsilon_g \rho_g \mathbf{v}_g h_g) = \nabla \cdot (\epsilon_g \langle k\nabla T \rangle^g) - \dot{m} h_g \quad \text{gas phase,}$$

$$\frac{\partial}{\partial t}(\epsilon_s \rho_s h_s) = \nabla \cdot (\epsilon_s \langle k\nabla T \rangle^s) \quad \text{solid phase and ,}$$

$$\frac{\partial}{\partial t}(\epsilon_l \rho_l h_l) = \dot{m} \, h_l \quad \text{liquid phase.}$$

(50)

Also, the average operator is dropped from here on, and a subindex g or s implies averaging on that phase. Also, $\mathbf{V}_g$ the volume averaged gas velocity is

$$\mathbf{V}_g = \frac{1}{\Omega} \int_{\Omega_g} \mathbf{v} \, d\Omega = \epsilon_g \mathbf{v}_g. \tag{51}$$

Note that in the body of the text $\mathbf{V}_g$ is used instead of vg. Now, summation of these three equations gives

$$\frac{\partial}{\partial t}(\epsilon_g \rho_g h_g + \epsilon_s \rho_s h_s + \epsilon_l \rho_l h_l) + \nabla \cdot (\epsilon_g \mathbf{v}_g \rho_g h_g) =$$
$$\nabla(\epsilon_g \langle k\nabla T \rangle^g + \epsilon_s \langle k\nabla T \rangle^s) + \dot{m} \, (h_l - h_g). \tag{52}$$

Now,

$$\epsilon_g \langle k\nabla T \rangle^g + \epsilon_s \langle k\nabla T \rangle^s = k_{\text{eff}} \nabla \langle T \rangle, \tag{53}$$

where k_{eff} is the average conductivity of the solid+water+gas mixture. The gas is assumed as an ideal mixture, so that the enthalpy is the sum of the enthalpy of its constituents, and neglect the contribution of the air constituent so that

$$\rho_g h_g = \rho_a h_a + \rho_v h_v \approx \rho_v h_v. \tag{54}$$

Taking a reference state for the entalphy at a point on the adsorbed state

$$h_v = C_{pv}(T - T_{\text{ref}}) + \lambda + Q_l. \tag{55}$$

We also neglect the entalphy of the gas phase with respect to the solid+water phases and put

$$\epsilon_s \rho_s h_s + \epsilon_l \rho_l h_l = (\rho C_p)_{\text{eff}} (T - T_{\text{ref}}), \tag{56}$$

where ρ_{eff}, and $C_{p\,\text{eff}}$ are averaged properties for the moist board, as a function of temperature and moisture content. Finally, the averaged equation is

$$\frac{\partial}{\partial t}[(\rho C_p)_{\text{eff}} (T - T_{\text{ref}})] + \nabla \cdot [\epsilon_g \rho_v \mathbf{v}_g (C_{pv}(T - T_{\text{ref}}) + \lambda + Q_l)] =$$
$$\nabla \cdot (k_{\text{eff}} T) - \dot{m}(\lambda + Q_l) \tag{57}$$

This is equivalent to (1) through relation (2) and assuming $T_{\text{ref}} = 0^\circ$C.

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