

# POSSIBILITY FOR MECHANICAL VAPOR RE-COMPRESSOR FOR STEAM BASED DRYING PROCESSES

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## ABSTRACT

The application of a high temperature heat pump for superheated steam drying (SSD) process was analyzed. The SSD chamber processes 10000 kg h<sup>-1</sup> of product into dried pet food. Two different systems (“open” and “closed”), which use water as a refrigerant, were evaluated. Both systems use energy from the surplus steam, which is formed due to evaporation of moisture during drying. It was concluded that compression in three stages is required to achieve temperature of steam at 170.0 °C, when radial turbo compressors were used to fulfill the volumetric flow demand. The most efficient system is the “open” system, which provides a coefficient of performance (COP) of steam generation at 3.84. Drying efficiency (DE) is calculated at 0.19 kWh kg<sup>-1</sup> of evaporated water. The system provides hot tap water (371.0 kW). The “open” system provides annual saving of 7.45 mln. NOK. Also low investment cost of compressor make the system beneficial.

## INTRODUCTION

Estimates evaluate that drying is responsible for 15-25% of the overall global industrial energy demand in developed countries and efficiencies in food drying can be as low as 10.0%, with 35.0-45.0% being the average (Mujumdar, 2007). The high temperature drying is applied in different branches of industry: dried food production (for human consumption, pet food, aquaculture etc.), wood industry (drying of woods); construction materials and many others. The so called superheated steam drying (SSD) can be used at several conditions with the aim of increasing drying rates, decrease energy consumption and prevent oxidation (Devahastin and Mujumdar, 2014; Pakowski and Adamski, 2011). A superheated steam drying can operate with water vapor at atmospheric pressure and in the temperature range between 160.0 and 200.0 °C. The evaporation of moisture from product goes at higher rates, when compared with hot air drying, at such temperatures (Mujumdar, 2007). A typical drying unit consists of few key elements: drying chamber, fan and steam generator. Vapor from the outlet of the drying chamber is usually in the range between 100.0 and 120.0 °C. It should be noted that steam consumption is very high for this process due the high enthalpy of water evaporation. Thus the energy recovery is essential for SSD. The simplest solution is recirculation of the exhausted steam

through steam generator; this method provides high efficiency of the process. The steam, which evaporates from the product during drying, can be used for different types of bottom cycles. This also improves the energy efficiency of the system.

However, a steam generator can be driven by combustion of fuel or by electrical heater. An efficient technology, which provides energy savings, is required for large scale industrial units. The heat pump technology can be a good solution for steam regeneration after drying chamber. Statistically, the heat pump assisted drying is approximately 4 times more efficient when compared with the hot air drying (Colak and Hepbasli, 2009a). At the same time the SSD is significantly different when compared with low and medium temperature heat pump drying processes:

- high inlet temperature of refrigerant on the condenser side ( $>180.0\text{ }^{\circ}\text{C}$ )
- high outlet temperature of refrigerant on the condenser side ( $> 115.0\text{ }^{\circ}\text{C}$ )
- high temperature on the evaporator side ( $> 90.0\text{ }^{\circ}\text{C}$ )
- high temperatures on both sides of the processes lead to a high discharge temperature of the compressor.

The scope of this article is an attempt to analyze possible ways of high temperature heat pump technology for SSD of pet food. The technological solutions are based on the newly developed radial turbo-compressor which uses water as a refrigerant.

## MATERIALS AND METHODS

### *Refrigerant description*

The high temperature on the condensing side limits the number of refrigerants which can be used for steam regeneration; only few of them are available on the market nowadays: R600, R600a, R717 and R718. The first three are suitable for trans-critical heat pump circle due to relatively low critical temperature (less than  $150.0\text{ }^{\circ}\text{C}$ ). This will cause a high pressure on the gas cooler side. At the same time water (R718) has high enough critical temperature ( $374.2\text{ }^{\circ}\text{C}$ ) to provide efficient condensation. The condensing pressure of water at  $180.0\text{ }^{\circ}\text{C}$  is approximately 10.0 bars, this will decrease manufacturing price of equipment. Relatively high condensation energy ( $2014.0\text{ kJ kg}^{-1}$ ) will decrease the swept volume of compressor. At the same time, the research paper of Lancher et al. (2007), which is devoted to commercial feasibility of the use of water vapor as refrigerant, explained that water-based vapor compression refrigeration systems will not be economically attractive without substantial and successful efforts to develop low-cost, high capacity compressors.

### *Compressor description*

The high speed radial turbo compressor is developed to provide capacities between 300.0 and 5000 kW. The construction is derived from supercharger and can operate at 100000 r.p.m., thus it is sustainable for high temperature heat pumps of high capacity. The oil is used for bearings lubrication (self-contained oil system) which helps to decrease the complexity of the heat pump system. The low investment costs, when compared with screw or piston compressor and relatively small size of the radial turbo compressors gives additional benefits for use as a compressor in high temperature heat pump circle. Turbo compressors are generally associated with significantly lower investment costs; compare to screw compressor the investment costs can be reduced by a factor of 100. For further details on the specification of the radial turbo charger concept the interested reader is referred to the work Weel et al., 2013.

### System description

Conventional superheated steam drying system is shown on Fig. 1. Such system is simple and does not require high investments. The steam generator (1) produces steam for warming up the whole system. The electrical regenerator of superheated steam (4) is used for regeneration of superheated steam; the inlet temperature of steam to the regenerator is 110.0 °C, the outlet temperature is 170.0 °C. Water evaporates from product during drying process and leaves the drying chamber (2) with the steam flow. The circulation of the steam is provided by fan (3). Product comes to the drying chamber after extruder and has temperature 100.0 °C, the outlet temperature is 105.0 °C. The heat balance of the drying chamber is presented in Table 1. Inlet moisture content is 25.0% w.b., outlet moisture content is 5.0 % w.b.

Table 1. Heat balance of superheated steam dryer

Type of load	Mass flow, kg s <sup>-1</sup>	Load, kW	Superheated steam demand kg s <sup>-1</sup>
Q <sub>heating of solids</sub>	2.19	23.70	0.20
Q <sub>evaporation</sub>	0.58	1329.80	11.12
Q <sub>losses</sub>	-	203.02	1.70
Total/Q <sub>load</sub>	2.77	1556.54	13.02

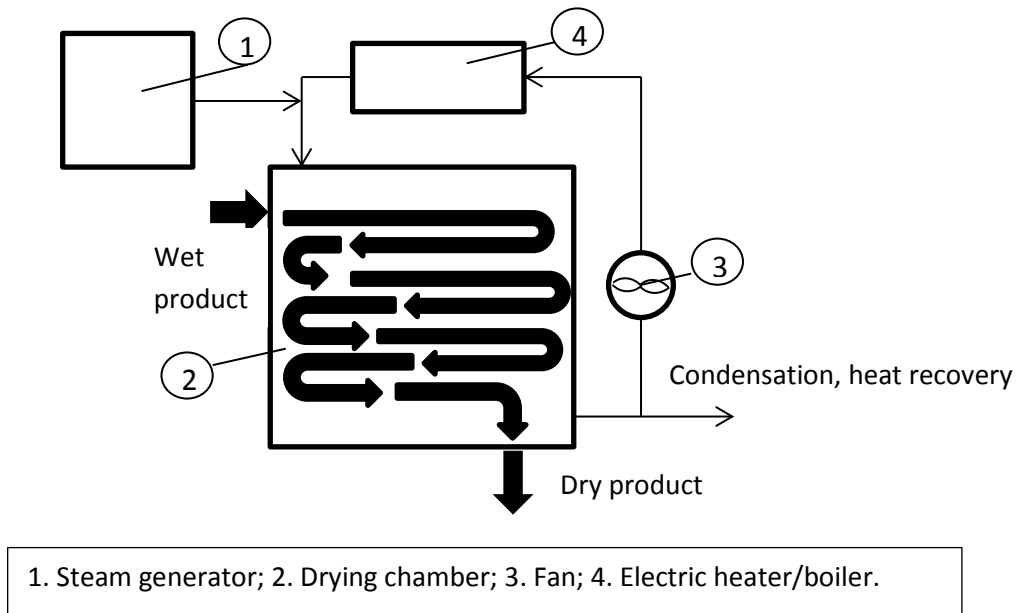


Figure 1. Conventional superheated steam dryer.

Total load (kW) is calculated as a sum of different loads using equation 1:

$$Q_{\text{load}} = Q_{\text{heating of solids}} + Q_{\text{evaporation}} + Q_{\text{losses}} \quad (1)$$

where energy which is used for heating of solids  $Q_{\text{heating of solids}}$  (the fraction which is not evaporated during drying) is determined by equation 2 as a product of mass flow of solids,  $\dot{m}_{\text{solids}}$  (kg s<sup>-1</sup>), by heat capacity,  $c_p$  (kJ kg<sup>-1</sup> K<sup>-1</sup>) and by temperature difference,  $\Delta t$  °C:

$$Q_{\text{heating of solids}} = \dot{m}_{\text{solids}} * c_p * \Delta t \quad (2)$$

Evaporated moisture  $Q_{evaporation}$  determined by equation 3 as a product of mass flow of evaporated water,  $\dot{m}_{water}$  ( $\text{kg s}^{-1}$ ), by enthalpy difference,  $\Delta H$  ( $\text{kJ kg}^{-1}$ )

$$Q_{evaporation} = \dot{m}_{water} * \Delta H \quad (3)$$

Heat losses through the walls and pipes during drying and heat transportation were determined by eq 4

$$Q_{losses} = \sum A * U * \Delta t \quad (4)$$

Where  $A$  – surface of the pipeline or drying chamber,  $\text{m}^2$ ,  $U$  – overall heat transfer coefficient,  $\text{W m}^{-2} \text{K}^{-1}$ ,  $\Delta t$  – temperature difference between environment and steam,  $^{\circ}\text{C}$ .

Most of the energy is consumed for water evaporation, and the superheated steam flow is higher in the outlet of the drying chamber, when compared with inlet to the drying chamber. This surplus of steam ( $0.58 \text{ kg s}^{-1}$ ), which is formed by moisture evaporation from product, can release high energy when condensing. The condensation temperature ( $100.0^{\circ}\text{C}$ ) is sufficient for tap water production. The use of electric or combustion energy for regenerator ( $\eta_{el}=0.95$  and  $\eta_{boil}=0.85$ ) is not effective from energetic point of view. The drying efficiency  $DE$   $\text{kWh kg}^{-1}$  of evaporated water is determined by equation 5:

$$DE = \frac{Q_{load}}{\dot{m}_{water} * 3600 * \eta_{el \text{ or } boil}} \quad (5)$$

Drying efficiency shows the amount of energy, which is required for evaporation of moisture from a product. The theoretical DE is calculated at  $0.75 \text{ kWh kg}^{-1}$  of water, at the same time, the real DE will be between  $0.79$  and  $0.88 \text{ kWh kg}^{-1}$  of water. The surplus steam can be used for heat pump circle, which will fulfill the demands of superheated steam regenerator. The DE will decrease as a result.

#### *Heat pump system description*

The use of heat exchanger as a regenerator of superheated steam requires a temperature difference to provide efficient heat exchange. The traditionally accepted difference of  $10.0^{\circ}\text{C}$  between fluids is taken for calculation, thus the condensing temperature is  $180.0^{\circ}\text{C}$  (pressure  $10.0$  bars).

Two systems, which use water as a refrigerant, are analyzed in this paper. The first system is an “open” system (Figure 2) which sucks in superheated steam from the main flow of steam at atmospheric pressure ( $1.0$  bar). The second system is a “closed” system (Figure 3), which absorbs energy of surplus steam through evaporator; the evaporating temperature is  $90.0^{\circ}\text{C}$  (pressure  $0.7$  bar).

The pressure ratios for systems are  $10.0$  and  $14.3$  respectively. Such pressure ratios are far above working range of a radial turbo compressor, thus several compression stages are required. It should be noted that even screw compressor will be not suitable for such processes due to the extremely high discharge temperature ( $>525.0^{\circ}\text{C}$ ). Three stages of compression are considered to be the most suitable solution for the both systems. The pressure in the intermediate pressure vessels is calculated by equation 6 using problem solver (Excel 2007):

$$\frac{P_{ev}}{P_{i1}} = \frac{P_{i1}}{P_{i2}} = \frac{P_{i2}}{P_{cond}} \quad (6)$$

where  $p_{ev, i1, i2, cond}$  – pressure on the evaporator side, in the first intermediate pressure vessel, in the second intermediate pressure vessel, in condenser.

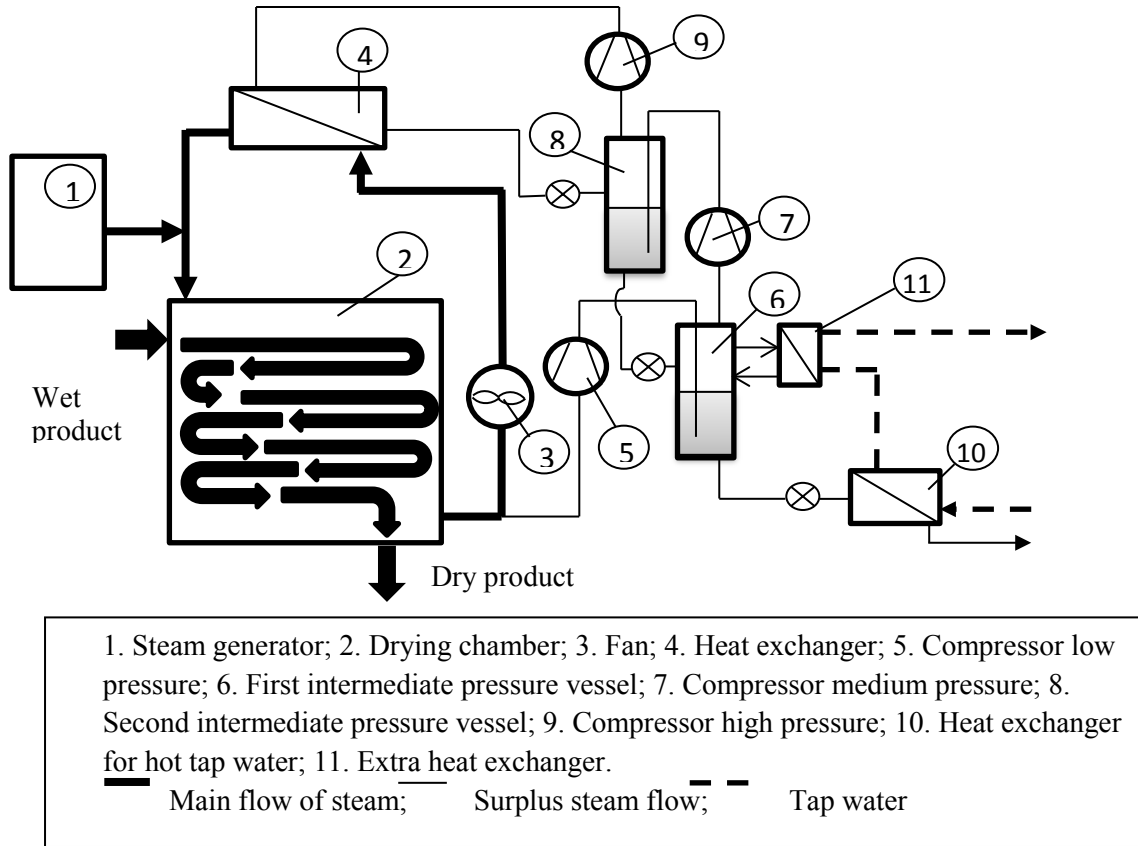


Figure 2. Heat pump assisted superheated steam dryer, "open" system.

The "open" system takes surplus vapor directly to low pressure compressor (5); Fig. 2. The superheated discharged gas is cooled down to saturated vapor point in the first intermediate pressure vessel (6) and then it is sucked into medium pressure compressor (7). The same operation is repeated for the second intermediate pressure vessel (8) and high pressure compressor (9). Then the refrigerant goes to the condenser (4), where it is condensed to saturated liquid point without sub-cooling. After that it is throttled to second intermediate pressure vessel (8). Vapor, which is formed after throttling, is sucked into the high pressure compressor (9). The same operations are repeated for first intermediate pressure vessel (6). The liquid water will be throttled to atmospheric pressure level after first intermediate pressure vessel. The water at 100 °C and water vapor (4.0 %) are cooled down and condensed in the heat exchanger for tap water (10). The "closed" system is introduced on Fig. 3. The main difference from the "open" system is that steam circulates in the separated circle, and evaporator (10) is used to recover energy from surplus steam.

## RESULTS AND DISCUSSION

The capacity of condenser (4) will equal to total load  $Q_{load}$  which is calculated by equation 1 (the zero losses in condenser are assumed). Thus the amount of refrigerant, which will circulate on the high pressure side, will be calculated by dividing  $Q_{load}$  by enthalpy difference. The refrigerant flow, which circulates between first and second intermediate pressure vessels, is calculated by energy balance ( $Q_{inlet}=Q_{outlet}$ ).

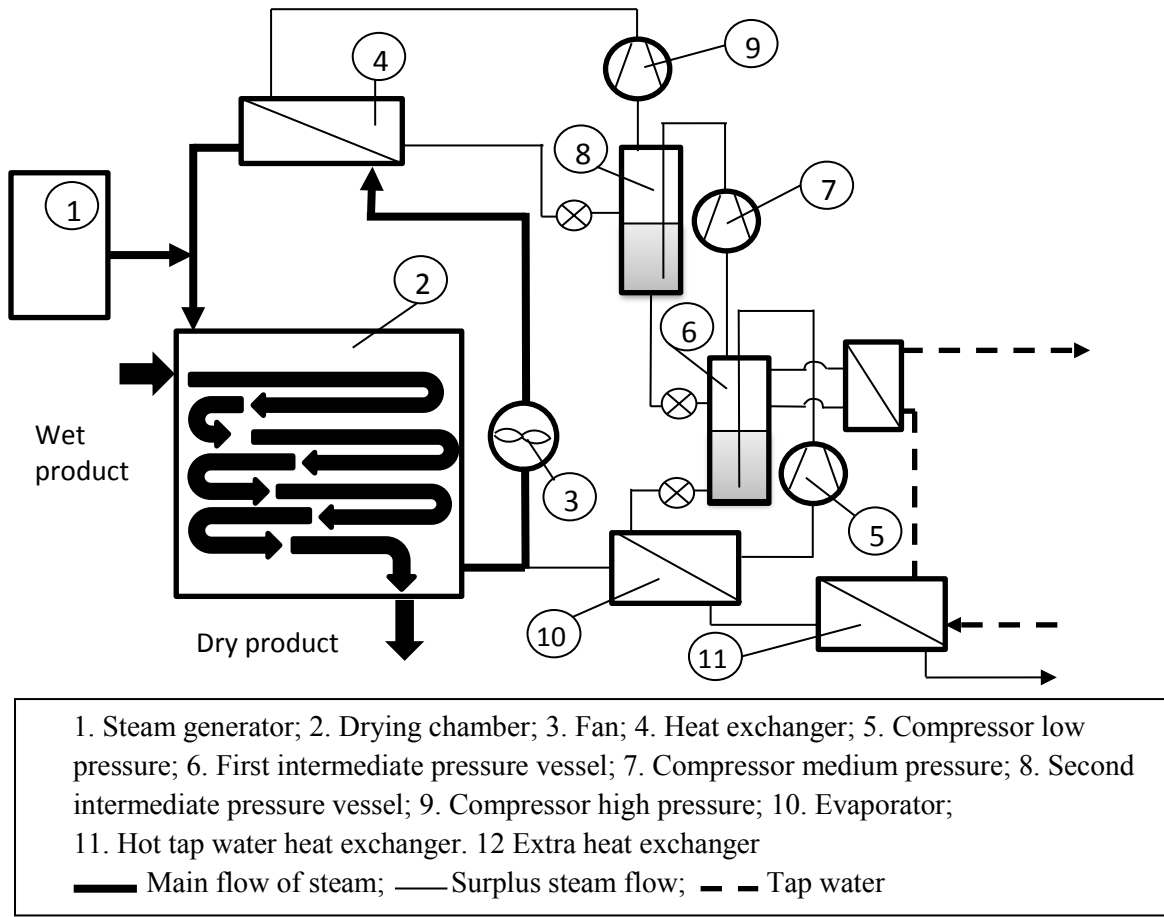


Figure 3. Heat pump assisted superheated steam dryer, “closed” system.

The mass flow on the low stage equals surplus steam flow for “open” system. The mass flow on the low stage of “closed” system is calculated by dividing energy of surplus steam condensation in evaporator by enthalpy of condensation at 0.7 bars. It was found that the mass flow on the low stage is significantly higher, that the flow required for stable and effective work of the system (valid for both systems), thus the additional heat exchanger was used with the first intermediate pressure vessel. The aim is to condense excess of superheated vapor from low pressure compressor. This will help to avoid the increasing of pressure in the intermediate pressure vessel during operation. The capacity of the heat exchanger,  $Q_{extra}$  (kW), is calculated by equation 7 (assuming constant pressure and enthalpies of the process)

$$Q_{extra} = \dot{m}_{LP} * (h_{com,LP, out} - h_{i1, out}) - \dot{m}_{MP} * (h_{comp, MP, in} - h_{i2, out}) \quad (7)$$

where  $\dot{m}_{LP, MP}$  – mass flow,  $kg\ s^{-1}$ , on low pressure side and on medium pressure side;  $h$  – enthalpy,  $kJ\ kg^{-1}$ , discharge of the low pressure compressor, outlet for first intermediate pressure vessel to expansion valve, inlet to medium pressure compressor, outlet form second intermediate pressure vessel to expansion valve. Additional heat exchanger helps to increase heat recovery by 2.4 % for tap water production, at the same time the performance of the heat pump will be lower, due to higher mass flow which should be compressed on the first stage. The coefficient of performance of the systems, COP (-), are determined by equation 8:

$$COP = \frac{Q_{load} * \eta_{dr}}{W_{total}} \quad (8)$$

where  $W$  – energy used to maintain process: energy for boilers or for compressors, kW,  $\eta_{dr}$  – is efficiency of the driving device (motor, boiler, etc.), the isentropic efficiencies of compressor is 0.75 for both systems.

The calculated parameters of the systems are introduced in Table 2. The discharge temperature of both heat pumps after high pressure compression is in the range between 255.0 and 269.0 °C, which is not critical for lubrication of bearings. The low stage of the closed system operates below atmospheric pressure, which can cause penetration of ambient air to the system.

Table 2. Characterization of “open” and “closed” heat pumps systems.

Stage		Mass flow, kg s <sup>-1</sup>	Volume flow, m <sup>3</sup> h <sup>-1</sup>	Suction pressure, bar	Discharge temperature, °C	Compressor work, kW
I	Open	<u>0.58</u>	<u>3619.0</u>	<u>1.01</u>	<u>210.0</u>	<u>113.9</u>
	Closed	0.61	5284.0	0.70	203.0	133.5
II	Open	<u>0.62</u>	<u>1821.0</u>	<u>2.17</u>	<u>225.0</u>	<u>123.0</u>
	Closed	0.61	2246.0	1.70	233.0	139.9
III	Open	<u>0.71</u>	<u>1023.0</u>	<u>4.67</u>	<u>255.0</u>	<u>148.2</u>
	Closed	0.71	1149.0	4.13	269.0	173.4
<b>Total</b>						<u>385.1</u> 446.7

Table 3. Comparison of the different systems for superheated steam regeneration.

	Used energy, kW	Steam regeneration, kW	Water heating, kW	COP total, -	COP steam, -	DE, kWh kg <sup>-1</sup>
<b>Energy source: Electricity</b>						
Traditional scheme	1638.6	1556.6	1542.6	1.89	0.95	0.79
Open system	405.4	1556.6	371.0	4.75	3.84	0.19
Closed system	470.2	1556.6	406.5	4.17	3.31	0.22
<b>Energy source: Fuel/Gas</b>						
Traditional scheme	1831.4	1556.6	1542.6	1.69	0.85	0.88
Open system	770.2	1556.6	371.0*	2.50	2.02	0.37
Closed system	893.4	1556.6	406.5*	2.20	1.74	0.43
* Exhausted gas recovery was not taken into account						

Comparison of the “open”, “close” and traditional systems are shown in Table 3. The high consumption of consumption for compression results in lower COP and higher DE of “closed” system, when compared to “open” system. However both systems show high drying efficiency, when compared with traditional superheated steam drying methods. The “open” system is found to be the most effective for steam regeneration. This can be explained by decreasing of the heat exchange losses (system without evaporator). The low investment costs of radial turbo compressor decreases the pay-back time,

when compared with, for example, screw compressors. The potential annual savings due to the decreasing of the energy consumption is calculated at 0.9 mln. €. This value can vary due to fluctuation of the electricity price.

## CONCLUSIONS

The application of a high temperature heat pump for superheated steam drying (SSD) process shows good potential when water is used as a refrigerant. In this study the SSD chamber processes 10000 kg h<sup>-1</sup> of product into dried pet food. The “Open” and “closed” systems (compression in three stages), which use water as a refrigerant, were evaluated. Both systems use energy of surplus steam, which is formed due to evaporation of moisture during drying. The condensing temperature is 180.0 °C. The systems provide production of hot tap water (371.0 and 406.5 kW) in addition to steam regeneration process. Radial turbo compressor was used to fulfill the volumetric flow demand. The most efficient system is the “open” system. It provides coefficient of performance (COP) of steam generation at 3.84 when “closed” system gives COP of 3.31. Drying efficiencies are calculated at 0.19 kWh kg<sup>-1</sup> of evaporated water for the “open system” and at 0.22 kWh kg<sup>-1</sup> for the “closed” system. The disadvantages of the “closed” system are heat exchange losses and high suction volumes, due to low suction pressure, when compared with the “open” system. The “open” system provides annual saving of 7.45 Mln. NOK. Also low investment costs of compressors make the system beneficial.

## NOTATION

A	area of heat exchange (m <sup>2</sup> )	W	work of process (kW)
c <sub>p</sub>	heat capacity (kJ kg <sup>-1</sup> K <sup>-1</sup> )	U	overall heat transfer coefficient (W m <sup>-2</sup> ·K <sup>-1</sup> )
DE	drying efficiency (kWh kg <sup>-1</sup> )	η	coefficient of efficiency (kW kW <sup>-1</sup> )
m	mass flow, (kg s <sup>-1</sup> )	ΔH	enthalpy difference (kJ kg <sup>-1</sup> K <sup>-1</sup> )
Q	capacity or load, (kW)	Δt	temperature difference (°C)
p	pressure, bar		

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