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Research Article

Performance Analysis of Organic Rankine-vapor Compression Ice Maker Utilizing Food Industry Waste Heat

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Abstract: To develop the organic Rankine-vapor compression ice maker driven by food industry exhaust gases and engine cooling water, an organic Rankine-vapor compression cycle system was employed for ice making and a thermodynamic model was developed and the effects of working fluid types, hot water temperature and condensation temperature on the system performance were analyzed and the ice making capacity from unit mass hot water and unit power waste heat were evaluated. The calculated results show that the working fluid type and the temperatures of heat source and condensation have important effects on the system performance. The system can achieve optimal performance when use R245fa as power and refrigeration medium. The ice quantity generated from per ton hot water is 86.42 kg and the ice-making rate for per kW waste heat is 2.27 kg/h, when the temperatures of hot water and condensation are respectively 100 and 40°C. A conclusion can be draw by the calculation and analysis that using organic Rankine-vapor compression system for ice making from food industry waste heat is feasible.

Keywords: Expander, food industry waste heat, ice maker, organic rankine cycle, vapor compression cycle

INTRODUCTION

Modern large food industry is generally equipped with mechanical refrigeration, but all of them use fuel oil or electricity to achieve refrigeration. While for middle type and mini-type food industry with power in the range of 136-441 kW, most of them cannot carry compressor-ice maker because of their small horsepower of diesel engine (Wang and Wang, 2005; Wang *et al*., 2004a). For the sake of food preservation, these middle type and mini-type food industry have to carry a lot of ice, which costs much. At the same time, waste heat discharged from the hot exhaust gases with temperature of 400°C or so in most of the food industry is rejected to the atmosphere, which accounts for 30-35% of the energy of a diesel engine. Besides, waste heat discharged from the engine cooling water with temperature of 60-100°C accounts for about 30% of the burning energy in a diesel engine, which is also rejected to the atmosphere (Wang *et al*., 2006a, 2008). So, making use of these waste heats for ice making for food preservation is of great significance.

At present, some effort has been devoted to the utilization of the vast amount of the waste energy for refrigeration. The heat-operated absorption/adsorption systems can utilize waste heat for refrigeration. Srikhirin *et al*. (2001) described a number of research options of absorption refrigeration technology and provided a comparison of the various types of absorption refrigeration systems. Meunier (1998) claimed solid sorption is very effective for low grade cooling, not only for air conditioning but also for deep freezing. Wang and his team (Wang *et al*., 2004b, 2005; 2006b, c; Lu *et al*., 2007; Wang and Wu, 2005) had devoted great efforts to research the adsorption ice maker utilizing waste heat from the food industry and had made a lot of achievements. Although the adsorption ice maker system has the advantages of simple control, low initial investment and circulating expenditure and less noise, the coefficient of performance is generally low. Because of sustainable development requirements, efficient refrigerators driven by low grade thermal energy from different sources have received much more attention in recent years. Currently, the use of thermal energy to operate an Organic Rankine Cycle-Vapor Compression Cycle (ORC/VCC) for refrigeration has become the subject of renewed interest and has been reported by several investigators. The ORC/VCC system converts waste heat into a cooling effect, which is accomplished at the

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site of the installation by using the organic Rankine cycle to generate the shaft work required to drive a vapor compression cycle. Aphornratana and Sriveerakul (2010) theoretically analyzed a combined Rankine vapor-compression refrigeration cycle powered by low grade thermal energy. Wang *et al*. (2011a, b) introduced a novel thermally activated cooling concept- a combined cycle couples an ORC and a VCC and developed a prototype with nominal cooling capacity of 5 kW. Demierre *et al*. (2012) presented the development of an ORC-ORC prototype with HFC-134a as working fluid and heating power about 20 kW at the condenser. However, few people studied ice making using ORC/VCC, especially ice making utilizing waste heat on food industry. The biggest difference between ice making and air conditioning is that the evaporation temperature for ice making is about -5°C, which is lower than that for air conditioning, leading to low efficiency and high pressure ratio for compressor used in ice maker.

In this study, to utilize the vast amount of waste energy discharged from the exhaust gases and from the engine cooling water in the food industry for ice making, the ORC/VCC system is employed and the thermodynamic model is developed and the effects of working fluid types, generation temperature, condensation temperature and evaporation temperature on the system performance are analyzed in order to demonstrate the feasibility of such a system for ice making.

METHODOLOGY

System design: The system of ORC/VCC for ice making in food industry mainly consists of engine cooling water outlet, engine cooling water inlet, engine exhaust gases outlet, exhaust port, waste heat boiler, hot water pump, generator, expander, compressor, ice maker, throttle valve, condenser, cooling water pump and working fluid pump, as shown in Fig. 1. The working principle of this system is as follows: waste heat collected from the engine exhaust gases and cooling water provides energy to heat and vaporize a working fluid with low boiling point. Energy is extracted from this vapor in an expansion engine that is used to drive a vapor compressor for ice making. Afterwards, the fluid exiting the expander is condensed and pumped back to generator where it is again vaporized. R134a, R245fa and R600a are, respectively selected as the working fluid to compare their cycle efficiency and to select the suitable working fluids for ice making. To prevent fouling, the exhaust gases do not enter generator directly and the waste heat boiler is used to recover heat from the exhaust gases. Water is used as the working medium for waste heat boiler and is then pumped to generator by hot water pump to heat the working fluid. Due to the temperature of exhaust gases about 400°C, the temperature of hot water from

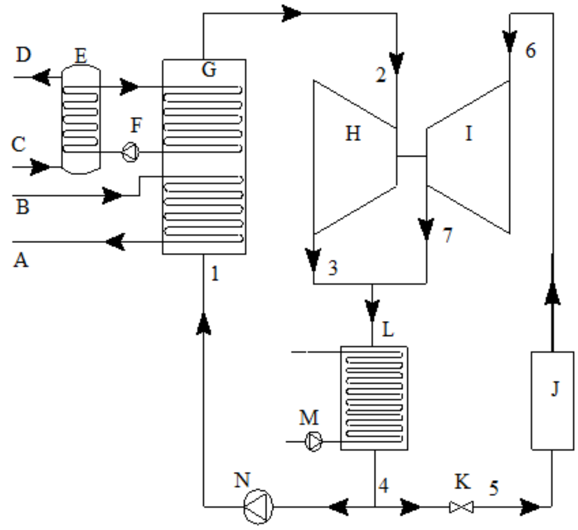

Fig. 1: System schematic diagram

A: Engine cooling water outlet; B: Engine cooling water inlet; C: Engine exhaust gases outlet; D: Exhaust port; E: Heat recovery boiler; F: Hot water pump; G: Generator; H: Expander; I: Compressor; J: Ice maker; K: Throttle valve; L: Condenser; M: Cooling water pump; N: Working fluid pump

waste heat boiler is generally higher than that of engine cooling water. A vertical generator is used and it is divided into two sections. The hot water from waste heat boiler flows in the upper section of generator and the engine cooling water flows in the lower section of generator, as shown in Fig. 1. Countercurrent is applied in generator to improve the system efficiency. That is, the hot water flows in generator from top to bottom and the working fluid flows from bottom to top in generator. Seawater is an ideal cooling medium for the ORC/VCC system, which makes the system more compact and more powerful comparing with the system cooled by air. The programmable logic controller, frequency converter and liquid level sensor are suggested to be adopted in the system to automatically control the liquid level of working fluid in generator and ensure the high heat exchange efficiency in generator. Waste heat from the food industry is usually instable because the engine often works under low load, at idle speed and even is sometimes turned off. Thus, the instable heat source will seriously impact the performance of the system. It is an unavoidable problem when such a system is put into practice. To adapt the instability of heat source, the radial and axial flow expander is employed. To improve the drive efficiency, the compressor and the expander are directly coupled on the same shaft without gear and coupling.

Thermodynamic models: To develop the thermodynamic models, an assumption is made that friction and heat losses in the ORC/VCC are negligible.

For ORC:

$$W_{\exp} = m_p \left(h_2 - h_{3s}\right)\eta_{\exp} \tag{1}$$

$$W_{pump,w} = m_p \frac{\left(h_{1s} - h_4\right)}{\eta_{pump,w}} \tag{2}$$

$$Q_{boi} = m_p \left(h_2 - h_1\right) \tag{3}$$

$$W_{net} = W_{\exp} - W_{pump,w} - W_{pump,h} - W_{pump,c} \tag{4}$$

$$\eta_p = \frac{W_{net}}{Q_{boi}} \tag{5}$$

$$T_{boi} = \sqrt{T_h T_c} \tag{6}$$

$$W_{pump,h} = m_h \frac{\Delta p_h v_h}{\eta_{pump,h}} \tag{7}$$

$$W_{pump,c} = m_c \frac{\Delta p_c v_c}{\eta_{pump,c}} \tag{8}$$

For VCC:

$$Q_{eva} = m_{\mathrm{ic}} \left(h_6 - h_5\right) \tag{9}$$

$$W_{com} = m_{\mathrm{ic}} \frac{\left(h_{7s} - h_6\right)}{\eta_{com}} \tag{10}$$

$$W_{com} = W_{\exp} \tag{11}$$

$$COP_c = \frac{Q_{eva}}{W_{com}} \tag{12}$$

$$m_{ice} = \frac{Q_{eva}}{h_{ice}} \tag{13}$$

The overall performance of ORC/VCC is defined as:

$$COP_s = \eta_p COP_c \tag{14}$$

$$n = \frac{m_{ice}}{m_h} \times 1000 \tag{15}$$

$$nr_{ice} = \frac{m_{ice}}{Q_{boi}} \times 3600 \tag{16}$$

RESULTS AND DISCUSSION

Considering the instability of hot water from food industry and the temperature variation of seawater in different seasons and places, the temperature of hot water at generator inlet, T_h, is in the range of 80-160°C and the condensation temperature is 20-50°C. For the sake of simplification, the evaporation temperature is -5°C and keeps invariable and the power of generator heated by exhaust gases and engine cooling water is 200 kW. The isentropic efficiencies for expander, compressor, working fluid pump, hot water pump and cooling water pump are shown in Table 1.

Effect of hot water temperature on system performance: The temperatures of exhaust gases and engine cooling water from food industry vary according to the operating conditions of engine, leading to the temperature variation of hot water at generator inlet. As a result, the generation temperature, T_{boi}, varies with T_h. Figure 2 and 3 show the system performance including Q_{eva}, n and m_{ice} as a function of hot water temperature. In Fig. 2 and 3 and Table 2, the condensation temperature and evaporation temperature are respectively 40 and -5°C and Q_{boi} is equal to 200 kW and the working fluid for ORC is R245fa and for VCC are respectively R245fa, R134a and R600a. The calculated results show that W_{exp} and η_p increase with the increasing T_h. This is due to the fact that the pressure, temperature and enthalpy at expander inlet increase with the increase of T_h, resulting in the increase of enthalpy drop between expander inlet and outlet and the increase of expander power when the condensation temperature keeps invariable. W_{exp} is 10.36, 18.11 and 23.96 kW and η_p is 5, 8.72 and 11.45% when T_h is, respectively 80, 120 and 160°C. W_{com}, Q_{eva} and m_{ice} also increase along with the increasing W_{exp}. It is obvious from Fig. 2 and 3 that m_{ice} and Q_{eva} depend largely on T_h and they increase with the increase of T_h. Q_{eva} is respectively 37.62, 32.18 and 35.44 kW and m_{ice} is respectively 323.5, 276.72 and 304.82 kg/h for the working fluid of R245fa, R134a and R600a at T_h = 80°C. When T_h increases to 160°C, Q_{eva} increases to 87.04, 74.45 and 82.01 kW and m_{ice} increases to 748.53, 640.29 and 705.32 kg/h for the working fluid of R245fa, R134a and R600a, respectively. It is thus clear that there is a great difference in the ice making capability for heat source with different temperature. There are two main parameters for evaluating the ice making capability of

Table 1: Parameter values

Parameter	$\eta_{\exp}$	$\eta_{pump,w}$	$\eta_{pump,h}$	$\eta_{pump,c}$	η_{com}	h_{ice} (kJ/kg)
Value	0.85	0.9	0.9	0.9	0.8	418.6

Table 2: Relationship between T_h and nr_{ice}

	T_h (°C)	80	100	120	140	160
nr_{ice} /kg/kW/h	R245fa	1.62	2.27	2.83	3.32	3.74
	R134a	1.38	1.94	2.42	2.84	3.20
	R600a	1.52	2.13	2.67	3.13	3.53

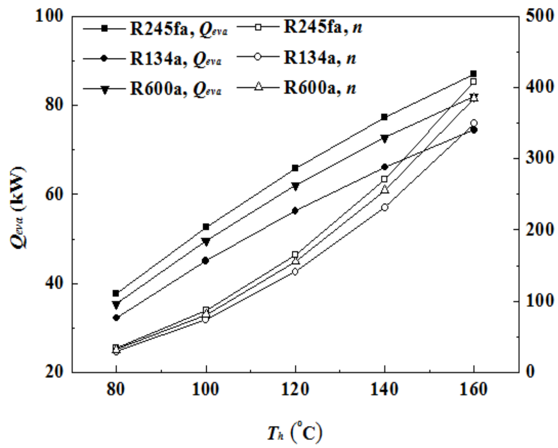

Fig. 2: Effects of heat source temperature on Q_{eva} and n

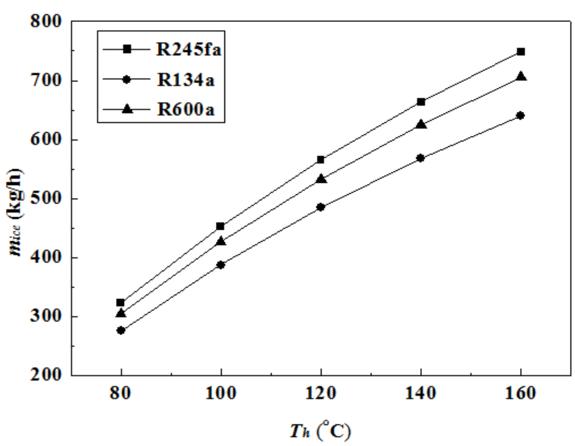

Fig. 3: Effect of heat source temperature on m_{ice}

hot water, one is the ice yield per ton hot water and the other is the production rate of ice per unit power heat source and they are respectively represented by symbols of *n* and nr_{ice}. In Fig. 2, n is 34.00, 29.08 and 32.04 kg/t at T_h = 80°C and it increases to 408.09, 349.07 and 384.53 kg/t at T_h = 160°C for the working fluid of R245fa, R134a and R600a respectively. For the working fluid of R245fa, R134a and R600a, nr_{ice} is 1.62, 1.38 and 1.52 kg/kW/h at T_h = 80°C and it increases to 3.74, 3.20 and 3.53 at T_h = 160°C respectively, as shown in Table 2. The above analyses show that T_h has an important influence on the system performance and the higher the T_h, the greater the nr_{ice} and *n*. Thus the actual demand of ice for food preservation and payback period should be comprehensively considered so as to decide the rated ice making capability during practical design owing to the invariability of waste heat from food industry.

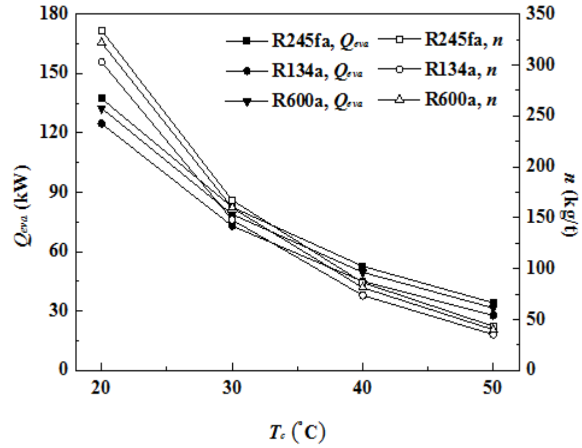

Fig. 4: Effects of condensation temperature on Q_{eva} and n

For a food industry with waste heat power of 200 kW and R245fa as working fluids for ORC and VCC, nr_{ice} equals to 1.62 and 3.74 kg/ (kW.h) at T_h = 80 and 160°C respectively according to Table 2. That is, the ice making rate, m_{ice}, is respectively 324 and 748 kg/h at T_h = 80 and 160°C, which can completely meet the demand of ice for food preservation. To simplify the calculation, Q_{boi} is equal to 200 kW and keeps invariable in this study. However, Q_{boi} is always variable along with the engine power output in practice and may be greater than or less than 200 kW. If T_h and Q_{boi} are 80°C and 100 kW, m_{ice} is 162 kg/h according to Table 2. m_{ice} increases to 1122 kg/h if T_h and Q_{boi} increase to 160°C and 300 kW respectively.

In Wang *et al*. (2003) article, an adsorption ice maker using waste heat from food industry is researched and activated carbon-methanol is used as the adsorption working pair. The experimental results show that m_{ice} and nr_{ice} are respectively 15.3 kg/h and 0.45 kg/ (kW.h) when the power and temperature of waste heat from food industry are 34 kW and 120°C. In Ni *et al*. (2011) article, an absorption ice maker with ammonia water as working medium is studied on food industry, m_{ice} and nr_{ice} are respectively 93.1 kg/h and 2.48 kg/ (kW.h) when the exhaust gases temperature and condensation temperature are 400 and 35°C. When the hot water temperature and condensation temperature are 120 and 40°C and R245fa is used as the working fluid for ORC and VCC, nr_{ice} equals to 2.83 kg/ (kW.h) in this study, which is greater than that in Wang *et al*. (2003) and Ni *et al*. (2011) articles.

Effect of condensation temperature on system performance: The condensation temperature varies with ambient and the effects of T_c on Q_{eva} and *n* are illustrated in Fig. 4. In Fig. 4, the hot water temperature and evaporation temperature are 100 and -5°C and the working fluid for ORC is R245fa and for VCC are, respectively R245fa, R134a and R600a and the condensation temperature is 20, 30, 40 and 50°C, respectively. The calculated results show that the

Table 3: Comparison of ice making performance for different medium

Working fluid for ORC	Working fluid for VCC	T_h (°C)	η_p (%)	COP_c	COP_s (%)	n (kg/t)	m_{ice} (kg/h)	nr_{ice} (kg/(kW.h))
R245fa	R245fa	80	5.00	3.63	18.17	34.00	323.50	1.62
R245fa	R134a	80	5.00	3.11	15.56	29.08	276.72	1.38
R245fa	R600a	80	5.00	3.42	17.11	32.04	304.82	1.52
R245fa	R245fa	160	11.45	3.63	41.60	408.09	748.53	3.74
R245fa	R134a	160	11.47	3.11	35.64	349.07	640.29	3.20
R245fa	R600a	160	11.44	3.42	39.14	384.53	705.32	3.53

pressure and enthalpy at expander outlet increase with the increasing T_c, leading to the decrease of enthalpy drop between expander inlet and outlet and the decrease of expander power. The increasing T_c also causes the decrease of COP_c and COP_s. Observing the profiles from Fig. 4 it is obvious that Q_{eva} and n depend largely on T_c and they decrease with the increase of T_c. For the working fluid of R245fa, R134a and R600a for VCC, Q_{eva} is respectively 137.38, 124.65 and 132.45 kW at T_c = 20°C and it decreases to 34.22, 27.89 and 31.63 kW respectively at T_c = 50°C. The temperature of 20°C is a representative condensation temperature at winter in some places and the temperature of 50°C is a representative condensation temperature at summer in some places. Taking R600a as working fluid for VCC as an example, Q_{eva} is 132.45 kW at winter and it is only 31.63 kW at summer, indicating that there is a great difference of ice making capability between summer and winter. Thus, the actual demand of ice for food preservation, waste heat condition and the temperature in winter and summer should be comprehensively considered in order to reduce the system investment when design an ice maker utilizing waste heat from food industry.

Effect of working fluid types on system performance: As evident in Fig. 2 to 4 and Table 2 that the VCC system with R245fa as the working fluid has the optimal system performance compared with other two working fluids. Table 3 shows the effect of working fluid types on the system performance. In Table 3, the condensation temperature and evaporation temperature are 40 and -5°C and Q_{boi} equals to 200 kW. The ORC/VCC system with working fluid of R245fa has the maximum of COP_c, COP_s, n, m_{ice} and nr_{ice} compared with two other working fluids. Known from the above analysis, the ORC/VCC system can achieve optimal performance when uses R245fa as ORC and VCC working fluid. For the practical and applied system, if the expander and compressor adopt different working fluids, which puts strict demands on the main shaft seal and gas separating.

To sum up, using organic Rankine cycle-vapor compression cycle for ice making utilizing waste heat from food industry is feasible and the keys are to develop the expander and compressor with high efficiency, especially the compressor with high pressure ratio.

CONCLUSION

To efficiently utilize waste heat from food industry exhaust gases and engine cooling water for ice making, the organic Rankine-vapor compression cycle system was employed and the effects of working fluid types, hot water temperature and condensation temperature on the system performance were analyzed by developing the thermodynamic models, the following conclusions can be drawn:

- The ice yield per ton hot water and production rate of ice per unit power waste heat are 86.42 kg/t and 2.27 kg/ (kW.h) at hot water temperature of 100°C and condensation temperature of 40°C with R245fa as the working fluid, which demonstrates the feasibility of using organic Rankine-vapor compression system for ice making from food industry waste heat.
- The hot water temperature and condensation temperature have important influences on the system performance. The ice yield per ton hot water is 86.42 kg/t at hot water temperature of 100°C and condensation temperature of 40°C, while it increases to 333.96 kg/t at condensation temperature of 20°C. Thus, the actual demand of ice for food preservation, waste heat condition and the temperature in winter and summer should be comprehensively considered in order to reduce the system investment when design an ice maker utilizing waste heat from food industry.

NOMENCLATURE

COP_c : Coefficient of Performance for VCC
COP_s : Overall Coefficient of performance for ORC/VCC
h_1 : Enthalpy at working fluid pump outlet, kJ/kg
h_{1s} : Enthalpy at working fluid pump outlet based on isentropic process, kJ/kg
h_2 : Enthalpy at expander inlet, kJ/kg
h_{3s} : Enthalpy at expander outlet based on isentropic process, kJ/kg
h_4 : Enthalpy at working fluid pump inlet, kJ/kg
h_5 : Enthalpy at evaporator inlet, kJ/kg
h_6 : Enthalpy at evaporator outlet, kJ/kg
h_{7s} : Enthalpy at compressor outlet based on isentropic process, kJ/kg
h_{ice} : Phase change latent heat of changing water into ice
m_c : Mass flow rate for cooling water, kg/sec

m_h : Mass flow rate for hot water, kg/sec
m_{ic} : Mass flow rate of working fluid for VCC, kg/sec
m_{ice} : Ice making rate, kg/sec
m_p : Mass flow rate of working fluid for ORC, kg/sec
n : Ice yields per ton hot water, kg/t
nr_{ice} : Production rate of ice per unit power heat source, kg/ (kW.h)
Q_{boi} : Generator heat input, kW
Q_{eva} : Evaporator power for ice maker, kW
T_{boi} : Generation temperature in generator, °C
T_c : Condensation temperature, °C
T_h : Hot water temperature at generator inlet, °C
v_c : Specific volume of cooling water, m^3/kg
v_h : Specific volume of hot water, m^3/kg
W_{com} : Compressor work input, kW
W_{exp} : Expander work output, kW
$W_{pump,c}$: Cooling water pump power consumption, kW
$W_{pump,h}$: Hot water pump power consumption, kW
$W_{pump,w}$: Working fluid pump power consumption, kW
W_{net} : Net work output for ORC, kW
η_{com} : Compressor isentropic efficiency
η_{exp} : Expander isentropic efficiency
η_p : Organic Rankine cycle efficiency
$\eta_{pump,c}$: Cooling water pump isentropic efficiency
$\eta_{pump,h}$: Hot water pump isentropic efficiency
$\eta_{pump,w}$: Working fluid pump isentropic efficiency
ΔP_c : Pressure difference between cooling water pump inlet and outlet, kPa
ΔP_h : Pressure difference between hot water pump inlet and outlet, kPa

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