ENERGY-EFFICIENT EVAPORATIVE DEHYDRATOR BASED ON STEAM HEAT PUMP TECHNOLOGY

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ABSTRACT

An energy-efficient dehydrator was conceived for the purpose of the watery-waste recyclings. The steam generated from preheated wastes is compressed to condense at an elevated temperature to reheat the wastes. The condensate is drained out by a steam trap. This is a kind of heat pump using H_2O as a working fluid in an open cycle.

An experimental dehydrator was prototyped to investigate the performance characteristics. The steam compressor used was an oil-free wing-type driven by a 7.5 kW-motor. A steam condenser acting as a rotating agitator was tried to improve heat transfer. Measured distillation rates and COPs were close to the predicted, 100 kg per hour and 10 respectively.

Lees of shochu (distilled wheat spirits) was dried from the initial water ratio of 9.00 to 0.13. Sum of the latent heat of vaporization divided by the integrated electricity consumption was approximately 7, that is translated into the reductions to less than one quarter in the primary energy consumption and in the carbon dioxide emission when compared with the conventional dryers.

1. INTRODUCTION

Watery wastes such as sludge, swill, and effluent are discarded in large quantity from our society and they are mostly hard-to-treat in separation, in storage, in transportation, and in disposal. They are sometimes incinerated with the aid of burning fossil fuels. Dehydrating these troublesome wastes effectively reduce their mass and occasionally facilitate their recyclings. However, due to large latent heat of vaporization (2,257 kJ/kg at 100°C), conventional dryers consume considerable amount of energy and consequently their running costs are high.

The author has been concerned with R&D of unique heat pumps (Hino 1996) and it was natural to conceive the vapor recompression cycle (VRC) for energy-efficient dehydration. The VRCs are usually large-scale equipments employing centrifugal compressors of hundreds kilowatts and they are mainly applied to concentrating liquid materials. In applications for the watery wastes, small-scale VRCs would be preferable for on-site dehydrations and additionally they must handle wet solid materials. The VRCs that can meet these requirements seemed unknown.

Feasibility studies of the small-scale steam heat pump dehydrator started in 1996. As the first step, tap water was distilled. The steam compressor employed was a modified wing-type oil-free vacuum pump driven by a 3.75 kW-motor. In 1998, the first prototype dehydrator with a 7.5 kW-compressor was made and applied to sewage-sludge. The results showed the heat transfer capability of the double-wall jacket condenser was inadequate (Hino 1999). Base on this experience, the second-generation prototype equipped with a rotating agitator-condenser was made and tested in 2000 (Hino 2001), which this paper mainly describes.

2. STEAM HEAT PUMP CYCLE

The steam generated from drying materials is compressed to condense at an elevated temperature to reheat the material. Thus, the latent heat of steam is recovered whereas conventional dryers dissipate the latent heat to the atmosphere. A fundamental configuration of the steam heat pump dehydrator is illustrated in Fig. 1. Suppose the watery waste in the vessel were preheated to 95°C, water boils at 84 kPa exerted by the suction of the compressor. Pressure of the steam is raised by the compressor to 143 kPa and its saturation temperature is 110°C in this example. Heat flows from the condenser to the drying material by the temperature difference of 15 K. The pressures may vary according to the circumstances. The steam trap passes the condensate to drain.

By giving compression work W, a great amount of heat Q is moved from a lower temperature T_1 [K] (i.e. drying material) to a higher temperature T_2 [K] (i.e. condenser). The theoretical rela-

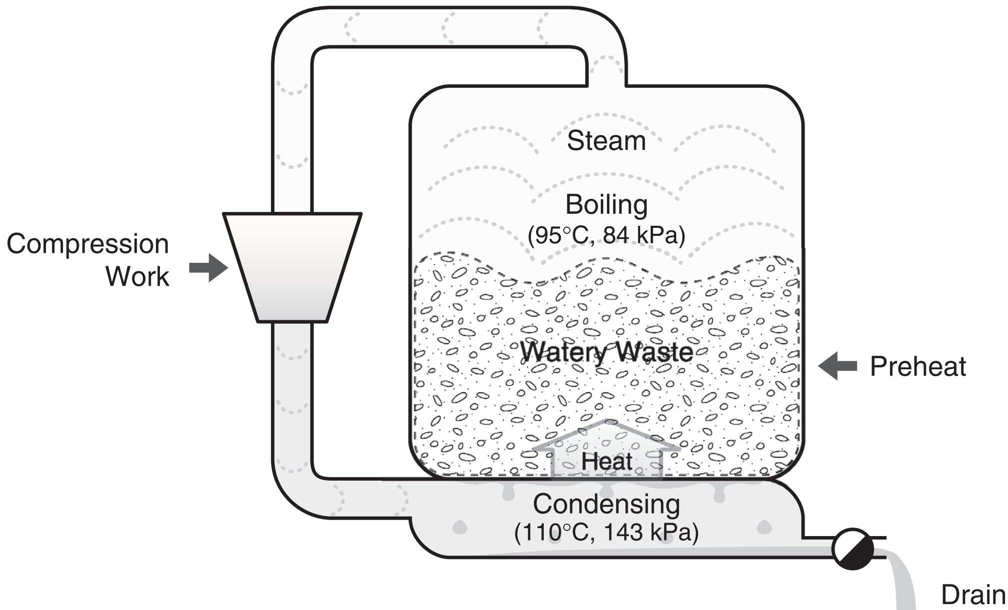

Fig. 1 Fundamental Configuration of Steam Heat Pump Dehydrator
(pressures and temperatures may vary)

tionship is derived by the preservation of entropy as follows.

$$Q/W = T_1/(T_2 - T_1) \quad \text{(1)}$$

Equation 1 expresses a theoretical coefficient of performance and substituting the temperatures in Fig. 1 yields 24.5. The COPs of real-world heat pumps may reduce almost in half but it is still energy-efficient.

Figure 2 depicts a pressure-enthalpy diagram of the steam heat pump cycle and the following mathematical expressions are formulated in a steady-state condition.

$$q_S = h_{sl}(T_b) - h_a = \int_{T_a}^{T_b} c_p dT \quad \text{(2)}$$

$$q_L = h_{sv}(T_b) - h_{sl}(T_b) \quad \text{(3)}$$

Adiabatic compression work is calculated as follows.

$$W_{ad} = \kappa/(\kappa - 1)\, p_{suc}\, v_{suc}\left[\Pi^{(\kappa-1)/\kappa} - 1\right] \quad \text{(4)}$$

$$\Pi = p_{dis}/p_{suc} \quad \text{(5)}$$

$$p_{suc} = p_b - \Delta p_{SL} \quad \text{(6)}$$

$$p_{dis} = p_c + \Delta p_{DL} \quad \text{(7)}$$

$$v_{suc} = v_b\,(p_b/p_{suc})(T_{suc}/T_b) \quad \text{(8)}$$

Mass flow rate of steam handled by a compressor is expressed as:

$$G = \eta_v\, V_{PD}/v_{suc} \quad \text{(9)}$$

$$V_{PD} = n\,V_{CY}\,. \quad \text{(10)}$$

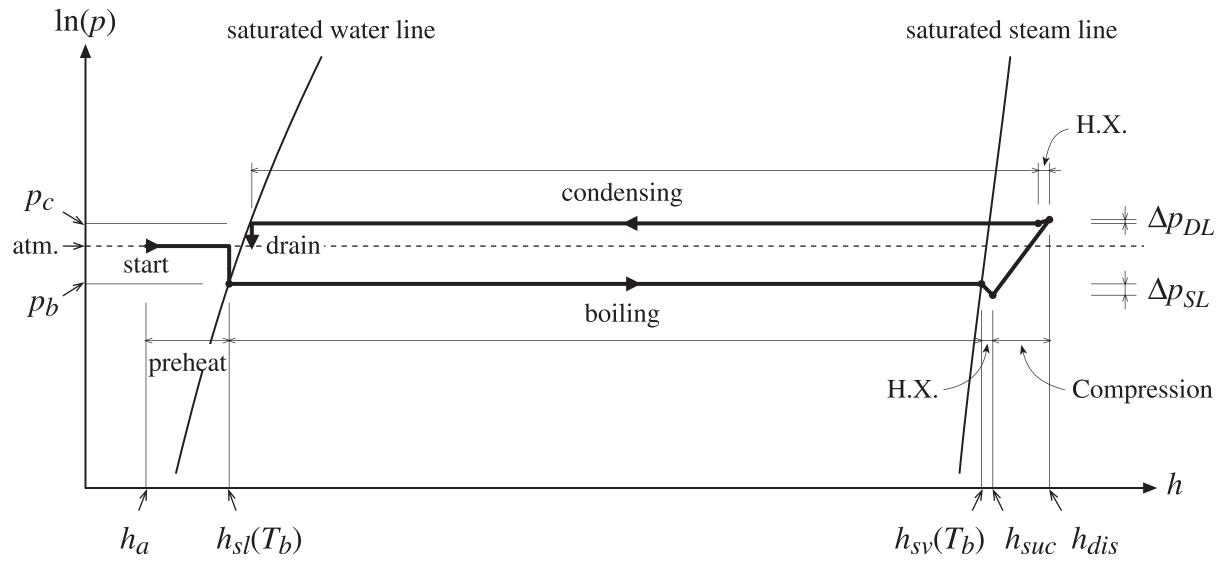

Fig. 2 Pressure-Enthalpy Diagram of Steam Heat Pump Dehydrator

Enthalpy change in a compressor is:

$$h_{dis} - h_{suc} = W_{ad} / \eta_{ad} - Q_{CL} / G \,. \qquad (11)$$

Heat is exchanged between the discharge and the suction steam for the purpose of superheating the suction steam as denoted H.X. in Fig. 2. Without superheating, wet steam was aspirated and the volumetric efficiency decreased some 20% in the early tests. Latent heat of boiling is:

$$Q_b = G\, q_L \,. \qquad (12)$$

Motor input and COP are figured out as follows:

$$E_{MT} = G\, W_{ad} / \left(\eta_{ad}\, \eta_{TR}\, \eta_{MT}\right) \qquad (13)$$

$$\mathrm{COP} = Q_b / E_{MT} \,. \qquad (14)$$

Above equations were referenced in the data processing of the experiments mentioned below.

3. PROTOTYPE

The prototype heat pump dehydrator shown in Fig. 3 was designed assimilating the findings of the previous experiments. The main specifications are the following.

Dehydrating Vessel

Type: horizontal cylinder, Inner diameter: 1,000 mm, Length: 1,800 mm, Inner volume: 1.4 m^3, Material: SUS304, Insulation: 50 mm-thick glass wool.

Steam Compressor

Type: Oil-free wing, Cylinder volume: 0.007 m^3, Number of revolutions: 850 rpm, Motor: 7.5 kW, Heat resistance: 200°C.

Agitator-Condenser

Type: concentric 4-deep helical coils, Outer diameter of tube: 34 mm, Material: SUS304, Surface area: 9.6 m^2, Number of revolutions: 10 rpm.

Preheat Boiler

Type: submerged combustion tube, Fuel: propane, Thermal output: 116.3 kW.

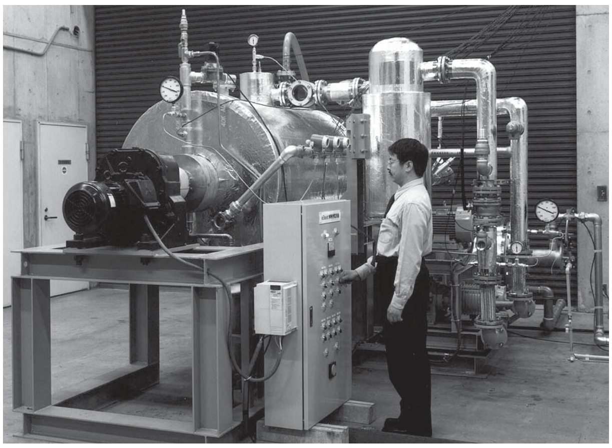

Fig. 3 View of Prototype Dehydrator

Decreasing the temperature difference between condensing steam and drying material is essential to attain high COP as predicted by Eq. 1. It is also important to prevent overheating during a compression process. In conventional steam dryers, this temperature difference may be around 60 K. In the case of the steam heat pumps, less than 20 K may be preferable. One approach taken was to enlarge surface area for heat exchange by using the coil-type condenser instead of the double-wall one. Another approach was to improve the heat-transfer coefficients on the drying material side by using the agitator-condenser.

Schematic diagram of the prototype dehydrator is shown in Fig. 4. Steam for preheating was generated below 100°C in the boiler maintained under vacuum conditions, as a trial of waste heat utilization. The boiler was turned off during the heat pump dehydration processes. The cyclone was used to superheat the suction steam of the compressor. The heat source was the compressor discharge steam during the early stage of testing as explained in Fig. 2 It was later altered to the exiting steam of the condenser to enhance air purge capability as depicted in Fig. 4.

Temperatures, pressures, electric power inputs, and a steam mass flow rate were measured every 20 seconds and recorded every minute. Drainage was weighed manually because accuracy of the ultrasonic steam mass-flowmeter was found inadequate.

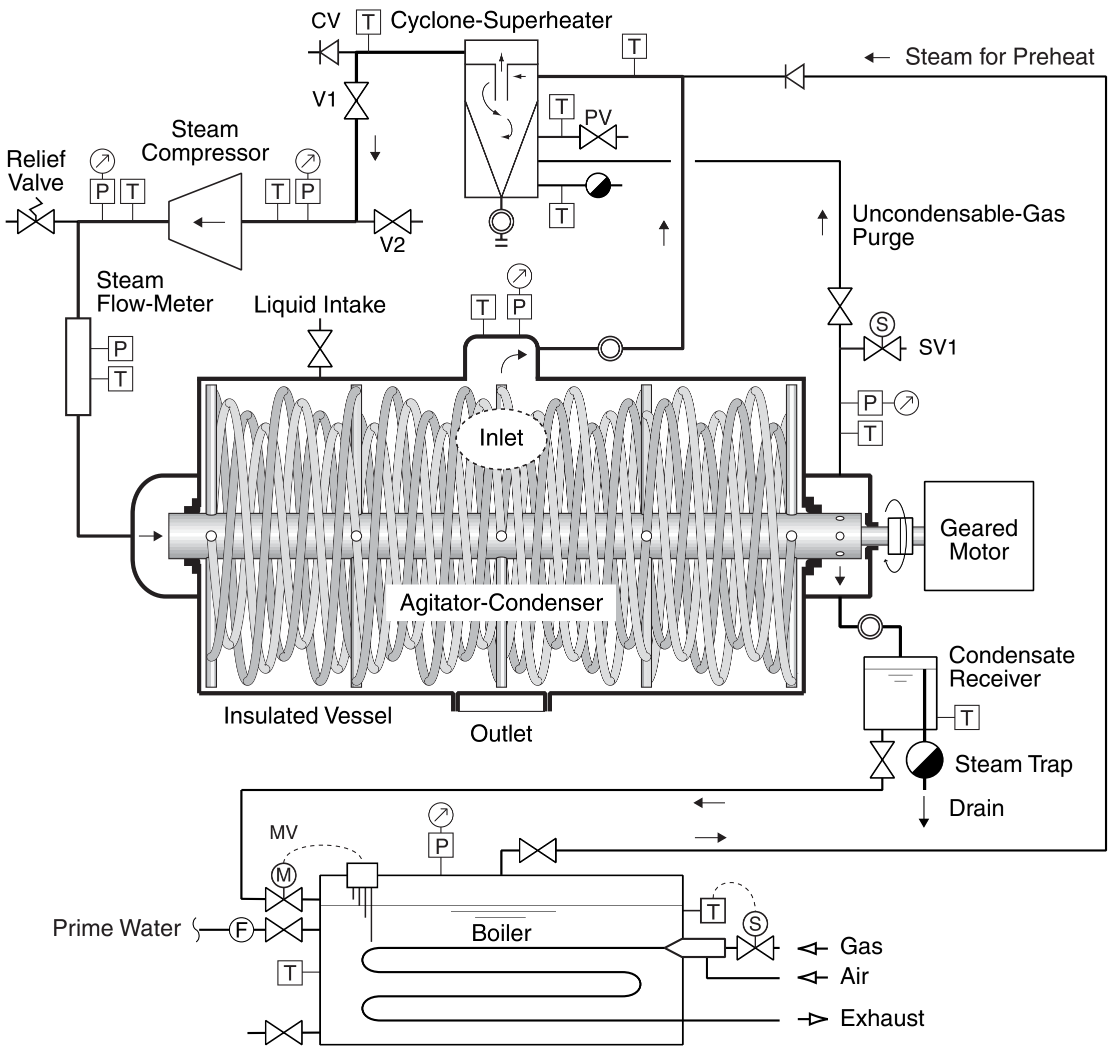

Fig. 4 Diagram of Prototype

4. RESULT AND DISCUSSION

4.1 Heat Pump Characteristic

The performance characteristics were measured by dehydrating watery bentonite. The drain rates versus boiling temperature are plotted in Fig. 6, that increased with rise in temperatures. A predicted evaporation rate of 100 kilograms per hour at 90°C was confirmed. The dependency of vaporization rates on the boiling temperatures can be explained mainly by the changes in the specific volume of suction steam that appears in Eq. 9.

Fig. 5 Drain of Condensate

Coefficients of performance defined by Eq. 14 versus temperature lift (condensing temp. minus boiling temp.) are plotted in Fig. 7. This correlation is suggested by Eq. 1, based on the premise that changes in boiling temperatures were relatively small. Higher COPs than 10 were attained when the temperature lifts were less than 18 K. The ratios of measured COP to the theoretical one were ranging from 0.49 to 0.57.

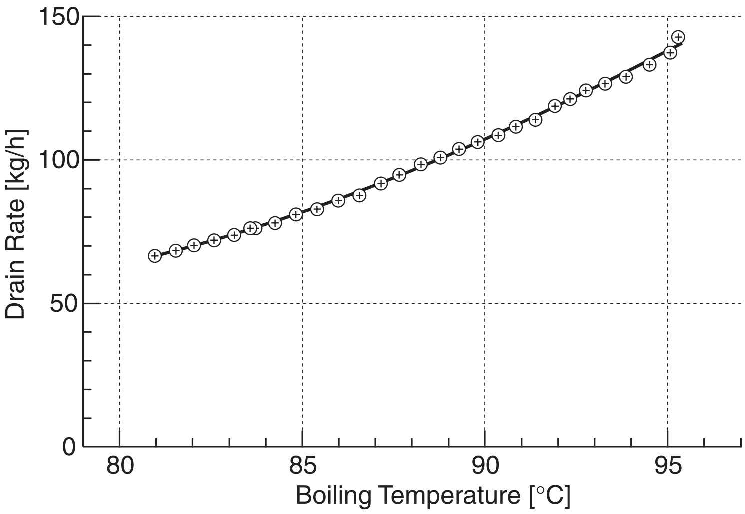

Fig. 6 Drain Rate vs. Boiling Temperature

Volumetric and adiabatic-compression efficiencies of the steam compressor were plotted against compression ratio defined by Eq. 5. These efficiencies are formulated in Equations 9 and 11 respectively. The approximate expressions in Fig. 8 will be used in simulations.

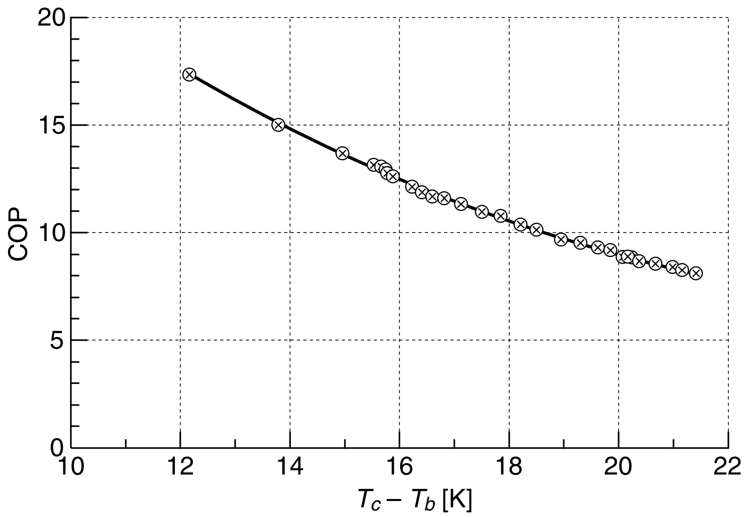

Fig. 7 Coefficient of Performance

4.2 Dehydration Test

In addition to the watery bentonite, malt leavings, wet sawdust, lees of shochu, and wet soil were tried. Among them, the test results on the lees of shochu of which Initial water content was 90.0% are mentioned. The

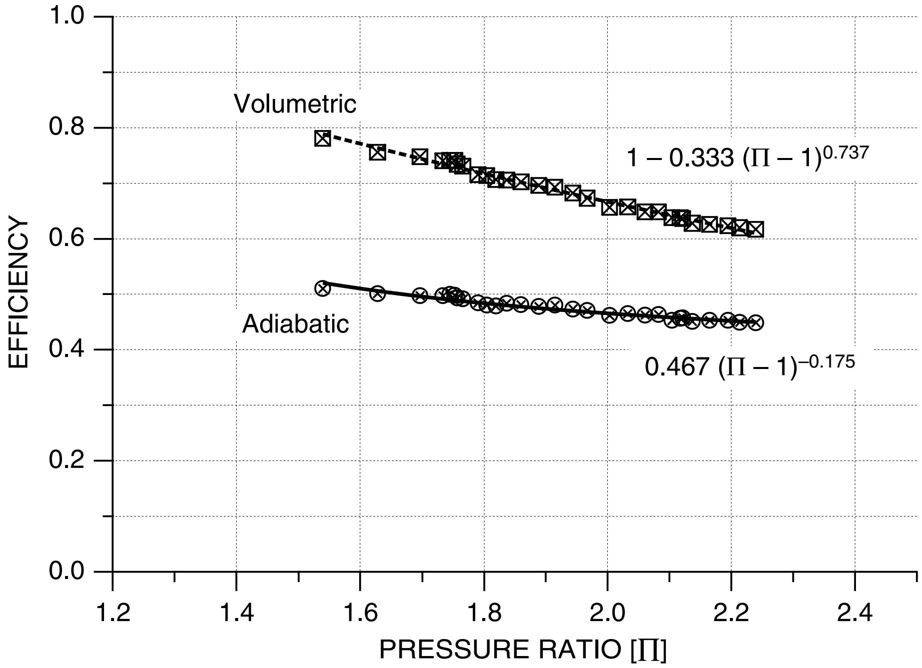

Fig. 8 Compressor Efficiencies vs. Compression Ratio

water content is the mass ratio of water to the whole while the water ratio is defined as the mass ratio of water to the dried mass and their relationship is expressed in Eq. 15. The water ratio is more suitable to formulate a dehydration process as expressed in Eq. 16.

$$w_d = w_w/(1 - w_w) \quad \text{.......................................} \quad (15)$$

$$G = M_d(-dw_d/dt) \quad \text{.......................................} \quad (16)$$

Lees whose water ratio of 9.00 and mass of 502 kg were preheated to 97°C for one hour using steam from a boiler. The boiler was stopped and the dehydration process followed for 320 minutes. The water ratios over time are plotted in Fig. 9. The dried lees resembled dried fruits and adhered firmly to the condenser surface. The measured water ratio was 0.13 and its mass was calculated to be 6.7 kg.

The condensate was analyzed and the results were, pH: 3.8 @ 19.6°C, BOD: 7,000 mg/litre, *n*-hexane: 5 mg/litre, SS: 10 mg/litre, T-N: 4.87 mg/litre, and T-P: 0.015 mg/litre. The BOD and acidity can be explained by acetic acid. The suspended solids may be rust from a steam trap made of cast-iron.

4.3 Energy-Efficiency

Compressor-only COPs were up to 14 in the beginning of the dehydration process when boiling temperatures were high. The COPs gradually fell down to 4 in the end of the process when the boiling temperatures dropped below 80°C: thermal insulation was apparently insufficient. Non-trivial COP deterioration was caused by uncondensable gas, i.e. air that remained in the apparatus

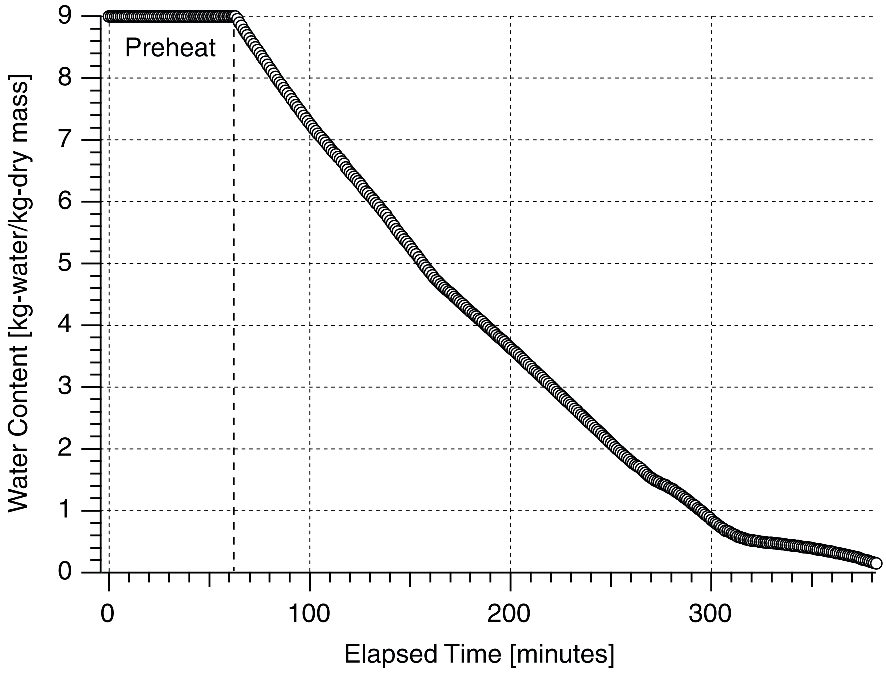

Fig. 9　Change of Water Content over Time

and dissolved in the waste. This problem was alleviated later by improving a purge mechanism.

Total condensate drained was 445 kg and its latent heat of vaporization was calculated to be 1.024 GJ or 284 kWh. Electricity inputs during the dehydration process were integrated to be 39.3 kWh by the compressor and 2.1 kWh by the agitator-motor. Therefore, a summation-based compressor-only COP was 7.23 and a total COP that includes agitator power was 6.9.

Representative dryers are compared from the viewpoint of energy in Table 1. Conductive dryers heated by steam boilers burn heavy-oil. Thermal efficiencies of a steam boiler and a conductive dryer are assumed to be 80% and 75% respectively and their multiplication makes overall

Table 1　Comparison of Dryer Systems
(1,000kg of water vaporization basis, preheat excluded)

Dryer System	Steam Conduction	Hot-Air	Microwave	Steam Heat Pump
Energy	heavy oil	natural gas	electricity	electricity
Thermal Efficiency	60%	50%	40%	COP = 7
Primary Energy Consumption [MJ]	3,769	4,514	16,065	928
CO_2 Emission [kg-C]	67.8	63.2	169.3	9.7

efficiency of 60%. Hot-air dryers can use combustion gas directly but considerable amount of sensible heat is lost by the exhaust. Microwave dryers are dispensable of heat exchangers and 80% of the electromagnetic energy can be absorbed by water, but energy efficiencies of magnetrons are more or less 50%. In the case of steam heat pump dryers, COP of 7 was assumed based on the test results, although further improvements are possible.

In Table 1, the figures of primary energy consumptions and carbon dioxide emissions are calculated on the premise to vaporize 1,000 kg-water. As for the electricity, generation/transmission efficiency of 35.1% and 0.108 kg-C/kWh were assumed. Advantages of the steam heat pump dehydrator are obvious.

5. CONCLUSIONS

A concept of the energy-efficient evaporative dehydrator employing a steam heat pump cycle was explained and its mathematical expressions were presented. A prototype dehydrator with a 7.5 kW-compressor and an agitator-condenser was made and its performance characteristics are investigated . The vaporization rate was approximately 100 kg per hour at 90°C boiling and the COP was around 10 at temperature lift of 18 K.

Lees of shochu was dehydrated from the initial water ratio 9 to 0.13, thus its drying capability was confirmed. Sum of the latent heat of vaporization divided by the integrated electricity consumption was around 7. Using this figure, primary-energy consumptions and carbon-dioxide emissions were compared with the conventional dryers. Both of them can be reduced greatly.

The author believes that this technology would have wide applications and should bring a breakthrough in watery-waste treatments in both material and thermal recyclings. Commercialization efforts may continue.

ACKNOWLEDGMENTS

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NOMENCLATURE

c_p	= specific heat of water	[J/(kg•K)]
E_{MT}	= motor input	[J/s]
G	= mass-flow rate of steam, vaporization or condensation rate	[kg/s]
h_a	= enthalpy of water at ambient temperature	[J/kg]
h_{dis}	= enthalpy of compressor discharge steam	[J/kg]
h_{sl}	= enthalpy of saturated water	[J/kg]
h_{suc}	= enthalpy of compressor suction steam	[J/kg]
h_{sv}	= enthalpy of saturated steam	[J/kg]

M_d	= mass of dried material	[kg]
n	= revolutions per second	[s^{-1}]
p_b	= boiling pressure	[Pa]
p_c	= condensing pressure	[Pa]
p_{dis}	= discharge pressure	[Pa]
p_{suc}	= suction pressure	[Pa]
q_L	= latent heat of water vaporization	[J/kg]
q_S	= sensible heat of water	[J/kg]
Q_b	= heat for water boiling	[W]
Q_{CL}	= compressor cooling loss	[W]
T_a	= ambient temperature	[K]
T_b	= boiling temperature	[K]
T_{suc}	= suction steam temperature	[K]
v_b	= specific volume of saturated steam at boiling temperature	[m^3/kg]
v_{suc}	= specific volume of suction steam	[m^3/kg]
V_{CY}	= cylinder volume	[m^3]
V_{PD}	= piston displacement	[m^3/s]
w_d	= water ratio	[kg-water/kg-dry mass]
w_w	= water content	[kg-water/kg-wet mass]
W_{ad}	= adiabatic compression work	[J/kg]
Δp_{DL}	= pressure drop in discharge line	[Pa]
Δp_{SL}	= pressure drop in suction line	[Pa]
η_{ab}	= adiabatic compression efficiency	
η_{MR}	= motor efficiency	
η_{TR}	= power transfer efficiency	
η_v	= volumetric efficiency	
κ	= specific heat ratio	
Π	= compression ratio	

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