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Evaluation of energy efficiency in biofuel drying by means of energy and exergy analyses

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Abstract

The calculation of heat consumption is based on the First Law and it gives quantitative information about the energy used in drying. However, it does not pay any attention to the quality of the energy used in drying. To take into account the quality of the energy, attention must be paid to the Second Law, too. Especially in those cases where the energy used in drying may be converted to mechanical work, it is important to consider the Second Law is. In this paper, the energy efficiency of biofuel drying in a pulp and paper mill is evaluated on the basis of energy and exergy analysis. The evaluation is based on the determination of the heat consumption and the irreversibility rate for energy and exergy analysis, respectively. The evaluation methods are applied to two different drying systems, single-stage-drying with partial recycle of spent air, and multi-stage-drying. Both drying systems are also provided with a heat recovery unit in which the inlet air is pre-heated using the outlet air of the dryer. There are two alternative heat sources available for the drying energy, steam at a pressure of 3 bar and water at a temperature of 80 °C. The results show that the heat consumption is only dependent to a small extent on the heat source type or the drying system. On the other hand, the irreversibility rate depends to a considerable on the heat source and the drying system.

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Keywords: Single-stage-drying; Recycling; Multi-stage-drying; Heat consumption; Irreversibility rate

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Nomenclature

A	area (m ²)
\dot{C}	heat capacity flow (W/K)
c_p	specific heat capacity (J/kg K)
E	exergy (W)
e	specific enthalpy of material (J/kg)
h	specific enthalpy of moist air (J/kg)
I	irreversibility rate (W)
l_v	vaporisation heat of water (J/kg)
M	molar mass (kg/mol)
\dot{M}_{dm}	mass flow of dry material (kg/s)
\dot{m}	mass flow (kg/s)
n	number of drying stages (-)
p	pressure (Pa)
p'_v	pressure of saturated vapour (Pa)
R	gas constant = 8.314 J/mol K, recycle ratio (-)
\dot{S}	entropy (W/K)
s	specific entropy (J/kg K)
T, t	temperature (K, °C)
Δt_{min}	minimum temperature difference in heat recovery unit (°C)
u	material moisture (kg/kg _{dm})
v	specific volume (m ³ /kg), velocity (m/s)
x	air moisture (kg/kg _{da})

Greek symbols

Φ	heat effect (W)
Φ_e	indirectly supplied heat (W)
Φ_m	mechanical work input (W)
Φ_l	heat losses (W)
γ	temperature coefficient (1/K)
ρ	density (kg/m ³)

Subscripts

a	air
da	dry air
dm	dry material
in	inlet
m	material
o	surrounding
out	outlet

<i>r</i>	boundary of the system
ref	reference
v	vapour
w	water

1. Introduction

In general, the main goal of drying is to decrease the moisture content of solid materials to below a certain limit. Drying is part of several industrial processes for example in the food, chemical, and paper and pulp industries. The main objectives of drying are extended storage life, quality enhancement, and ease of handling and further processing [1].

In this paper, the term biofuel refers to biofuels consumed by the pulp and paper industry. The most important biofuels are bark, forest residues, sawdust, and chips. They are by-products of the main processes and are burnt in fluidised bed boilers to produce heat and electricity for the mill. Power is generated by a steam turbine-generator. The moisture content of these biofuels typically varies between 50% and 60% (water per total mass). One of the main objectives of biofuel drying is to increase the electricity production.

The heat sources with the greatest potential for drying energy in pulp and paper mills are secondary heat flows and steams at different pressures. Secondary heat is, by definition, heat transferred from primary heat sources to process flows—pulp, liquor, and washing waters—and further to circulation water, flue gas, steam, and vent gases [2]. Usually, the technically available secondary heat is hot water in the temperature range 50–90 °C. Steam pressures used in pulp and paper mills typically vary from 3–4 bar (back pressure steam) to 10–12 bar (extraction steam).

The most common way to evaluate the energy efficiency of various drying processes is the specific heat consumption (amount of heat supplied per mass of water evaporated); see e.g. [3–6]. Because biofuel drying is part of the electricity production process, the specific heat consumption does not necessarily give an exact overview of energy efficiency in drying. In addition to heat consumption, the temperature of the heat source used in drying must be included in the analysis of the drying process. This necessitates Second Law analysis for the drying process. For example, Refs. [7–9] mention the importance of the Second Law/exergy analysis associated with the drying processes, but it is not used in these papers. In Ref. [10], the exergy method is applied to determine the so-called energy saving ability function.

In this paper, the energy efficiency of the drying is evaluated using two analysis methods: the energy method and the exergy method. The energy analysis is based on the determination of the heat consumption. The exergy analysis is based on the determination of the irreversibility rate. The evaluation methods are applied to two different drying systems, single-stage-drying with partial recycle of spent drying air (SSD) and multi-stage-drying (MSD). Both drying systems are also provided with a heat recovery unit where the inlet air is pre-heated with the outlet air of the dryer.

2. Determination of heat consumption

2.1. Single-stage dryer with partial recycle of drying air

Fig. 1 shows a continuous single-stage dryer with partial recycle of spent air including heat recovery unit. As a result of the partial recycle of air and the heat recovery, the air temperature before the heater increases. This decreases the heat consumption of the drying process. Because of the recycle, the fresh drying air has less potential for taking up moisture, which leads to a bigger air mass flow through the drying chamber.

By using the notations in Fig. 1, the energy and mass balance equations over the drying system, excluding the heat recovery unit, become

$$x_1 = x_2 \quad (1)$$

$$x_3 = x_4 \quad (2)$$

$$x_5 = x_6 = x_7 \quad (3)$$

$$t_5 = t_6 = t_7 \quad (4)$$

$$t_2 = t_7 - \Delta t_{\min} \quad (5)$$

$$\Phi = \dot{m}_{da}(h_4 - h_3) + \Phi_e \quad (6)$$

$$\dot{m}_{da}[(1 - R)x_2 + Rx_6] = \dot{m}_{da}x_3 \quad (7)$$

$$\dot{m}_{da}(x_5 - x_4) = \dot{M}_{dm}(u_1 - u_2) \quad (8)$$

$$\dot{m}_{da}[(1 - R)h_2 + Rh_6] = \dot{m}_{da}h_3 \quad (9)$$

$$\dot{m}_{da}(h_4 - h_5) = \dot{M}_{dm}(e_2 - e_1) - \Phi_m - \Phi_e + \Phi_1 \quad (10)$$

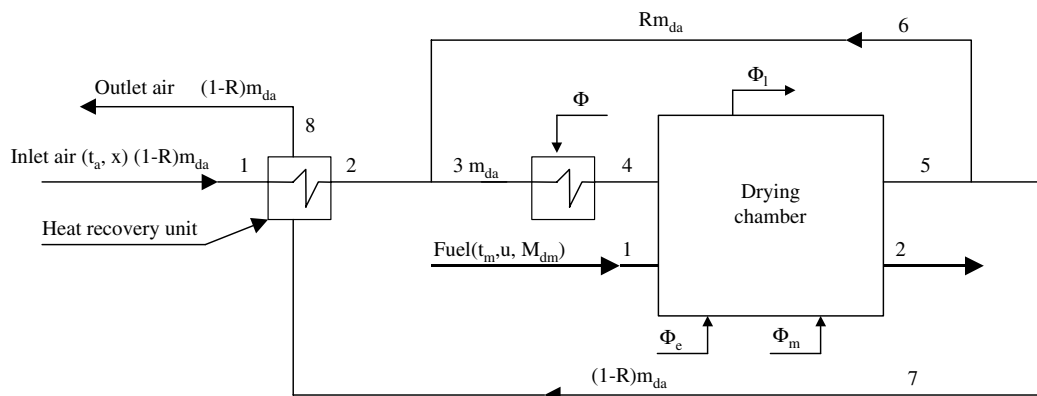


Fig. 1. Single-stage-drying with partial recycle of spent air.

The equations are based on [11]. If the moist air is treated as a perfect gas the specific enthalpy of the moist air becomes

$$h = c_{pa}t_a + x(c_{pv}t_a + 2501) \quad (11)$$

where the zero level for the enthalpy of the dry air is dry air at a temperature of 0 °C, and the zero level for the enthalpy of the vapour is water at temperature of 0 °C. The specific enthalpy of the moist material e is defined as follows:

$$e = (c_{pdm} + uc_{pw})t_m + u\Delta H_w \quad (12)$$

where ΔH_w is the mean enthalpy of sorption to material. Usually, the product $u\Delta H_w$ is much smaller than $c_{pdm}t_m$ and may be neglected [12]. The reference temperature where the specific enthalpy of the moist material is zero is 0 °C.

By eliminating some of the variables, the number of unknown variables becomes seven (x_3 , x_5 , t_{2a} , t_{3a} , t_{5a} , Φ , \dot{m}_{da}). There are only six Eqs. (5)–(10) at our disposal. To obtain the seventh equation we assume that after the drying chamber the air is fully saturated. In this case, the air moisture x_5 can be expressed as a function of saturated vapour pressure

$$x_5 = 0.622 \frac{p'_v(t_5)}{p_{da}} \quad (13)$$

where $p'_v(t_5)$ is the saturated vapour pressure as a function of outlet temperature t_{5a} . Ref. [13] gives the following approximate equation for the saturated vapour pressure

$$p'_v = 10^5 e^{\frac{11.78(t-99.64)}{(t+230)}} \quad (14)$$

where t is the temperature of the saturated air in degrees Celsius. Substituting (13) and (14) in the energy and mass balance equations, all unknown variables can be calculated. The use of (13) and (14) necessitates the dryer having so much evaporation surface that the drying air reaches its saturation point. However, this should be one of the targets in the design of energy-efficient dryers.

The outlet air moisture and temperature (x_8 , t_8) are calculated from the energy and water balance of the heat recovery unit. Because the air moisture after the drying chamber is high, part of the moisture condenses in the heat recovery unit. Generally, condensation occurs when the surface temperature of the heat exchanger is below the dew point of air. By using the notations in Fig. 2(a) the energy and water balance over the heat recovery unit become

$$(1 - R)\dot{m}_{da}(h_2 - h_1) = (1 - R)\dot{m}_{da}(h_7 - h_8) - \dot{m}_w c_{pw} t_8 \quad (15)$$

$$\dot{m}_w = (1 - R)\dot{m}_{da}(x_7 - x_8) \quad (16)$$

The air after the heat recovery unit is fully saturated and the air moisture x_8 may be expressed using (13) and (14). Substituting (13), (14) and (16) in (15) the outlet air temperature and moisture can be determined.

2.2. Multi-stage-drying

Fig. 2 shows a continuous multi-stage dryer equipped with a heat recovery unit. As a result of several drying stages and heat recovery, the air temperatures before the heaters increase, which decreases heat consumption. Adding the drying stages also decreases the air mass flow for a given evaporation rate, if all drying stages operate at the same temperature levels.

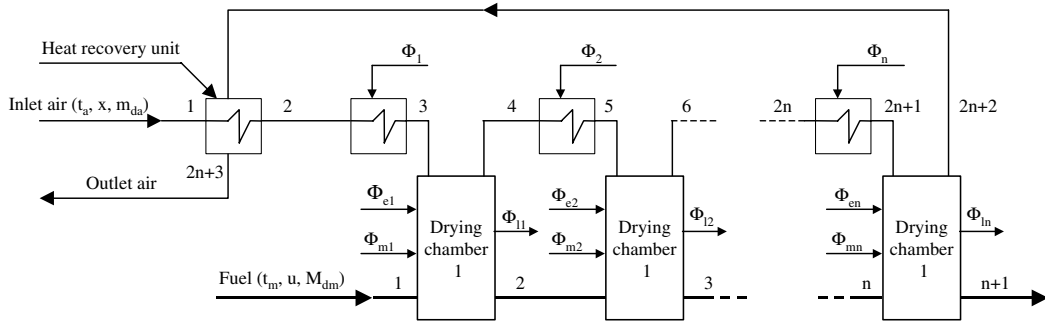


Fig. 2. Multi-stage-drying with n drying stages.

As in SSD, we assume that the drying air is fully saturated after each drying chamber. By using the notations in Fig. 2, the energy and mass balances over the entire drying system for n drying stages become

$$x_1 = x_2 = x_3 \tag{17}$$

$$x_4 = x_5 \tag{18}$$

⋮

$$x_{2n} = x_{2n+1} \tag{19}$$

$$t_2 = t_{2n+3} - \Delta t_{\min} \tag{20}$$

$$\dot{m}_{da}(x_4 - x_3) = \dot{M}_{dm}(u_1 - u_2) \tag{21}$$

$$\dot{m}_{da}(x_6 - x_5) = \dot{M}_{dm}(u_2 - u_3) \tag{22}$$

⋮

$$\dot{m}_{da}(x_{2n} - x_{2n-1}) = \dot{M}_{dm}(u_{n-1} - u_n) \tag{23}$$

$$\dot{m}_{da}(x_{2n+2} - x_3) = \dot{M}_{dm}(u_1 - u_{n+1}) \tag{24}$$

$$x_4 = 0.622 \frac{10^5 e^{\frac{11.78(t_4 - 99.64)}{(t_4 + 230)}}}{P_a} \tag{25}$$

⋮

$$x_{2n+2} = 0.622 \frac{10^5 e^{\frac{11.78(t_{2n+2} - 99.64)}{(t_{2n+2} + 230)}}}{P_a} \tag{26}$$

$$\dot{m}_{da}(h_3 - h_4) = \dot{M}_{dm}(e_2 - e_1) - \Phi_{m1} - \Phi_{e1} + \Phi_{11} \tag{27}$$

⋮

$$\dot{m}_{da}(h_{2n+1} - h_{2n+2}) = \dot{M}_{dm}(e_{n+1} - e_n) - \Phi_{mn} - \Phi_{en} + \Phi_{1n} \tag{28}$$

$$\Phi = \sum \dot{m}_{da}(h_i - h_j) + \sum \Phi_{ek}, \quad i = 3, 5, \dots, 2n + 1, \quad j = 2, 4, \dots, 2n, \quad k = 1, 2, \dots, n \tag{29}$$

where n is the number of drying stages (heating + drying period).

To determine the temperature of the material after each drying stage, there must be a function which gives the correlation between the material moisture and temperature. Drying is usually divided into three successive phases:

- A short initial heating period, when the temperature of the material rises.
- A constant drying rate period when the temperature of the material remains constant at the temperature close to the wet bulb temperature of the drying medium.
- A falling drying rate period when the temperature of the material rises.

According to the drying experiments represented in [14] and [15] the constant drying rate period is relatively short in wood drying with moist air. This means that the temperature of the material rises almost constantly as the moisture content decreases. In this paper, we assume that there is a simple linear correlation between the fuel temperature and moisture

$$t_m(u) = \frac{t_{\min} - t_{\text{ain}}}{u_{\text{in}}} u + t_{\text{ain}} \quad (30)$$

where t_{\min} is the inlet temperature of the fuel, t_{ain} the inlet temperature of the air, and u_{in} the inlet moisture of the fuel.

The outlet temperature and moisture (t_{2n+3} , x_{2n+3}) are calculated from the energy and water balance of the heat recovery unit in the same way as in SSD (see Eqs. (15) and (16)).

3. Determination of irreversibility rate

The dryer is an open steady-state system with two flows of matter (moist material and moist air) and one or several heat inputs and outputs to the system. The irreversibility rate for an open steady-state system is derived in [16], and its general form is

$$I = T_o \left[\sum m_i \Delta s_i - \sum \frac{\Phi_i}{T_{ri}} \right] \quad (31)$$

where Φ_i represents the heat inputs and outputs from the system. In this paper, the heat input to the system is assumed to be positive. T_r is the temperature at the point on the boundary of the system where the heat transfer is taking place [16]. Generally, the term Φ_i/T_i is known as a thermal entropy flux [16]. If the energy flows representing heat inputs and outputs are assumed to be inside the control region, the expression in parentheses in (31) gives the entropy generation rate for the drying system. Defining the irreversibility rate, they are outside the control region.

If there are infinite numbers of thermal entropy fluxes with constant heat capacity flow (W/K), the T_r in (31) can be expressed as follows [17]:

$$T_r = \frac{T_{r2} - T_{r1}}{\ln \frac{T_{r2}}{T_{r1}}} \quad (32)$$

where T_{r2} and T_{r1} are the lowest and highest temperatures of the heat flow, respectively.

The total differential for the change of entropy may be expressed as follows [18]:

$$ds = \frac{c_p}{T} dT - \left(\frac{\partial v}{\partial T} \right)_p dp \quad (33)$$

Eq. (33) is valid for reversible and irreversible processes [18]. Next, the entropy change is separately determined for dry air, vapour, water and, moist material using (33). The temperature and pressure of the triple point of water are taken as reference values ($T_{\text{ref}} = 273.16 \text{ K}$, $p_{\text{ref}} = 40.6221 \text{ Pa}$). If dry air and vapour are treated as a perfect gas, the result of the (33) becomes [18]

$$\Delta s = c_p \ln \frac{T_2}{T_1} - \frac{R}{M} \ln \frac{p_2}{p_1} \quad (34)$$

where c_p is the average heat capacity over the temperature range of T_1 and T_2 .

For dry air the pressure remains constant, and the change of entropy becomes

$$\Delta \dot{S}_{\text{da}} = \dot{m}_{\text{da}} c_{pa} \ln \frac{T_{\text{out}}}{T_{\text{in}}} \quad (35)$$

For vapour the entropy change is calculated as the difference between the entropy associated with the flow of vapour into and out of the dryer

$$\dot{S}_v = \dot{m}_{\text{da}} x \left[\frac{l_v}{T_{\text{ref}}} + c_{pv} \ln \frac{T}{T_{\text{ref}}} - \frac{R}{M_v} \ln \frac{p_v}{p_{\text{ref}}} \right] \quad (36)$$

$$\Delta \dot{S}_v = \dot{S}_{v\text{out}} - \dot{S}_{v\text{in}} \quad (37)$$

where (36) is applied to in and out values, respectively.

For water the entropy change is also calculated as the difference between the entropy associated with the flow of water in and out of the dryer. The entropy change for water is derived from (33) and the result is [18]

$$\dot{S}_w = \dot{m}_w \left[c_{pw} \ln \frac{T}{T_{\text{ref}}} - v(T_{\text{ref}}) \gamma(T_{\text{ref}}) (p - p_{\text{ref}}) \right] \quad (38)$$

$$\Delta \dot{S}_w = \dot{S}_{w\text{out}} - \dot{S}_{w\text{in}} \quad (39)$$

where (38) is applied to in and out values, respectively. The term γ is the temperature coefficient of water

$$\gamma = \frac{1}{v} \left(\frac{\partial v}{\partial T} \right)_p \quad (40)$$

If the pressure is not much higher than the normal air pressure, the latter terms in (38) is very small and may be neglected.

The entropy change for moist material is calculated as follows:

$$\Delta \dot{S}_m = \dot{M}_{\text{dm}} \left[c_{p\text{dm}} \ln \frac{T_{\text{outm}}}{T_{\text{inm}}} + c_{pw} \left(u_{\text{out}} \ln \frac{T_{\text{outm}}}{T_{\text{ref}}} - u_{\text{in}} \ln \frac{T_{\text{inm}}}{T_{\text{ref}}} \right) \right] \quad (41)$$

Total entropy change in the dryer is the sum of all entropy changes

$$\Delta\dot{S}_{\text{tot}} = \Delta\dot{S}_{\text{da}} + \Delta\dot{S}_{\text{v}} + \Delta\dot{S}_{\text{w}} + \Delta\dot{S}_{\text{m}} \quad (42)$$

If the steam in a turbine can expand to a pressure corresponding to the temperature of the surroundings, the exergy balance for a steam power plant with a dryer inside the control region is [16]

$$E_{\text{f}} + E_{\text{a}} - E_{\text{p}} = W + I_{\text{d}} + I_{\text{x}} \quad (43)$$

where E_{f} is the exergy of the fuel, E_{a} the exergy of the air entering the control region, E_{p} the exergy of the streams leaving the control region, W the mechanical work, I_{d} the irreversibility rate of the drying and I_{x} the irreversibility rates of the other sub-processes. It can be assumed that E_{p} and I_{x} remain almost the same, regardless of the drying process. The difference between the irreversibility rates of two drying processes indicates how much mechanical work (electricity) will be lost if the dryer with a higher irreversibility rate is used instead of the dryer with a lower rate.

4. Results and discussion

The main objective of these calculations is to theoretically compare the energy efficiency of two different drying processes (SSD and MSD) using two evaluation methods (specific heat consumption and irreversibility rate). Both drying systems are provided with heat recovery units (HRU).

Moist biofuel is dried from an initial moisture of 1.5 kg/kg_{dm} to a final moisture of 0.3 kg/kg_{dm}, and the dry mass flow of the fuel is 1 kg_{dm}/s. Two alternative heat sources are available for the heating: steam at a pressure of 3 bar (133 °C), and water at a temperature of 80 °C. The minimum temperature difference between air leaving the heat exchanger and the heat source entering the heat exchanger is 5 °C, in which case the inlet air temperatures of the drying chambers are 128 and 75 °C. Because the water temperature in the heat exchanger decreases, the temperature T_{r} in the case of hot water is calculated using (32). In the case of steam, the temperature of the heat source remains constant. The outlet temperature of water is assumed to be 10 °C higher than the inlet air temperature of the heat exchanger (t_3 for SSD and t_2, t_4, \dots, t_{2n} for MSD). In the heat recovery unit, the minimum temperature difference ($t_7 - t_2$ for SSD and $t_{2n+2} - t_2$ for MSD) is 5 °C. The outdoor temperature is 15 °C and the absolute moisture 0.0064 kg/kg_{da} corresponding to a relative humidity of 60%.

The specific heat capacities for dry air, vapour, and water are taken as mean values between the corresponding temperature limits from [19] and [20]. Ref. [21] recommends the following formula for the specific heat capacity of dry wood:

$$c_{\text{pm}} = 0.1031 + 0.003867T \text{ [kJ/kgK]}$$

Mechanical work inputs into the dryer are usually small compared to heat consumption. If the dryer is well insulated, the heat losses may be assumed to be small, too. In these calculations, the mechanical work inputs and the heat losses are neglected. There are no indirectly supplied heat inputs into the drying chamber.

Figs. 3 and 4 show the heat consumption and the irreversibility rate for inlet air temperatures of 75 and 128 °C. Fig. 5 also shows the heat consumption and the irreversibility rate for the multi-stage-drying system, which has no HRU.

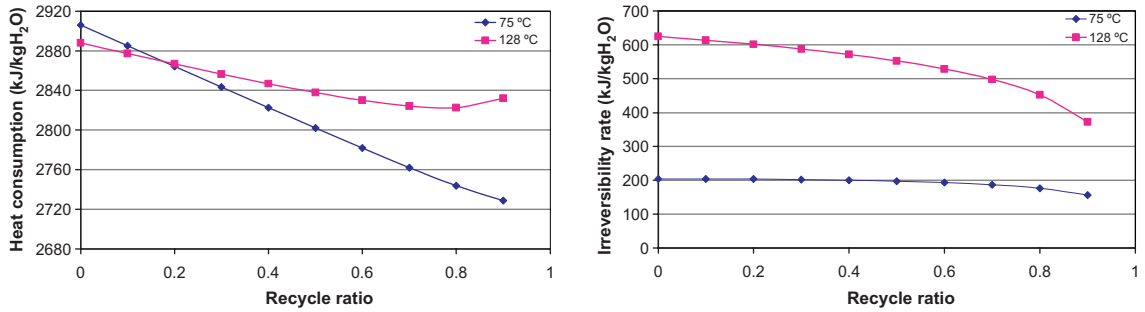


Fig. 3. Heat consumption and irreversibility rate for single-stage-drying with partial recycle of spent air.

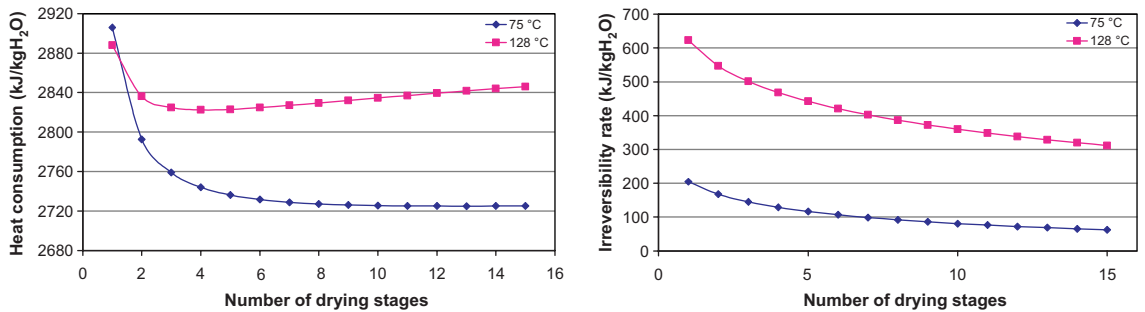


Fig. 4. Heat consumption and irreversibility rate for multi-stage-drying.

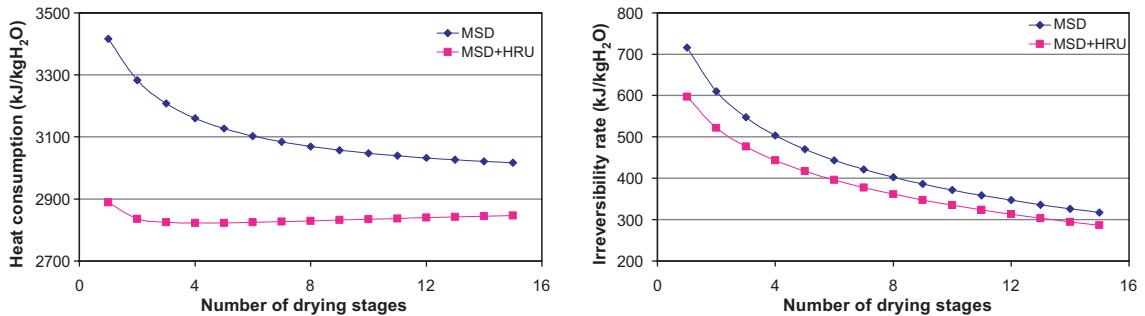


Fig. 5. Effect of heat recovery unit (HRU) on heat consumption and irreversibility rate in multi-stage-drying for inlet air temperature 128 °C.

To compare the energy and exergy efficiency between the drying systems, the dryer dimensions are altered so that they are the same. It is assumed that the dryer dimensions are proportional to the cross-sectional area of the dryer, which depends on the air mass flow as follows:

$$A = \frac{\dot{m}_a}{\rho_{da} v_a} \tag{44}$$

The term \dot{m}_a is defined as a product of drying stages and mass flow of dry air through the multi-stage dryer ($\dot{m}_a = n\dot{m}_{da}$). The mass flow \dot{m}_a is first determined for the MSD, because the number of drying stages must be an integer. In SSD, the recycle ratio is given such a value that the air mass flow through the drying chamber is the same as $n\dot{m}_{da}$. Instead of the air mass flow or the cross-sectional area, the energy consumption and the irreversibility rate are expressed as a function of dimensionless value, defined as follows:

$$\text{RDA} = \frac{A}{A_{\text{ref}}} = n \frac{\dot{m}_{da}}{\dot{m}_{a\text{ref}}} \quad (45)$$

where RDA is the ratio of dryer areas and $\dot{m}_{a\text{ref}}$ the mass flow of the dry air in single-stage-drying with no recycle ($R = 0$ and $n = 1$). This is the smallest dryer, and the value of RDA is always bigger than 1 when the number of the drying stages increases. In the example calculations, the maximum number of the drying stages is reached when the recycle ratio is approximately 0.9. The comparison results for the inlet air temperatures 75 and 128 °C are shown in Figs. 6 and 7, respectively.

Figs. 3 and 4 show that the heat consumption seems to be slightly lower for the inlet temperature of 75 °C in most cases. This mainly results from the heat recovery (see Table 1), which covers a clearly greater part of heat consumption for the inlet temperature of 75 °C. However, the difference between heat consumptions is under 100 kJ/kg_{H₂O} in most cases. This is a relatively

Table 1
Heat effects of heat recovery unit (Φ_{HRU}) in some calculation cases

R	$\Phi_{\text{HRU}}, t_{\text{dry}} 78 \text{ }^\circ\text{C}$ (kW)	$\Phi_{\text{HRU}}, t_{\text{dry}} 128 \text{ }^\circ\text{C}$ (kW)
0	583	633
0.1	604	624
0.2	620	612
0.3	632	595
0.4	638	572
0.5	637	543
0.6	626	503
0.7	601	451
0.8	555	378
0.9	471	269
n		
1	583	633
2	633	536
3	595	459
4	556	405
5	523	365
6	496	334
7	474	309
8	455	288
9	439	271
10	425	256

R Recycle ratio, n number of drying stages.

small difference, and on the basis of heat consumption the energy efficiency of the drying is not substantially dependent on the drying temperature in the example cases.

Instead, the irreversibility rate clearly depends on the drying temperature (Figs. 3 and 4). When steam is used as a heat source, heat transfer into the drying system occurs at a higher temperature level than in the case of secondary heat. The average temperature difference between the heat source and air is also higher for steam, because the steam temperature remains constant in a heat exchanger. For secondary heat the maximum temperature difference between the heat source and air is only 10 °C. Heat transfer occurring over a high temperature difference also increases the irreversibility rate. The differences between irreversibility rates are several hundred kilojoules per mass of water evaporated in both drying systems. The difference gives theoretical losses in electricity production. For example, using 25 €/MWh as the price of electricity, annual losses in electricity production for multi-stage-drying with three drying stages are c. 78,000euro per mass of water evaporated if steam is used instead of secondary heat. It is important to consider that the irreversibility rate gives only a theoretical estimate of earnings or losses in electricity production. In practice, it is difficult to utilise secondary heat in electricity production.

Figs. 3 and 4 also show that a certain recycle ratio (R), or the number of drying stages (n), gives a minimum value for heat consumption (in SSD, $R = 0.97$ gives a minimum heat consumption for an inlet temperature of 75 °C). The irreversibility rate in both drying systems decreases constantly as the values of R and n increase. The reason why a clear minimum value exists for heat consumption is the heat recovery unit. At certain values of R and n , heat recovery flux has a maximum value (see Table 1). As the values of R and n increase, the heat recovery flux decreases as a result of the smaller mass flow into and out of the drying system. This means that the heat consumption begins to approach the heat consumption of the drying system without a heat recovery unit. If the drying system has no heat recovery unit, the heat consumption decreases constantly as the values of R and n increase (see Fig. 6). The constant decrease of the irreversibility rate is a result of entropy change ΔS , which becomes clearly smaller for drying systems with high values of R and n . Fig. 6 shows that the irreversibility rate in MSD is slightly greater for a drying system without heat recovery. However, the difference is not as remarkable as in the case of heat consumption.

The comparison between the drying systems with the same dimensions (Figs. 6 and 7) shows that the heat consumption is almost the same for both systems. The difference varies from 0 to 50 kJ/kg_{H₂O}. However, the irreversibility rate seems to be smaller for MSD than for SSD. The

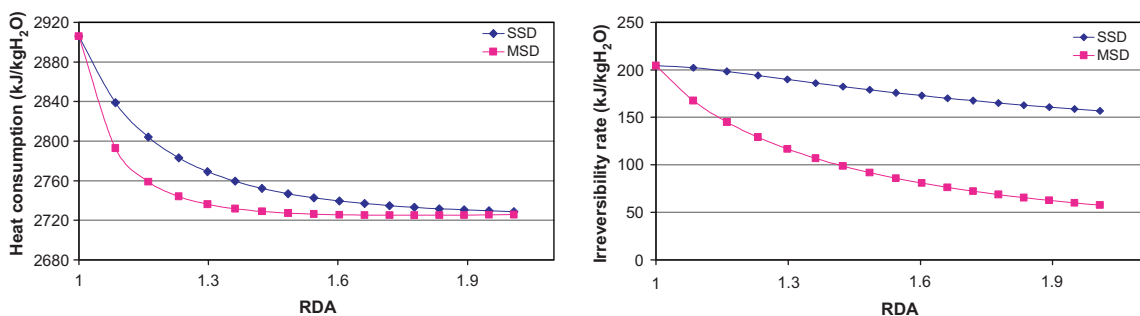


Fig. 6. Comparison of energy consumption and irreversibility rate between single-stage-drying with recycle (SSD) and multi-stage-drying (MSD) for inlet temperature 75 °C. $n_{\max} = 17$, $R_{\max} = 0.8995$.

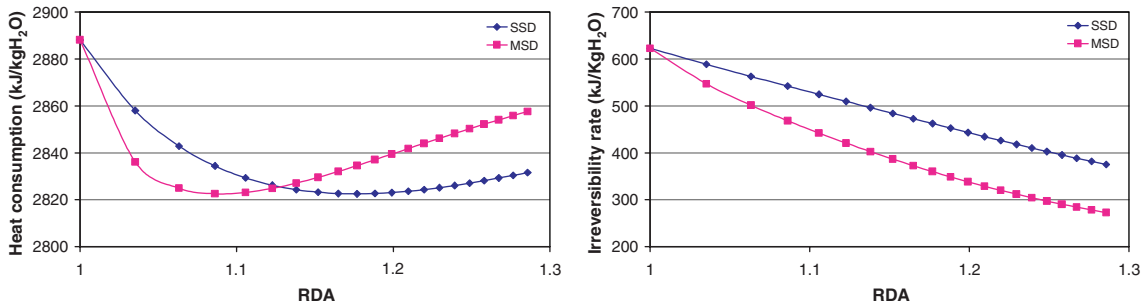


Fig. 7. Comparison of energy consumption and irreversibility rate between single-stage-drying with recycle (SSD) and multi-stage-drying (MSD) for inlet temperature 128 °C. $n_{\max} = 21$, $R_{\max} = 0.8974$.

main reason for the lower irreversibility rate is again the entropy change, which becomes smaller for MSD. According to these results, MSD seems to be a slightly more advanced drying system than SSD. One can also speculate whether it is possible to find an even more advanced drying system by combining air recycling with multi-stage-drying.

The calculated values for heat consumption and irreversibility rate are relatively good, because the heat losses and the pressure drops of the drying system are neglected and the drying air is assumed to be fully saturated after the drying chamber. On the other hand, the increase in fuel temperature is considered, and the temperature differences used in the heat exchangers represent actual values. The results are good reference values for the estimation of the energy efficiency of real drying systems operating in the same way as the example drying systems.

5. Conclusions

Two methods to evaluate the energy efficiency of the biofuel drying have been presented and applied to two different drying systems. Both drying systems are also provided with a heat recovery unit, so as to use the best possible drying systems as example dryers. The traditional evaluation method, which is based on the determination of heat consumption, only gives information about the quantity of the energy used in drying. It does not pay any attention to the quality of the energy. The results show that the heat consumption is quite similar in each comparison example, regardless of the drying temperature or system.

To take into account the quality of the energy used in drying, the evaluation must be based on exergy analysis. In this paper, the exergy analysis has been performed by means of the irreversibility rate. Results show that the irreversibility rate depends to a considerable extent on the temperature of the heat source and also the drying system. A high temperature difference between the heat source and air also increases the irreversibility rate. To decrease the value of the irreversibility rate, the drying temperature should be as low as possible, and the heat transfer should occur over small temperature differences.

Besides the traditional heat consumption, the irreversibility rate is a useful method to compare the performance of drying processes. Especially in the cases where the energy used in drying can be converted to mechanical work, the evaluation of energy efficiency should rather be based on exergy than on energy analyse.

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