GENERALIZED EFFECTIVENESS-NTU APPROACH FOR SENSIBLE AND LATENT HEAT REGENERATORS

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Abstract. *This paper presents a generalized approach for sensible and latent heat exchangers, commonly called enthalpy exchangers, or enthalpy wheels. The methodology is based on the well established effectiveness-NTU (number of transfer units) approach, widely used for thermal analysis and design of both recuperative and regenerative heat exchangers. Different from traditional sensible heat exchangers, enthalpy exchangers employ hygroscopic materials, thereby introducing mass transfer and adsorption into the energy transfer mechanism. As a result, the developed methodology generalizes the ε-NTU approach, extending the traditional method for including additional effects related to mass transfer, phase-change and adsorption. As with the traditional ε-NTU method, dimensionless groups are introduced and the exchanger performance is calculated as a function of these groups. A multidimensional formulation for heat and mass transfer in regenerative enthalpy exchangers is normalized using the generalized approach and then numerically solved. The simulation results illustrate the employment of the ε-NTU methodology for sensible and latent heat regenerators, showing how the involved dimensionless groups can affect heat and mass transfer performance.*

Keywords: *heat exchanger, regenerator, enthalpy wheel, desiccant wheel*

1. INTRODUCTION

Enthalpy wheels are, in essence, heat regenerators having an additional energy transport mechanism due to coupled mass transfer effects. This enhancement enables these devices to provide, in addition to *sensible* heat transfer, *latent* heat exchange. Because of this characteristic, enthalpy wheels are employed in a variety of industrial applications where both forms of heat exchange is necessary. The latent heat transfer is achieved by using hygroscopic materials in the regenerator's matrix, introducing mass transfer and adsorption into the problem, thereby considerably increasing the complexity of the analysis. For the sake of better understanding and designing enthalpy exchangers, a number of formulations and computational simulations have been performed. Over the last decades, a number of studies were presented (Holmberg, 1979; Banks, 1985; Van den Bulck, Mitchell et al., 1985; Klein, Klein et al., 1990; Stiesch, Klein et al., 1995; Simonson and Besant, 1999; Sphaier and Worek, 2004; Gao, Mei et al., 2005; Nóbrega and Brum, 2009b; Chung, Lee et al., 2009; Ruivo, Costa et al., 2009; Nóbrega and Brum, 2009a; Chung and Lee, 2009; Demir, Mobedi et al., 2009), and during this period, a number of improvements in modeling heat and mass transfer in enthalpy exchangers were achieved. A common feature seen in previous studies is that a dimensional framework is widely used, which can be quite limiting since the number of variables involved to describe the problem is extensive. In order to better analyze the problem of heat and mass transfer in these type of regenerative exchangers, a generalized effectiveness-NTU (number of transfer units) aproach is proposed. This methodology, which was gradually developed in recent works (Sphaier and Worek, 2004, 2006a,b, 2008, 2009, 2010), consists of an extension of the well established ε-NTU (Shah and Sekulic, 2002; Kays and London, 1998) for sensible heat exchangers, leading to a set of dimensionless groups for de-

termining energy exchange performance. After describing the generalized methodology, it is applied to a multidimensional formulation for simulating heat and mass transfer in regenerative exchangers. The results show how the extended ε-NTU approach can be useful for assessing the behavior of enthalpy exchangers, illustrating the influence that the dimensionless groups can have on exchanger effectiveness.

2. PROBLEM FORMULATION

The general problem considered in this study is that of a an exchanger, which periodically alternates between two different process streams. The streams flow through the numerous exchanger's mini-channels transferring mass and energy to the channel's walls, which are composed of a porous sorbent material. The overall process is adiabatic such that during an entire cycle the energy removed from a process stream is entirely delivered to the other one, and vice-versa. Although the solution of each process is transient, after a number of cycles a quasi-steady state is reached and these individual transient solutions are repeated for consecutive cycles. The governing equations for this class of problem, as derived in Sphaier and Worek (2004), are given by:

$$\epsilon\,\rho_a\,\frac{\partial Y_f}{\partial t} \;+\; (1-\epsilon)\,\rho_{fs}\,f_s\,\frac{\partial W}{\partial t} \;=\; \rho_a\,\nabla\cdot(\mathcal{D}_g\nabla Y_f) \;+\; \rho_{fs}\,f_s\,\nabla\cdot(\mathcal{D}_s\nabla W)\,, \tag{1}$$

$$\rho_f\,c_f\,\frac{\partial T_f}{\partial t} \;=\; \nabla\cdot(k_f\nabla T_f) \;+\; \rho_{fs}\,f_s\,\left((1-\epsilon)\frac{\partial W}{\partial t} \;-\; \nabla\cdot(\mathcal{D}_s\nabla W)\right) i_{sor}, \tag{2}$$

$$\frac{DY}{Dt} \;=\; -\,\frac{4\,h_{\mathsf{m}}}{D_H}\,(Y \;-\; Y_f|_{p.s.}), \tag{3}$$

$$\rho\,c_p\,\frac{DT}{Dt} \;=\; -\,\frac{4\,h_{\mathsf{h}}}{D_H}\,(T \;-\; T_f|_{p.s.}). \tag{4}$$

2.1 Dimensionless groups

Following the normalization scheme described in (Sphaier and Worek, 2009) the dimensionless groups are now presented. Fourier and Fick numbers are defined based on the period of one process (τ):

$$\mathrm{Fo} \;=\; \frac{\alpha_f^\star\,\tau}{\Delta R_f^2}, \qquad \mathrm{Fi}_g \;=\; \frac{\mathcal{D}_g^\star\,\tau}{\Delta R_f^2}, \qquad \mathrm{Fi}_s \;=\; \frac{\mathcal{D}_s^\star\,\tau}{\Delta R_f^2}, \tag{5}$$

where two Fick numbers are employed due to gas-phase and surface diffusion in the porous medium. The ratios between Fick and Fourier numbers naturally lead to the definition of Lewis numbers:

$$\mathrm{Le}_g \;=\; \frac{\mathrm{Fo}}{\mathrm{Fi}_g} \;=\; \frac{\alpha_f^\star}{\mathcal{D}_g^\star}, \qquad \mathrm{Le}_s \;=\; \frac{\mathrm{Fo}}{\mathrm{Fi}_s} \;=\; \frac{\alpha_f^\star}{\mathcal{D}_s^\star}. \tag{6}$$

The presence of diffusion in the porous felt also requires definitions of Biot numbers:

$$\mathrm{Bi}_h \;=\; \frac{h_{\mathsf{h}}\Delta R_f}{k^\star}, \qquad \mathrm{Bi}_g \;=\; \frac{h_{\mathsf{m}}\Delta R_f}{\mathcal{D}_g^\star}, \qquad \mathrm{Bi}_s \;=\; \frac{h_{\mathsf{m}}\Delta R_f}{\mathcal{D}_s^\star}, \tag{7}$$

The numbers of transfer units, for heat and mass transfer, are defined as:

$$\mathrm{N}_{tu}^{\mathsf{h}} \;=\; \frac{(h_{\mathsf{h}}A_s)}{\mathrm{C}}, \qquad \mathrm{N}_{tu}^{\mathsf{m}} \;=\; \frac{(h_{\mathsf{m}}A_s)}{\mathrm{V}}, \tag{8}$$

where V is the volumetric flow rate (for an entire regenerator section) and C is the sensible heat capacity rate ($\mathrm{C} = \rho^\star c_p^\star \mathrm{V}$). Overall N_{tu}s are also defined:

$$\mathrm{N}_{\mathrm{tu}}^{\mathsf{m},o} = \frac{1}{\mathrm{V}_{\min}}\left[\frac{1}{(h_{\mathsf{m}}\,A_s)|_I} + \frac{1}{(h_{\mathsf{m}}\,A_s)|_{II}}\right]^{-1}, \quad \mathrm{N}_{\mathrm{tu}}^{\mathsf{h},o} = \frac{1}{\mathrm{C}_{\min}}\left[\frac{1}{(h_{\mathsf{h}}\,A_s)|_I} + \frac{1}{(h_{\mathsf{h}}\,A_s)|_{II}}\right]^{-1}. \tag{9}$$

The psychrometric ratio and the dimensionless dwell time are defined as:

$$\psi_r = \frac{\mathrm{N}_{tu}^{\mathsf{h}}}{\mathrm{N}_{tu}^{\mathsf{m}}} = \frac{\mathrm{N}_{\mathrm{tu}}^{\mathsf{h},o}}{\mathrm{N}_{\mathrm{tu}}^{\mathsf{m},o}}, \qquad \tau_{dw}^* = \frac{L}{u\,\tau}. \tag{10}$$

The fluid capacity ratios and total matrix capacity ratios are defined for heat and mass transfer:

$$\mathrm{C}^* = \frac{\mathrm{C}_{\min}}{\mathrm{C}_{\max}}, \qquad \mathrm{C}_r^* = \frac{\mathrm{C}_r}{\mathrm{C}_{\min}}, \qquad \mathrm{V}^* = \frac{\mathrm{V}_{\min}}{\mathrm{V}_{\max}}, \qquad \mathrm{V}_r^* = \frac{\mathrm{V}_r}{\mathrm{V}_{\min}}. \tag{11}$$

where the matrix capacity rates based on the total mass of the porous solid material in the regenerator and the period of an entire revolution (comprising both processes):

$$\mathrm{V}_r = \frac{(m_f^\star)|_T}{\rho_f^\star\,\tau_T}, \qquad \mathrm{C}_r = \frac{(m_f^\star\,c_f^\star)|_T}{\tau_T}. \tag{12}$$

The last set of effectiveness-NTU parameters are the convective-conductance ratios:

$$(h_{\mathsf{h}}\,A_s)^* = \frac{(h_{\mathsf{h}}\,A_s)\text{ on the }\mathrm{C}_{\min}\text{ section}}{(h_{\mathsf{h}}\,A_s)\text{ on the }\mathrm{C}_{\max}\text{ section}}, \quad (h_{\mathsf{m}}\,A_s)^* = \frac{(h_{\mathsf{m}}\,A_s)\text{ on the }\mathrm{V}_{\min}\text{ section}}{(h_{\mathsf{m}}A_s)\text{ on the }\mathrm{V}_{\max}\text{ section}}. \tag{13}$$

A last set of dimensionless parameters are also introduced due to the presence of adsorption and coupled heat and mass transfer:

$$i_{sor}^* = \frac{i_{sor}\,\rho_g^\star\,\Delta Y_{\mathrm{ref}}}{\rho_f^\star\,c_f^\star\,\Delta T_{\mathrm{ref}}}, \qquad i_{v,\Delta T}^* = \frac{i_{v,\Delta T}\,\rho_g^\star\,\Delta Y_{\mathrm{ref}}}{\rho_f^\star\,c_f^\star\,\Delta T_{\mathrm{ref}}}, \qquad \Omega = f_s\,\frac{\rho_s^\star\,W_{\max}}{\rho_g^\star\,\Delta Y_{\mathrm{ref}}}. \tag{14}$$

Finally, it is important to notice that the parameters $\mathrm{N}_{tu}^{\mathsf{m}}$, $\mathrm{N}_{tu}^{\mathsf{h}}$, Fo, Fi_g, Fi_s and τ_{dw}^* can assume different values for process I and process II, *i.e.* for each regenerator section.

2.2 Normalized formulation

Using the normalization scheme described in Sphaier and Worek (2009), these equations are transformed into the following dimensionless form:

$$(1-\epsilon)\,\Omega\,\frac{\partial W^*}{\partial t^*} + \epsilon\,\frac{\partial Y_f^*}{\partial t^*} = \mathrm{Fi}_s\,\Omega\,\nabla_*\cdot(\delta_s\nabla_* W^*) + \mathrm{Fi}_g\,\nabla_*\cdot(\delta_g\nabla_* Y_f^*), \tag{15}$$

$$\chi_f\,\frac{\partial T_f^*}{\partial t^*} = \mathrm{Fo}\,\nabla_*\cdot(\kappa_f\,\nabla_* T_f^*) + (1-\epsilon)\,\Omega\left(\frac{\partial W^*}{\partial t^*} - \mathrm{Fi}_s\,\frac{\nabla_*\cdot(\delta_s\nabla_* W^*)}{1-\epsilon}\right) i_{sor}^*, \tag{16}$$

$$\tau_{dw}^*\,\frac{\partial Y^*}{\partial t^*} + \frac{\partial Y^*}{\partial x^*} = \mathrm{N}_{tu}^{\mathsf{m}}\,(Y_f^*|_{p.s.} - Y^*), \tag{17}$$

$$\chi\left(\tau_{dw}^*\,\frac{\partial T^*}{\partial t^*} + \frac{\partial T^*}{\partial x^*}\right) = \mathrm{N}_{tu}^{\mathsf{h}}\,(T_f^*|_{p.s.} - T^*), \tag{18}$$

where T_f^* and Y_f^* are, respectively, temperature and sorbate concentrations in the gas phase of the porous material, W^* is the sorbate concentration in the adsorbed phase (also in the porous material), and T^* and Y^* are temperature and sorbate concentration in the process stream. The boundary

conditions are given by:

$$-\Omega\,\frac{\delta_s}{\mathrm{Bi}_s}\frac{\partial W^*}{\partial r^*}\;-\;\frac{\delta_g}{\mathrm{Bi}_g}\frac{\partial Y_f^*}{\partial r^*}\;=\;(Y^*\;-\;Y_f^*), \tag{19}$$

$$-\kappa_f\,\frac{\partial T_f^*}{\partial r^*}\;=\;\mathrm{Bi}_h\,(T^*\;-\;T_f^*)\;+\;\frac{\mathrm{Bi}_g}{\mathrm{Le}_g}\,(Y^*\;-\;Y_f^*)\,i_{v,\Delta T}^*\;-\;\frac{\Omega}{\mathrm{Le}_s}\,\delta_s\frac{\partial W^*}{\partial r^*}\,i_{sor}^*, \tag{20}$$

at $r^*=0$. At the remaining boundary one finds:

$$\frac{\partial Y_f^*}{\partial r^*}\;=\;\frac{\partial T_f^*}{\partial r^*}\;=\;0,\quad\text{at}\quad r^*=1, \tag{21}$$

$$\frac{\partial Y_f^*}{\partial x^*}\;=\;\frac{\partial T_f^*}{\partial x^*}\;=\;0,\quad\text{at}\quad x^*=0,\;1. \tag{22}$$

The periodicity of the problem appears in the inlet conditions:

$$Y^*(0,t^*)\;=\;Y_{in}^*,\qquad T^*(0,t^*)\;=\;T_{in}^*, \tag{23}$$

where the inlet quantities have different values in each process stream. Each cycle (one full rotation) is divided in two processes, comprehending the period under which a regenerator channel is exposed to one of the two process streams. Moreover, considering a counterflow arrangement, the change of variable $x_{next}^*=1-x_{current}^*$ is applied at the end of each process.

The dimensionless variables in the presented equations are defined by:

$$T^*\;=\;\frac{T\;-\;T_{\mathrm{ref}}}{\Delta T_{\mathrm{ref}}},\qquad T_f^*\;=\;\frac{T_f\;-\;T_{\mathrm{ref}}}{\Delta T_{\mathrm{ref}}}, \tag{24}$$

$$Y^*\;=\;\frac{Y-Y_{\mathrm{ref}}}{\Delta Y_{\mathrm{ref}}},\qquad Y_f^*\;=\;\frac{Y_f-Y_{\mathrm{ref}}}{\Delta Y_{\mathrm{ref}}},\qquad W^*\;=\;\frac{W}{W_{\max}}, \tag{25}$$

$$x^*\;=\;\frac{x}{L},\qquad r^*\;=\;\frac{r-R_p}{\Delta R_f},\qquad \nabla_*\;=\;\left(K_f\frac{\partial}{\partial x^*},\frac{\partial}{\partial r^*}\right), \tag{26}$$

$$t^*\;=\;\frac{t\;-\;N\,(\tau_I+\tau_{II})}{\tau_I},\qquad\text{for process } I, \tag{27}$$

$$t^*\;=\;\frac{t\;-\;\tau_I\;-\;N\,(\tau_I+\tau_{II})}{\tau_{II}},\qquad\text{for process } II, \tag{28}$$

where K_f is the aspect ratio $\Delta R_f/L$. The remaining coefficients account for variations in physical properties:

$$\delta_g\;=\;\frac{\mathcal{D}_g}{\mathcal{D}_g^\star},\qquad \delta_s\;=\;\frac{\mathcal{D}_s}{\mathcal{D}_s^\star},\qquad \chi\;=\;\frac{\rho\,c_p}{\rho^\star c_p^\star},\qquad \chi_f\;=\;\frac{\rho_f\,c_f}{\rho_f^\star\,c_f^\star},\qquad \kappa_f\;=\;\frac{k_f}{k^\star}, \tag{29}$$

2.3 Performance assessment

The performance of the considered heat and mass exchanger is measured in terms of mass and enthalpy effectiveness, defined as:

$$\varepsilon_{\mathsf{m}}\;=\;\frac{\mathrm{V}_I}{\mathrm{V}_{\min}}\,\frac{\bar{Y}_{out,I}^*-\bar{Y}_{in,I}^*}{\bar{Y}_{in,II}^*-\bar{Y}_{in,I}^*}\;=\;\frac{\mathrm{V}_{II}}{\mathrm{V}_{\min}}\,\frac{\bar{Y}_{out,II}^*-\bar{Y}_{in,II}^*}{\bar{Y}_{in,I}^*-\bar{Y}_{in,II}^*}, \tag{30}$$

$$\varepsilon_{\mathsf{i}}\;=\;\frac{\mathrm{C}_I}{\mathrm{C}_{\min}}\,\frac{\bar{\bar{i}}_{I,out}-\bar{\bar{i}}_{I,in}}{\bar{\bar{i}}_{II,in}-\bar{\bar{i}}_{I,in}}\;=\;\frac{\mathrm{C}_{II}}{\mathrm{C}_{\min}}\,\frac{\bar{\bar{i}}_{II,out}-\bar{\bar{i}}_{II,in}}{\bar{\bar{i}}_{I,in}-\bar{\bar{i}}_{II,in}}, \tag{31}$$

where time-averaged quantities (over the duration of each process), denoted by the bar overscripts, are employed for providing the average inlet and outlet quantities for the entire regenerator sections.

3. RESULTS AND DISCUSSION

After presenting the generalized ε-NTU analysis methodology, simulation results using the obtained normalized formulation are presented for illustrating the effects of dimensionless groups on the regenerator's energy exchange performance. Simulation results are calculated using the numerical algorithm proposed in Sphaier and Worek (2008), and results of different test-cases are presented. As similar to the traditional ϵ-NTU approach used for sensible heat exchangers, in the method herein employed, the performance of the exchanger depends on values of characteristic dimensionless groups. However, due to nonlinear effects of property variations and heat and mass transfer couplings, there can be some variation with operating conditions (temperature and concentration) as well. As a result, a specific operating condition was defined for the simulations, as displayed in table 1.

Table 1. Inlet operating conditions.

stream I inlet temperature	30°C
stream II inlet temperature	15°C
stream I inlet relative humidity	40%
stream II inlet relative humidity	40%

Once the inlet conditions have been defined, illustrative results showing the effects of varying the dimensionless parameters on wheel performance are presented. In this paper, a balanced and symmetric wheel is selected, *i.e.* the flow areas and heat capacity flow rates (as well as $\dot{m}_a$) for both process streams are equal. Under this considerations, one finds that

$$\mathrm{V}^* = \mathrm{C}^* = (h_{\mathsf{m}}\, A_s)^* = (h_{\mathsf{h}}\, A_s)^* = 1, \tag{32}$$

$$\mathrm{N}^{\mathsf{h}}_{tu}|_I = \mathrm{N}^{\mathsf{h}}_{tu}|_{II} = 2\,\mathrm{N}^{\mathsf{h},o}_{\mathrm{tu}}, \tag{33}$$

$$\mathrm{N}^{\mathsf{m}}_{tu}|_I = \mathrm{N}^{\mathsf{m}}_{tu}|_{II} = 2\,\mathrm{N}^{\mathsf{m},o}_{\mathrm{tu}}, \tag{34}$$

and the mass and energy (enthalpy) transfer effectiveness, which assess wheel performance, can be calculated from the following expressions:

$$\varepsilon_{\mathsf{m}} = \frac{\bar{Y}^*_{out,I} - \bar{Y}^*_{in,I}}{\bar{Y}^*_{in,II} - \bar{Y}^*_{in,I}}, \qquad \varepsilon_{\mathsf{i}} = \frac{\bar{\bar{i}}_{I,out} - \bar{\bar{i}}_{I,in}}{\bar{\bar{i}}_{II,in} - \bar{\bar{i}}_{I,in}} \tag{35}$$

Since the employed model is normalized within an effectiveness-NTU framework, the effects of varying material properties, construction parameters, and operation-related quantities, are analyzed in terms of different values for the dimensionless parameters. This has the advantage of facilitating the analysis since a reduced number of cases needs be examined.

The effect of sorbate uptake, is studied by modifying the isotherm shape and the effect of varying the heat of sorption is investigated by writing its expression in terms of a dimensionless parameters that measures the relative amount by which it exceeds the heat of vaporization. Simple expressions involving a separation factor r and the excess parameter e_{sor} are employed:

$$W^* = \phi/(r + (1-r)\,\phi), \tag{36}$$

$$i_{sor} = (1 + e_{sor})\, i_{vap}. \tag{37}$$

For analyzing variations in the isotherm shape, a value for the heat of sorption for a silica-gel desiccant was used for calculating the dimensionless heat of sorption parameter e_{sor}, leading to a value of

$e_{sor} = 0.08$; this value was calculated from a silica-gel desiccant, as described in (Chua, Ng et al., 2002). Then, for investigating the effects of varying the heat of sorption, e_{sor} was gradually increased.

An enthalpy wheel in operation with a relatively low storage effect in the process stream ($\tau^*_{dw} = 0.001$), low diffusional resistances in the porous layer ($\mathrm{Bi}_h = \mathrm{Bi}_g = 0.1$) is selected for the analysis. The value of Bi_s is chosen as 10^4 because, in the considered temperature range, the surface diffusivity is five orders of magnitude higher than that of the gas-phase diffusivity. For the sake of evaluating mass and energy exchange performance for different situations, for each of the analyzed configurations, the values of ε_{m} and ε_{i} were calculated for different values of N_{tu} and C^*_r, as similarly done for sensible heat exchangers (Kays and London, 1998). Besides these considerations, the Lewis approximation, which assumes that the psychrometric ratio equals unity (*i.e.* $\mathrm{N}^{\mathsf{h}}_{tu}/\mathrm{N}^{\mathsf{m}}_{tu} = 1$), is also considered. This approximation is considered in virtually all previous studies related to enthalpy exchangers. The verification of such assumption is out of the scope of this investigation, and will be conducted in a future study.

The first analysis focuses on comparing variations in enthalpy transfer effectiveness for different isotherm shapes (described buy the separation factor, r) considering $f_s = 1\%$. Figure 1 portrays the effect of using different values for the separation factor r. The variation of ε_{i} (in percentage) with the overall N_{tu}s (considering $\mathrm{N}^{\mathsf{m}}_{tu} = \mathrm{N}^{\mathsf{h}}_{tu}$) is plotted for for different values of C^*_r. Each figure compares

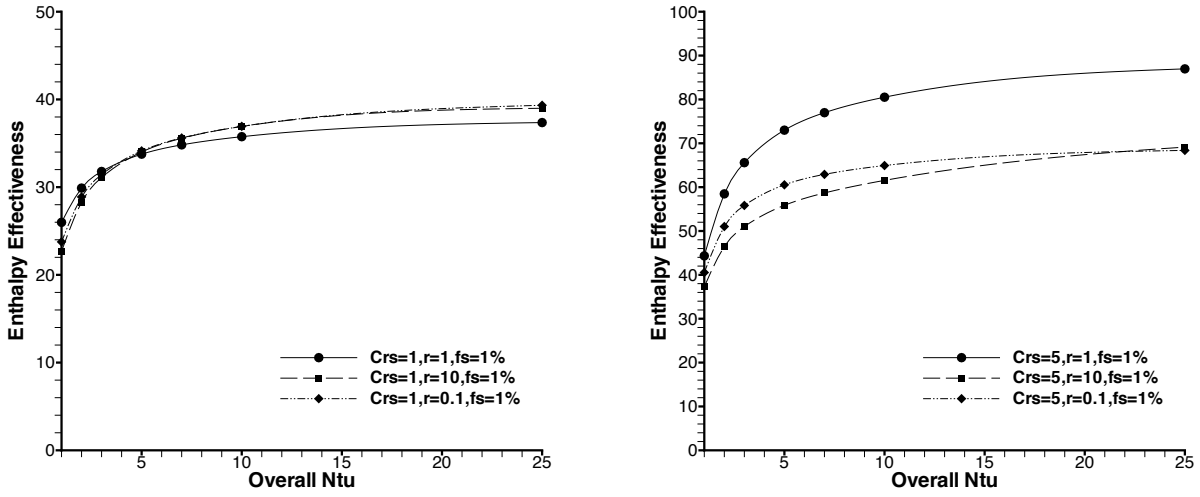

Figure 1. Variation of $\varepsilon_{\mathsf{i}}(\%)$ with overall N_{tu}s for $f_s = 1\%$ and different values of r and C^*_r.

three cases of isotherm shape: $r = 1$ (linear isotherm), $r = 1/10$ (favorable isotherm) and $r = 10$ (unfavorable isotherm). As can be seen, as C^*_r is increased the effectiveness rises. When examining the variation of ε_{i} with N_{tu} a monotonic increase is seen. It is interesting to note that, for low C^*_r the linear isotherm generally yields a slightly lower value of effectiveness; nevertheless for $\mathrm{C}^*_r = 5$, the linear isotherm produces a notably higher effectiveness.

Next, figure 2 presents results for a higher sorbent mass fraction ($f_s = 10\%$). As observed, the enthalpy transfer effectiveness values are considerably higher for $f_s = 10\%$. This result has been previously documented and is, to some extent, naturally expected. However, when the isotherm shape is varied, interesting effects are found. Comparing figures 1 and 2, for $\mathrm{C}^*_r = 1$, one can easily see that the separation factor r has a greater effect in performance for $f_s = 10\%$ than for $f_s = 1\%$. On the other hand, for $\mathrm{C}^*_r = 5$ an opposite behavior is encountered: the variation of r has a much more pronounced effect with $f_s = 1\%$ than with $f_s = 10\%$.

Another observation that should be made regarding the performance variation with isotherm shape

is that the favorable isotherm ($r = 1/10$) not necessarily yields better results. For the cases simulated in this study, the linear isotherm ($r = 1$) presented an overall better result. The only case in which the linear isotherm did not give better results was for low C_r and low f_s (figure 1). In this figure both favorable and unfavorable ($r = 10$) isotherms give higher ε_i values (for overall N_{tu} greater than 4). Looking at lower N_{tu} values one sees that, for this case, the linear isotherm is again a better choice. Speaking of low N_{tu} values, by looking at all other simulated cases one can easily see that, for lower N_{tu}s, variations in ε_i with r are generally less pronounced. A last observation that can be made with respect to the effects of isotherm shape regards the unfavorable isotherm, $r = 10$. With the exception of the low C_r^* and low f_s case, for which this value (together with $r = 1/10$) gives a slightly better performance than the linear isotherm, in all other cases there is a general tendency where the unfavorable isotherm gives the worst performance results.

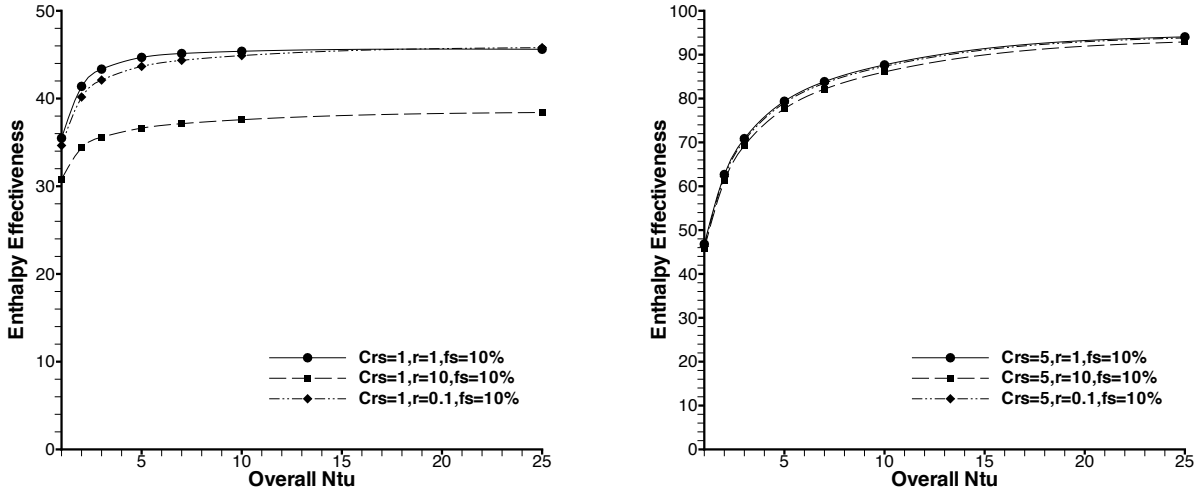

Figure 2. Variation of $\varepsilon_\mathrm{i}(\%)$ with overall N_{tu}s for $f_s = 10\%$ and different values of r and C_r^*.

Next, in order to investigate the effects of the heat of sorption in the performance of enthalpy exchangers, the parameter e_{sor} is varied from its original value $e_{sor} \approx 0.08$ to higher values. Figure 3 presents the variation of the enthalpy effectiveness with the overall number of transfer units for different C_r^* values and $e_{sor} = 0.08$, 0.3 and 0.5. As can be seen, as the heat of sorption is increased (by augmenting the value of e_{sor}) the performance is worsened. Nevertheless, this effect is less pronounced for larger C_r^* values. This suggests that for slower rotations the sorption heating will be less significant to wheel performance.

The last comparisons are focused on examining the effect of heat and mass transfer resistances within the porous material. Figure 4 presents the variation of enthalpy effectiveness with the overall number of transfer units for different values of C_r^* and different Biot numbers. The gas-phase mass transfer Biot numbers and the heat transfer Biot numbers are varied jointly such that $\mathrm{Bi}_h = \mathrm{Bi}_g$. The value of surface diffusion Biot number was maintained at the same ratio $\mathrm{Bi}_g/\mathrm{Bi}_s = 10^5$, due to the much smaller diffusivity encountered in the adsorbed phase. Looking at the presented figure one notices that as the Biot numbers are augmented the effectiveness progressively decrease. Nevertheless this decrease is almost unfelt for low C_r^* and $\mathrm{Bi} = 1$, especially for larger values of N_{tu}. Nevertheless, if the Biot numbers are increased to 10, there is a significant reduction in the effectiveness value. The reduction in ε_i is clearly greater for larger C_r^* values, such that even with $\mathrm{Bi} = 1$ and large values of N_{tu}, there is still a noticeable reduction in ε_i.

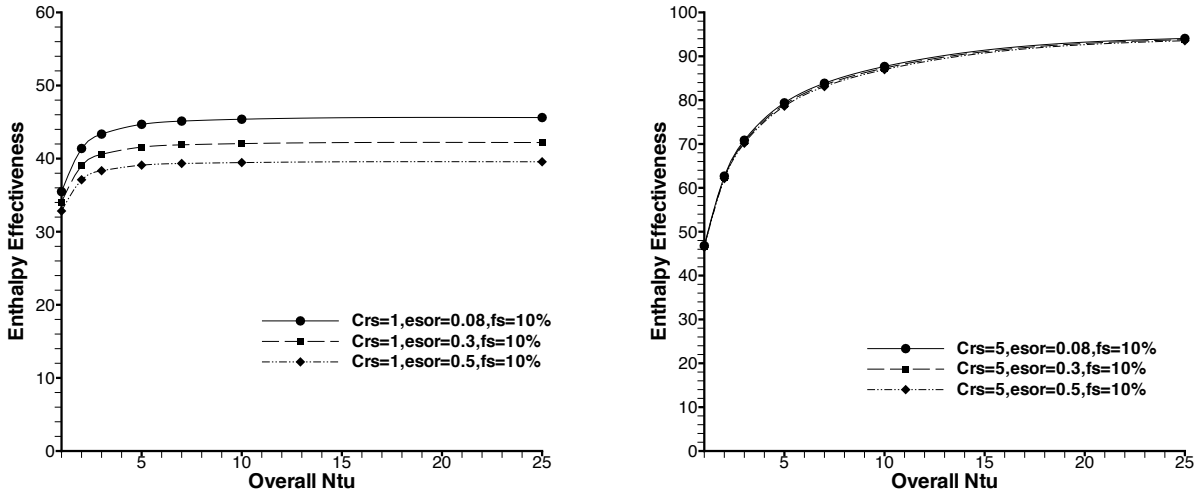

Figure 3. Variation of $\varepsilon_{\text{i}}(\%)$ with overall N_{tu}s for and $f_s = 10\%$ for different heats of sorption.

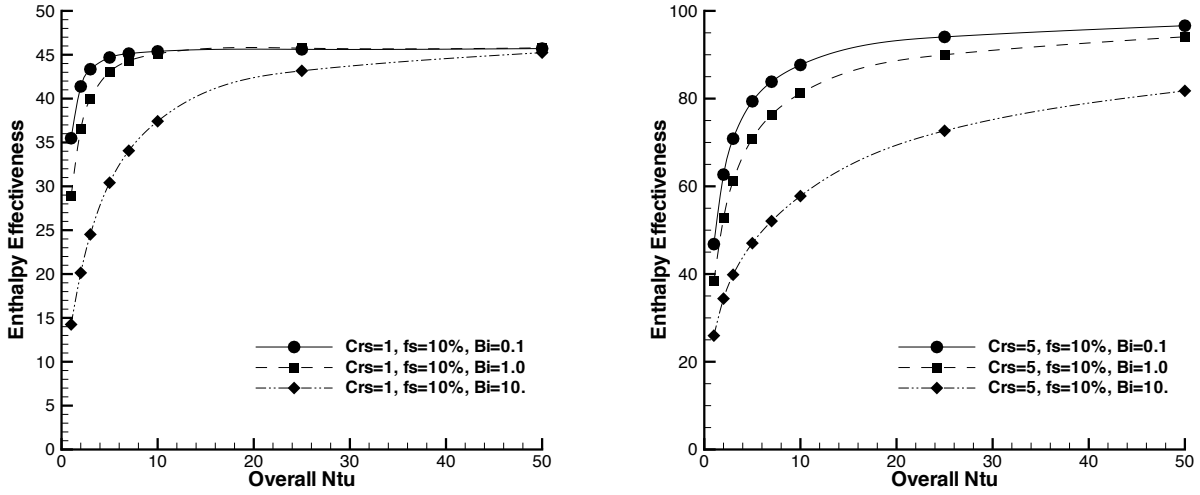

Figure 4. Variation of $\varepsilon_{\text{i}}(\%)$ with overall N_{tu}s for and $f_s = 10\%$ for different Biot numbers.

CONCLUSIONS

This paper presented a generalized effectiveness-NTU methodology for analyzing and thermal designing sensible and latent heat regenerators. The methodology was shown to be a generalization of the traditional ε-NTU developed for sensible heat exchangers. The generalized approach leads to dimensionless groups which are used for normalizing a multi-dimensional formulation for heat and mass transfer regenerators. This formulation was then numerically solved and the impact of the dimensionless parameters on the exchanger performance was analyzed by varying parameters' values. The results were analyzed by observing how different parameters affect the energy exchange performance, measured in terms of the enthalpy transfer effectiveness, ε_{i}. Two of the varied parameters were the heat capacity ratio (C_r^*), and the number of transfer units (N_{tu}) for heat and mass transfer, as traditionally done for periodic heat exchangers. In order to analyze the effect of having different

uptake relations, the isotherm separation parameter r was varied, and the effect of modifying the heat of sorption was examined by varying the parameter e_{sor}. The results showed the expected behavior seen in heat exchangers in which higher C_r^* and N_{tu} values yield better energy exchange performance results. In addition, it was seen that higher sorbent mass fractions (f_s) also give better performance results, and it was demonstrated that increasing the heat of sorption resulted in a degrading effect on energy performance.

When investigating the effect of isotherm shape it was found, in general, that the linear isotherm leads to better performance results. The only exception was the case with lower capacity ratio and lower sorbent mass fraction.It was also observed that, in general, the unfavorable isotherm produced the worst performance results. Nevertheless, these effects could be altered for different operating conditions, and hence this should be further investigated. Finally, the effect of varying heat and mass transfer resistances (within the adsorbent material) was examines by simulating the operation for different values of the heat and mass transfer Biot numbers; the results provided an indication that the energy exchange performance decreases with increasing values of heat and mass transfer resistances. This effect was magnified for larger values of C_r^* and smaller values of numbers of transfer units.

As closing comments, one should mention that, although the results herein presented are still preliminary, they already demonstrate how relevant can be selecting an adequate material. Moreover, in spite of the findings presented in this contribution, the performance will vary for different types of heat exchangers, and it can also depend on operating conditions, such that further research should be carried-out.

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