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Disposal of Olive Mill Waste Waters through Concentration and Combustion: Comparison between Different Process Options

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Different process options for concentration of olive mill waste water by evaporation are compared with the aim of minimizing the overall energy requirements. In particular, vapor mechanical and steam recompression, as well as the use of heat pumps to qualify the low temperature heat of the vapor stream are considered. Comparison is carried out in terms of overall energy end exergy balances as required from the water evaporation process and made available by the combustion process of the residue.

1. Introduction

Since a long time, the olive mill waste waters (OMWW) represent a well-known and unsolved environmental problem in the Mediterranean countries (Niaounakis and Halvadakis, 2006). This problem became even worse after the introduction of continuous processes using centrifuges, which have increased the volume of produced waste waters. In Italy OMWWs are produced at a rate above $2 \cdot 10^6$ t/y. The main issue associated with the treatment of these waters is the high COD and the contemporary presence of organic compounds recalcitrant to biological treatments. Also, the use as soil fertilized is limited by law and good agricultural practice. This has encouraged the research of treatment process for OMWW based on non-biological operation. Incineration is a suitable application as it is proved by both fundamental (Vitolo et al., 1999) and applied (Di Giacomo et al., 1991) research. Unless a primary fuel is used, incineration in general requires an evaporation/concentration stage to bring the organic content to a level high enough to sustain combustion. Evaporation, however, is an energy-intensive process and the economical feasibility of the whole waste water treatment is strongly dependent on the possibility to achieve some energy recovery from the vapor stream. In this work different process options aiming at reducing the overall energy requirements for an actual plant are compared.

2. Process options

In the following all calculations are carried out with reference to an OMWW flow rate of 0.85 kg s^{-1}. This is the flow rate that we would expect in a 4 months production campaign using a total amount of about 6650 t of olives. Typical OMWW composition

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Table 1. Composition (Pacifico, 1979) and characteristics of OMWW

Component	mass %		
Fats	0.5		
Proteins	1.8		
Sugars	4.5		
Organic acids	0.9		
Polyphenols	1.7		
Polyalcohols	1.1		
Pectins, insolubles, tans	1.5		
Total organic compounds		12	
Inorganic componds		1.5	
Total dry matter at 105 °C		13.5	
Water			86.5

Caracteristic	value
pH	5.4
COD (g/L)	208
BOD (g/L)	90.2

Dry matter elements	weight %
C	55.0
O	14.25
H	8.30
N	1.5
S	0.25
ashes	20.0

is reported in Table 1, with the elemental composition of the dry matter calculated assuming key component formulas. This composition allows the evaluation of the Lower Heating Value (LHV) "as is" by means of the Du Long equation:

$$\Delta H_{CI} = [34.0C(1-x_{\mathrm{w}})+144(H-O/8)(1-x_{\mathrm{w}})-2.60(9H(1-x_{\mathrm{w}})+x_{\mathrm{w}})]\cdot 10^{6} \qquad (1)$$

where x_{w} is the water mass fraction of OMWW. LHV turns out about 1.4 MJ kg^{-1}. The high water content, however, makes them unsuitable to be directly fed to a combustor without the use of additional fuel. In order to sustain combustion it is necessary to reduce the water content in the effluent stream at values lower than 65% (Jenkins et al., 1998). We assume, therefore, a sequence of operations such as those reported in Figure 1 Waste waters are fed to an evaporation stage in which the liquid phase is concentrated to x_1=50%. According to (1), the residue with this water content exhibits a LHV of about 11.9 MJ kg^{-1}. Two main energy fluxes are associated to the whole process in Figure 1 and these are an inward energy flux, Q_{ev}, necessary for liquor evaporation in the evaporator, and an outward energy flux, Q_{comb}, produced by liquor combustion. The mass and energy balances on the evaporator say that

$$L_1 = F(1-x_{\mathrm{F}})/(1-x_1) \qquad (2)$$

$$V_1 = F - L_1 \qquad (3)$$

$$Q_{\mathrm{ev}} = Fh_{\mathrm{F}} - V_1 H_1 - L_1 h_1 \qquad (4)$$

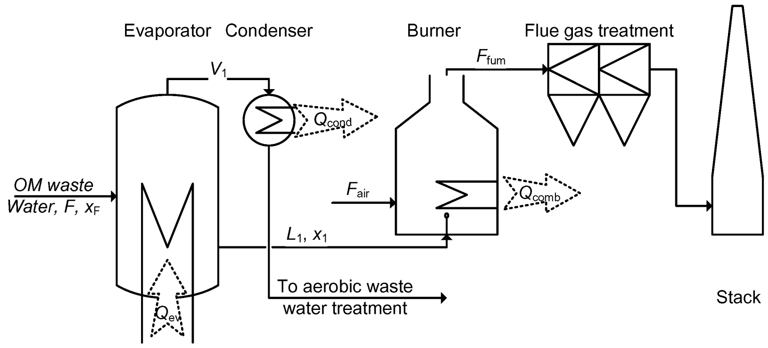

Figure 1. Scheme of the disposal process

Assuming x_F=0.865, x_1=0.5 and the evaporator temperature at 80°C, we have L_1=0.223 kg s^{-1} and Q_{ev}= 1.62 MW. Accordingly the heat released by the combustor is

$$Q_{comb} = L_1 \Delta H_{CI} + F_{air} H_{air} + L_1 h_1 - F_{fum} h_{fum} \tag{5}$$

Assuming fume temperature at 1123°C we have Q_{comb}= 1.95 MW. It is worth noting that Q_{comb}> Q_{ev} and, therefore, the difference Q_{comb} - Q_{ev} can be recovered and rewarded by local legislation as a renewable source of energy. This difference is only a small portion of the whole energy used in the waste water treatment process, but a third heat flux, at lower temperature, is also available in the treatment scheme appearing in Figure 1 and it is the heat released by vapor condensation, Q_{cond}. For environmental reasons the condenser cannot mix the refrigerating stream with the vapor which inevitably contains some volatile components. Its temperature is too low to be really useful outside the treatment process but, as it is well known, the adoption of an energy integration approach allows re-qualifying this heat by increasing its temperature, recovering a large portion of this heat and reducing the energy consumption for the evaporation stage. We considered some options that are summarized in Figure 2.

2.1 Base case

In the base case no heat requalification is made. Saturated steam is used within the evaporator calandria. Its flow rate largely depends on its pressure, which defines the latent heat of condensation, λ_c, and it is approximately:

$$S_v = Q_{ev} / \lambda_c \tag{6}$$

Assuming atmospheric pressure in the calandria, we have S_v= 0.715 kg s^{-1}.

2.2 Mechanical recompression

In this case almost all the vapor produced by the evaporator is compressed to the calandria pressure to a superheated state as represented in the schematic TS diagram in Figure 3a, where S_v*, with enthalpy H_{Sv}*, is the point representing the final state of an isoentropic compression point. The definition of the compressor efficiency, η_c, and the recycle fraction, α, the energy balance on the calandria and that on the compressor provide the equations to calculate the compression work W_c:

$$H_{Sv} = H_1 + \left(H_{Sv}^* - H_1\right) / \eta_c \tag{7}$$

$$\alpha = Q_{ev} / \left(H_{Sv} - h_{D1}\right) V_1 \tag{8}$$

$$W_c = \left(H_{Sv} - H_1\right) \alpha V_1 \tag{9}$$

Using the flow rates of these case and a compressor efficiency η_c = 0.75, we obtain α = 1 and W_c = 120 kW for this process option.

2.3 Ejector recompression

With thermal recompression the energy for steam compression is given by a secondary high pressure steam injected in the low pressure stream by means of an ejector. The

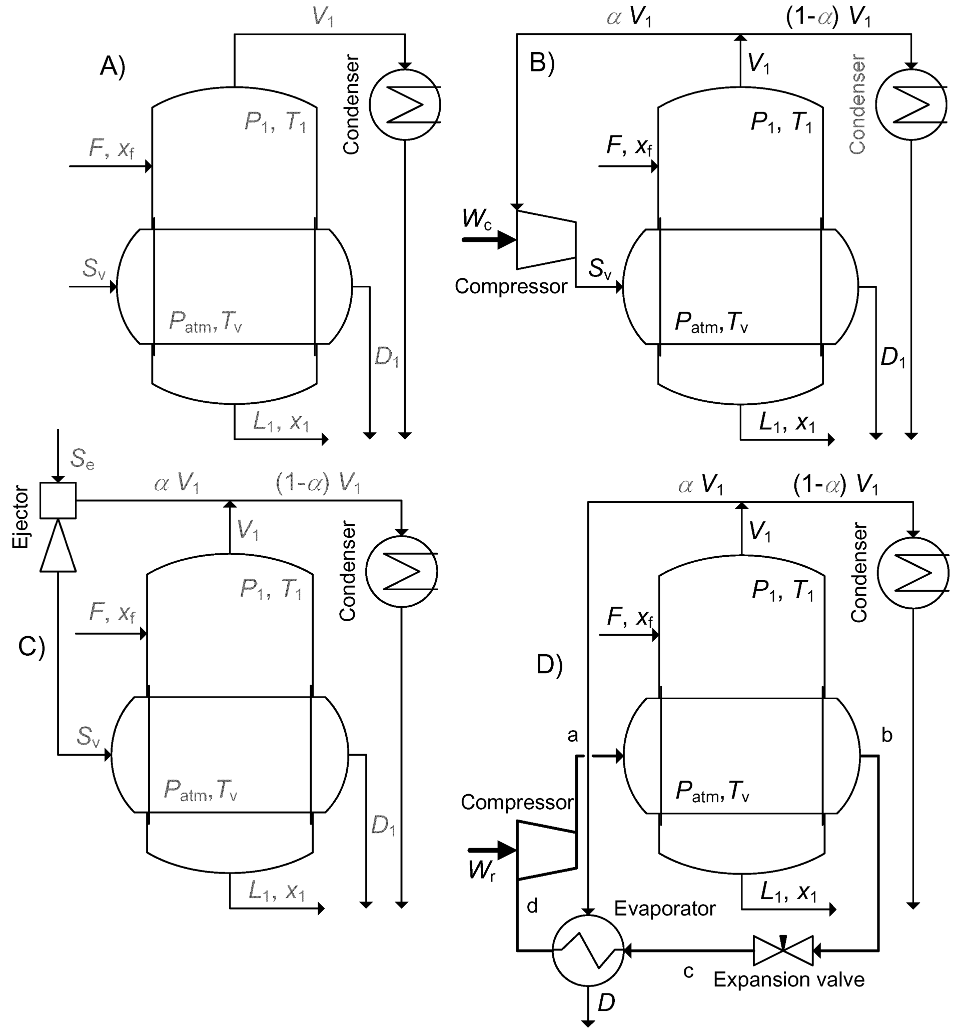

Figure 2. Schemes of the evaporation operation. A) base case; B) vapor mechanical recompression; C) vapor recompression with an ejector; D) use of a heat pump.

effectiveness of the ejector is characterized by the entrainment ratio R, that is mainly a function of the primary steam pressure and of the required pressure increase for the steam leaving the stage. We assume a saturated secondary stream at 10 bar and 180°C and, hence, $R \cong 2$ according to Kern (1950). The definition of the entrainment ratio and the mass and energy balances are, respectively:

$$\alpha V_1 / S_e = R \tag{10}$$

$$S_v = \alpha V_1 + S_e \tag{11}$$

$$S_v H'_{Sv} = \alpha V_1 H_1 + S_e H_e \tag{12}$$

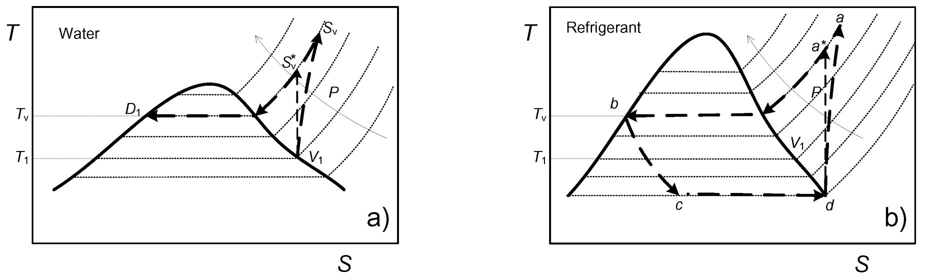

Figure 3. Thermodynamic cycles: a) mechanical recompression; b) heat pump.

This set of equations allows evaluating $\alpha = 0.76$, $S_e = 0.238\ \mathrm{kg\ s^{-1}}$ and $H'_{Sv} = 2691\ \mathrm{kJ\ kg^{-1}}$. The circumstance that $H'_{Sv} \cong H_{Sv}$ indicates that the vapor state obtained with thermal recompression is very similar to that used in the base case. The total energy used in this process is $S_eH_e = 660$ kW.

2.4 Use of a heat pump

The heat pump has to transfer heat from the condensing stream of outgoing vapor to the calandria operating at higher temperature. To this purpose a closed cycle heating system is to be used (see Figure 2D). The corresponding thermodynamic cycle is given in Figure 3b. The working fluid (refrigerant) is chosen on the basis of the cycle temperatures. In this case the most appropriate was Du Pont SUVA 236fa. The refrigerant pressure in the calandria (line a-b in Figure 3b) will be that for which the condensation temperature is 100°C in order to keep unchanged the OMWW evaporator temperature. A temperature of 60°C will be assumed in the refrigerant evaporator (line c-d in Figure 3b) in order to keep also in this exchanger the same temperature difference existing in the calandria and, therefore, have similar exchange areas. Energy balances on the refrigerant condenser, the compressor and the refrigerant evaporator (internal and external) allow the evaluation of the refrigerant mass flow rate, F_r, the compressor work, W_r, the heat exchanged in the refrigerant evaporator Q_{re} and α:

$$F_r = Q_{ev} / (H_a - h_b) \tag{13}$$

$$W_r = (H_a - H_d)F_r \tag{14}$$

$$Q_{re} = (H_d - h_c)F_r \tag{15}$$

$$\alpha = Q_{re} / \lambda_1 V_1 \tag{16}$$

where λ_1 is the latent heat of condensation of the vapor stream V_1. Applying equations (13) to (16) we obtain $\alpha = 0.85$ and $W_r = 390$ kW.

3. Comparison of results

Energy requirements of the different process options are reported in Table 2. The different processes require heats of different qualities or even mechanical work. A simple energy balance, therefore, is not sufficient for a correct comparison between the process options. An exergetic analysis is a simple approach that, without the evaluation of the different investment costs, compares the different process options in term of the potential production of valuable energy. Exergy $Ex = Q(1 - T/T_{env})$ is the maximum

Table 2. Energy and exergy balance on the process

Case	Combustion Energy (kW)	Combustion Temp (K)	Combustion Exergy (kW)	Evaporation Energy (kW)	Evaporation Temp (K)	Evaporation Exergy (kW)	Condensation α	Condensation Energy (kW)	Condensation Exergy (kW)	Total ΔEx (kW)
Base case	2650	1123	1950	-1620	373	-325	0	1450	328	1950
Mech. compr.	2650	1123	1950	-120	work	-120	1	0	0	1830
Therm. compr.	2650	1123	1950	-660	453	-227	0.76	347	79	1800
Heat pump	2650	1123	1950	-390	work	-390	0.85	220	49	1610

amount of work that is ideally obtainable from a certain energy flux, Q, according to the second thermodynamic principle. T_{env} is the ambient temperature.

A summary of the energetic and exergetic fluxes for different process options is given in Table 2. Furthermore, in all process options the net available exergy is given by the exergy released by combustion minus the exergy consumed by the evaporation operation, Ex_{ev} plus the exergy available from the outgoing vapor condensation:

$$Ex_{cond} = (1-\alpha)V_1\lambda_1(1-T_1/T_{\text{env}}) \tag{17}$$

$$\Delta Ex = Q_{comb}(1-T_{comb}/T_{\text{env}}) - Ex_{evap} + Ex_{cond} \tag{18}$$

Inspection of Table 2 clearly indicates that the evaporation stage based on mechanical compression is the least exergy using case. However, when considering the recovery of the condensation heat, most of the options are equivalent. In absence of recovery of condensation heat, thermal compression might compete with mechanical compression in terms of overall convenience for the considerably larger simplicity of the vapor compressing device. The heat pump is the most ineffective process scheme, even worse than the simple evaporator case. The increased number of equipment pieces required by this process option with respect to the others indicates that including investment costs in the analysis would bring to even larger economic disadvantages.

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