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ENERGY AND EXERGY ANALYSIS OF A WASTE HEAT DRIVEN CYCLE FOR TRIPLE EFFECT REFRIGERATION

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ABSTRACT

In this paper waste heat driven triple effect refrigeration cycle is analyzed from the viewpoint of both energy and exergy concept of thermodynamics. In this cycle ejector organic Rankine cycle, absorption refrigeration cycle and cascade vapour compression refrigeration cycles are integrated in order to obtain a range of temperature for varied simultaneous use. Exergy analysis determines the destruction and losses of exergy in various components and hence in overall system. Exergy efficiencies provide measure of approach to ideality while exergy destruction and losses provide measure of the deviation from ideality. Energy efficiency is found to be around 21.79% while exergy efficiency is 12.91%. The maximum thermodynamic irreversibility occurs in heat recover vapour generator followed by ejector and condenser of combined ejector organic Rankine refrigeration cycle.

INTRODUCTION

Energy plays a significant role in the development of a country. Industries require huge amount of energy which is being presently supplied by fossil fuels. Rapid consumption of primary energy sources like fossil fuels to meet the burgeoning demand of energy is not only causing their fast depletion but also causing global environmental degradation. At the same time, a large amount of low grade waste thermal energy is released in the biosphere by the industries, which further causes ecological imbalance. Up to 70% of energy input in the manufacturing sector is lost in Canada (Galanis et al., 2009). In the US, industrial waste heat amounts to about 20–50% (Inc. BCS. 2008). In the UK heat losses in the industrial sector could represent few tens of TWh (Retting et al., 2011). Therefore, utilization of waste heat for power generation, refrigeration and air conditioning, heating etc. will be beneficial both from the view point of energy as well as environment. Waste heat sources are grouped in three categories based on temperature level (Tchanche BF et al., 2011): low (<230 °C), medium (230–650 °C) and high (>650 °C)

There are many industrial processes where heat is available as low temperature waste heat. Similarly, there are numerous applications which require cooling at different temperatures and at different rates. Thus, there exist an opportunity to utilize the otherwise waste heat to obtain cooling by integration of various cooling technologies.

It is a well proven that absorption refrigeration cycle (ARC) can be operated easily by waste heat. The required temperature of the heat source varies from 60 °C to 200 °C, depending on its configuration. The required generator temperature for single effect cycle is 80-100 °C (Kaushik et al., 2009), double effect cycle 120-150 °C (Arora et al., 2016 a), and half effect 60-75 °C (Arora et al., 2016 b). Triple effect cycles require generator temperature even higher than 150 °C. The Organic Rankine cycle (ORC) are capable of converting low quality thermal energy into high grade mechanical work or electrical power. Heat source with temperature up to 230 °C can operate an ORC (Tchanche BF et al., 2011). Ejector refrigeration cycle (ERC) can also be operated by a waste heat having temperature up to 65 °C.

Many efforts have been made by researchers to combine various cycles for efficient utilization of waste heat for different purposes. Sun et al. (1996) proposed a combined ejector absorption cycle for refrigeration and air conditioning in order to obtain high coefficient of performance (COP). Hong et al. (2011) proposed the combination of double effect ARC and ejector. The COP of the combined cycle was reported to be 30% higher than that of conventional single effect ARC. Wang et al. (2009) theoretically studied a novel combined power and ejector refrigeration cycle and concluded that heat recovery vapour generator (HRVG) accounts for highest exergy destruction. Dai et al. (2009) combined the ejector refrigeration cycle with the Rankine cycle and found that the exergy loss was maximum in heat addition process, followed by the ejector.

Zheng and Weng (2010) proposed a combined organic Rankine and ejector refrigeration cycle. The turbine of Rankine cycle produced power and its exhaust was used to drive the ejector cycle for refrigeration. Ejector suffered highest irreversibility. Habibzadeh et al. (2013) studied the combined organic Rankine cycle and the ejector refrigeration cycle with various substances like R123, R254fa, R141b, R601a and R600a. For power to refrigeration ratio of 10, R601a showed the highest thermal efficiency and the lowest exergy losses. Many other researchers (Goswami and Xu, 1999; Hasan et al., 2002; Martin and Goswami, 2006) also studied the combined power and ejector refrigeration cycles. Zhang and Lior (2007) proposed many configurations for combined cooling and power applications, having large cooling capacity, but they require high temperature (450 °C).

Although a lot of work is reported in literature on the combined ejector absorption refrigeration system, combined vapour compression ejector refrigeration system for cooling and heating, combined power and refrigeration system etc., yet no/little work is reported that combine ARC, ERC, VCR cascaded and ejector organic Rankine cycle (EORC) for simultaneous production of cooling of wide range using single waste heat source.

In this paper ARC, EORC and VCR cascaded are combined. The waste heat is used to operate EORC and ARC directly while VCR cycle is operated by turbine power. For thermodynamic performance both energy and exergy analysis is carried out. Exergy analysis provides more meaningful information as compared to energy analysis. It helps in identifying site, quantity and source of thermodynamic inefficiency. Exergy destruction in each component is evaluated in order to find out the direction for potential improvement. In order to investigate the influence of various operating parameters on system performance, the parametric analysis is also performed.

SYSTEM DESCRIPTION

The waste heat driven triple effect cycle is shown in fig. 1. The industrial waste heat transfers its heat to HRVG (1-2) and generator (2-12).The working substance used in ORC and ERC is R-141b. The superheated vapour of R-141b (4) is generated in HRVG and expanded in turbine. The turbine exhaust (5) act as a primary motive fluid for ERC (5-6-7-9-10-11). At the exit of the nozzle section of the ejector, the high velocity primary refrigerant vapours create vacuum before mixing section and thereby extracting secondary refrigerant vapour (11) from evaporator. This causes cooling in ejector organic Rankine cycle (EORC). The mixture of primary and secondary fluid (6) is condensed in condenser-1. The work output of turbine is fed to two compressors of VCR cascade cycle (23-24-25-26-27-28-29-30-31-32-33-34). The refrigerant used in both high temperature cycle (HTC) and low temperature cycle (LTC) is nitrous oxide (N_2O). The single effect LiBr-H_2O absorption refrigeration cycle (13-14-15-16-17-18-19-20-21-22) consists of generator, condenser-2, evaporator-2, absorber, solution heat exchanger (SHE), pump-2 (P-2) and throttle valves (TV-2 and TV-3). The description of ARC cycle is given elsewhere (Herold et al., 1996).

THERMODYNAMIC ANALYSIS

To perform the thermodynamic analysis of the triple effect refrigeration cycle, the assumptions made are:

(1) The cycle is under steady state operation.

(2) Pressure drop in pipes is neglected.

(3) Pump work is neglected

(4) The states at the exit of condenser and evaporator are saturated liquid and saturated vapour respectively.

(5) Chemical, kinetic and potential exergies are neglected and only physical exergy is considered.

(6) In order to avoid crystallization, temperature of strong solution exiting the solution heat exchanger is kept 7-8 °C above crystallization temperature.

(7) Lithium bromide solution in the absorber and generator are in equilibrium state at their respective pressure and temperature.

(8) The Lithium bromide solution exiting the generator and absorber are saturated.

(9) Heat loss to the surrounding is neglected.

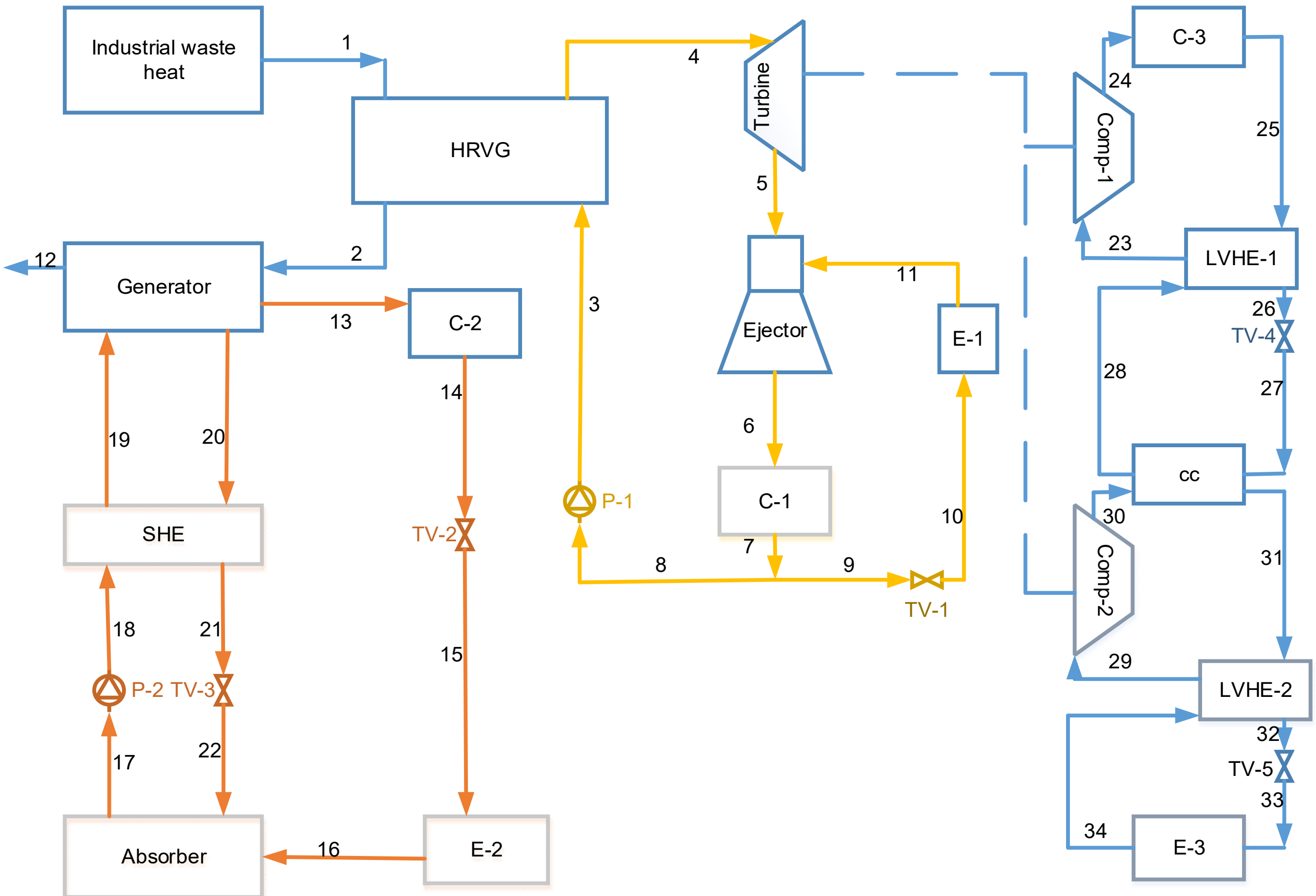

Figure 1 Schematic diagram of waste heat driven triple effect refrigeration cycle

For the thermodynamic analysis, the required parameters are given in Table 1.

The energy and exergy analysis of the refrigeration system incorporates the use of concept of mass and energy conservation and second law of thermodynamics. The performance of the system, based on first law alone, can be evaluated using thermal efficiency (η_{th}) which is defined as the ratio of the cooling effect to the heat input and is described as:

$$\eta_{th} = (\dot{Q}_{e1} + \dot{Q}_{e2} + \dot{Q}_{e3})/\dot{Q}_{in} \tag{1}$$

where $\dot{Q}_{in}$ is the total heat input given by waste heat to the system and $\dot{Q}_{e1}$, $\dot{Q}_{e2}$ and $\dot{Q}_{e3}$ are the refrigeration effect in ERC, ARC and VCR cycle respectively.

The second law of thermodynamics facilitates in assessing the thermodynamic performance of the system based on exergy. For a control volume, the exergy flow rate of a fluid, considering only physical exergy, is expressed as:

$$\dot{E} = \dot{m}[(h - h_0) - T_0(s - s_0)] \tag{2}$$

Exergy balance for a control volume undergoing steady state process is expressed as:

$$\dot{ED} = \sum \dot{E}_{in} - \sum \dot{E}_{out} + \left(\sum \dot{Q}\left(1 - {T_0}/{T}\right)\right)_{in} + \sum \dot{Q}\left(1 - {T_0}/{T}\right)_{out} \pm \sum \dot{W} \tag{3}$$

where $\dot{ED}$ is the exergy destruction rate and $\dot{E}$ is the exergy flow rate.

The exergy efficiency is expressed on the basis of second law of thermodynamics as

$$\eta_{ex} = (\Delta\dot{E}_{e1} + \Delta\dot{E}_{e2} + \Delta\dot{E}_{e3})/\dot{E}_{in} \tag{4}$$

where $\dot{E}_{in}$ is the exergy input of waste heat to the system and $\Delta\dot{E}_{e1}$, $\Delta\dot{E}_{e2}$, and $\Delta\dot{E}_{e3}$ are the exergy change in evaporator-1 of ERC, evaporator-2of ARC and evaporator-3 of VCR cycle respectively, which are expressed as:

$$\Delta \dot{E}_{e1} = \dot{E}_{11} - \dot{E}_{10} \tag{5}$$

$$\Delta \dot{E}_{e2} = \dot{E}_{16} - \dot{E}_{15} \tag{6}$$

$$\Delta \dot{E}_{e3} = \dot{E}_{34} - \dot{E}_{33} \tag{7}$$

$$\dot{E}_{in} = \dot{E}_{1} - \dot{E}_{12} \tag{8}$$

Table 1 Parameters used in the modelling

Parameters	Value
Environment temperature (℃)	15
Environment pressure (kPa)	101.35
Waste heat source temperature (℃)	160-180
Turbine inlet pressure (kPa)	900-1700
Turbine back pressure (kPa)	220-300
Turbine isentropic efficiency (%)	85
ERC evaporator temperature (℃)	-1 to -9
ARC evaporator temperature (℃)	5
VCR evaporator temperature (℃)	-85
Condenser-2 temperature (℃)	35
Condenser-3 temperature (℃)	35
Absorber temperature (℃)	35
Mass flow rate of waste flue gas (kg s^{-1})	20
Pump isentropic efficiency (%)	70
HRVG efficiency (%)	100
Pinch point temperature difference (℃)	10
Nozzle efficiency (%)	90
Mixing chamber efficiency (%)	85
Diffuser efficiency (%)	85

The relations obtained by the application of mass, energy and exergy balance to each component are outlined in Table 2.

RESULTS AND DISCUSSION

Based on the thermodynamic relations, a program is developed in engineering equation solver (EES) (Klein and Alvarado, 2005). The results thus obtained are presented and discussed in this section.

Fig. 2 shows the results of energy analysis of triple effect refrigeration cycle. It is found that about 21.79% of total waste industrial heat can be converted into useful output and rest of the energy is lost to environment.

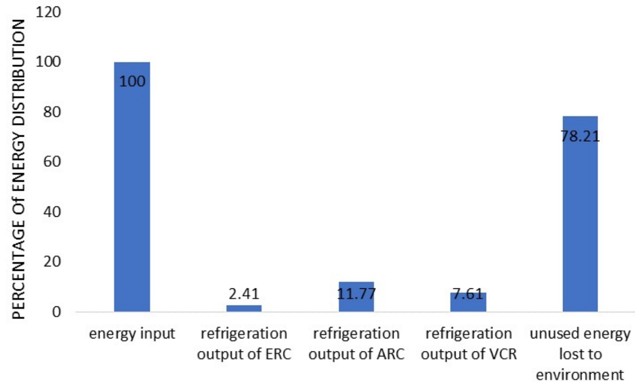

Figure 2 Percentage of energy distribution for the triple effect refrigeration cycle

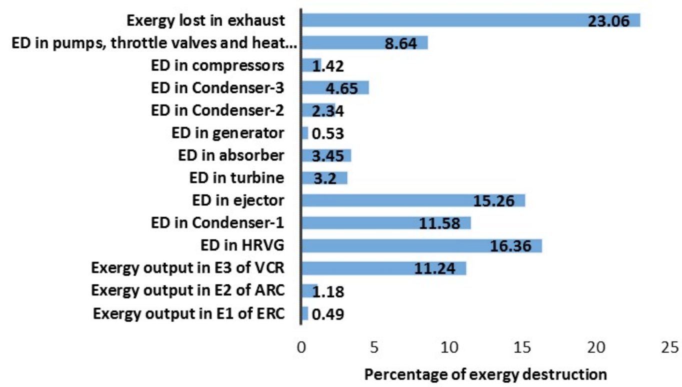

Figure 3 Percentage of exergy distribution in output and destruction for triple effect refrigeration cycle

Fig. 3 shows the result of application of exergy analysis to the triple effect refrigeration cycle. It is found that about 12.91% of the total input exergy is available as useful exergy output. Of the total exergy input, the useful exergy output of VCRC, ARC and EORC accounts for 11.24%, 1.18% and 0.49% respectively. About 23.06% of the input exergy is lost to the environment by hot flue gas and the remaining 64.03% exergy input is destroyed because of irreversibilities associated with various components and exergy loss in absorber and condenser. The second law analysis reveals that the maximum improvement is possible in HRVG as it accounts for highest exergy destruction (16.36%), followed by ejector and condenser of ejector refrigeration cycle. Therefore, from the perspective of second law of thermodynamics, HRVG, ejector and condenser-1 needs expert attention so as to enhance the overall performance of the system.

Table 2 Energy and exergy relations for the sub system of triple effect refrigeration cycle

Component	Energy relations	Exergy relations
HRVG	$\dot{m}_1(h_1 - h_2) = \dot{m}_3(h_4 - h_3)$	$\dot{ED}_{HRVG} = T_0(\dot{m}_2 s_2 - \dot{m}_1 s_1 + \dot{m}_4 s_4 - \dot{m}_3 s_3)$
Turbine	$\dot{W}_T = \dot{m}_4(h_4 - h_5)$	$\dot{ED}_T = T_0(s_5 - s_4)$
Ejector	$\dot{m}_5 h_5 + \dot{m}_{11} h_{11} = (\dot{m}_5 + \dot{m}_{11}) h_6$ $\mu = [\eta_n \eta_m \eta_d (h_{pf,n1} - h_{pf,n2,s})/(h_{mf,d,s} - h_{mf,m}$	$\dot{ED}_{ej} = T_0(\dot{m}_6 s_6 - \dot{m}_5 s_5 - \dot{m}_{11} s_{11})$
Condenser-1	$\dot{Q}_{c1} = \dot{m}_6(h_6 - h_7)$	$\dot{ED}_{c1} = T_0(\dot{m}_7 s_7 - \dot{m}_6 s_6 - \dot{Q}_{c1}/T_{c1})$
Evaporator-1	$\dot{Q}_{e1} = \dot{m}_{11}(h_{11} - h_{10})$	$\dot{ED}_{e1} = T_0(\dot{m}_{11} s_{11} - \dot{m}_{10} s_{10} - \dot{Q}_{e1}/T_{e1})$
Generator	$\dot{Q}_g = \dot{m}_f(h_2 - h_{12}) = \dot{m}_{13} h_{13} + \dot{m}_{20} h_{20} - \dot{m}_1$	$\dot{ED}_g = T_0(\dot{m}_{20} s_{20} + \dot{m}_{13} s_{13} + \dot{m}_{12} s_{12} - \dot{m}_{19} s_{19} - \dot{m}_2 s_2)$
Absorber	$\dot{Q}_a = \dot{m}_{16} h_{16} + \dot{m}_{22} h_{22} - \dot{m}_{17} h_{17}$	$\dot{ED}_a = T_0(\dot{m}_{17} s_{17} - \dot{m}_{16} s_{16} - \dot{m}_{22} s_{22} - \dot{Q}_a/T_a)$
Condenser-2	$\dot{Q}_{c2} = \dot{m}_{13}(h_{13} - h_{14})$	$\dot{ED}_{c2} = T_0(\dot{m}_{14} s_{14} - \dot{m}_{13} s_{13} - \dot{Q}_{c2}/T_{c2})$
Evaporator-2	$\dot{Q}_{e2} = \dot{m}_{16}(h_{16} - h_{15})$	$\dot{ED}_{e2} = T_0(\dot{m}_{16} s_{16} - \dot{m}_{15} s_{15} - \dot{Q}_{e2}/T_{e2})$
Compressor-1	$\dot{W}_{comp1} = \dot{m}_{24}(h_{24} - h_{23})$	$\dot{ED}_{comp1} = T_0 \dot{m}_{24}(s_{24} - s_{23})$
Compressor-2	$\dot{W}_{comp2} = \dot{m}_{24}(h_{30} - h_{29})$	$\dot{ED}_{comp2} = T_0 \dot{m}_{30}(s_{30} - s_{29})$
Evaporator-3	$\dot{Q}_{e3} = \dot{m}_{34}(h_{34} - h_{33})$	$\dot{ED}_{e3} = T_0(\dot{m}_{34} s_{34} - \dot{m}_{33} s_{33} - \dot{Q}_{e3}/T_{e3})$
Condenser-3	$\dot{Q}_{c3} = \dot{m}_{24}(h_{24} - h_{25})$	$\dot{ED}_{c3} = T_0(\dot{m}_{25} s_{25} - \dot{m}_{24} s_{24} - \dot{Q}_{c3}/T_{c3})$
CC	$\dot{m}_{30} h_{30} - \dot{m}_{31} h_{31} = \dot{m}_{28} h_{28} - \dot{m}_{27} h_{27}$	$\dot{ED}_{cc} = T_0[\dot{m}_{28}(s_{28} - s_{27}) + \dot{m}_{31}(s_{31} - s_{30})]$
SHE	$\dot{m}_{20} h_{20} - \dot{m}_{21} h_{21} = \dot{m}_{19} h_{19} - \dot{m}_{18} h_{18}$	$\dot{ED}_{she} = T_0[\dot{m}_{19}(s_{19} - s_{18}) + \dot{m}_{21}(s_{21} - s_{20})]$
LVHE-1	$\dot{m}_{25} h_{25} - \dot{m}_{26} h_{26} = \dot{m}_{23} h_{23} - \dot{m}_{28} h_{28}$	$\dot{ED}_{lvhe1} = T_0 \dot{m}_{28}(s_{23} - s_{28} + s_{26} - s_{25})$
LVHE-2	$\dot{m}_{31} h_{31} - \dot{m}_{32} h_{32} = \dot{m}_{29} h_{29} - \dot{m}_{34} h_{34}$	$\dot{ED}_{lvhe2} = T_0 \dot{m}_{34}(s_{29} - s_{34} + s_{32} - s_{31})$
TV-1	$h_9 = h_{10}$	$\dot{ED}_{tv1} = T_0 \dot{m}_{10}(s_{10} - s_9)$
TV-2	$h_{14} = h_{15}$	$\dot{ED}_{tv2} = T_0 \dot{m}_{15}(s_{15} - s_{14})$
TV-3	$h_{21} = h_{22}$	$\dot{ED}_{tv3} = T_0 \dot{m}_{22}(s_{22} - s_{21})$
TV-4	$h_{26} = h_{27}$	$\dot{ED}_{tv4} = T_0 \dot{m}_{27}(s_{27} - s_{26})$
TV-5	$h_{32} = h_{33}$	$\dot{ED}_{tv5} = T_0 \dot{m}_{33}(s_{33} - s_{32})$
Pump-1	$\dot{W}_{p1} = \dot{m}_3(h_3 - h_8)$	$\dot{ED}_{p1} = T_0 \dot{m}_3(s_3 - s_8)$
Pump-2	$\dot{W}_{p2} = \dot{m}_{18}(h_{18} - h_{17})$	$\dot{ED}_{p2} = T_0 \dot{m}_{18}(s_{18} - s_{17})$

Fig. 4 shows the variation of desired refrigeration output with industrial waste heat temperature. It is clear from the figure that increase in temperature of waste heat (flue gas) causes increase in refrigeration output of EORC and VCRC. Increased temperature results in better quality of refrigerant vapour at turbine inlet, thus more power is available from turbine to VCRC, which increases refrigerant mass flow rate in evaporator of VCRC. This results in higher refrigerating effect in vapour compression refrigeration cycle. Improved quality of vapour refrigerant at turbine inlet because of increased waste heat temperature, results in better quality at turbine exit. This causes increase in velocity of primary motive fluid at nozzle exit creating greater vacuum at secondary vapour entrance to ejector. The increased vacuum results in increase in the mass flow rate of the secondary refrigerant through the evaporator of ejector refrigeration cycle and thus higher refrigeration effect in EORC. The decrease in cooling capacity of ARC with the increase in industrial waste heat temperature is because of the decrease in flue gas temperature at the outlet of HRVG. Thus, heat input to the ARC decreases, ensuing decrease in mass flow rate of refrigerant (water) through the evaporator-2. The combined refrigeration output follows the similar declining behaviour with the increase in waste heat temperature because ARC dominates over EORC and VCRC.

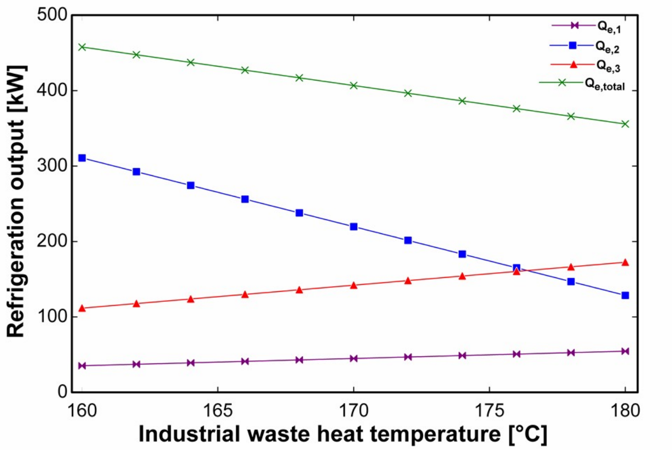

Figure 4 Effect of industrial waste heat temperature on refrigeration output

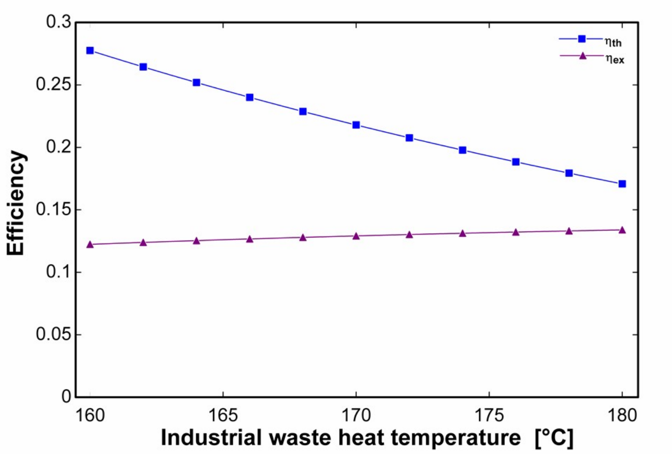

Figure 5 Effect of industrial waste heat temperature on energy and exergy efficiency

Fig. 5 represents the variation of first law based thermal efficiency and second law based exergy efficiency with industrial waste heat temperature. There is appreciable drop in thermal efficiency of the system with the rise in temperature of waste heat. It is because of the dominating nature of absorption refrigeration output over VCR refrigeration output and ejector refrigeration output. In contrast, insignificant increase in exergetic efficiency of the overall system with the increase in waste heat temperature is observed and it is attributed to the fact that the exergy output of VCRC is much higher than that of EORC and ARC. With the rise in flue gas temperature, the exergy output of VCRC exhibits rising behaviour while that of ARC shows declining trend. Hence, the exergetic efficiency of the given system shows minimal increase with the rise in temperature of waste heat.

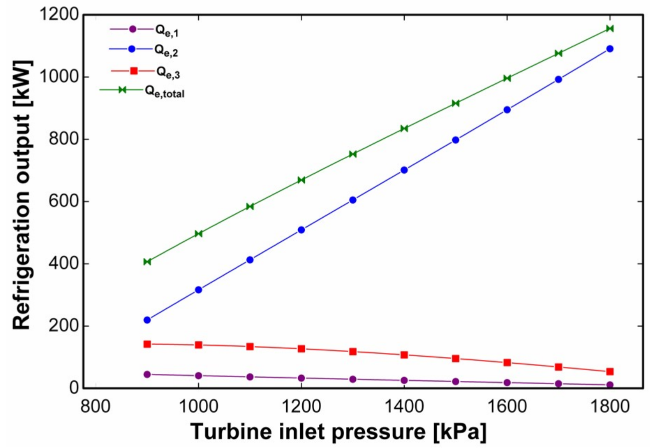

Figure 6 Effect of turbine inlet pressure on refrigeration output

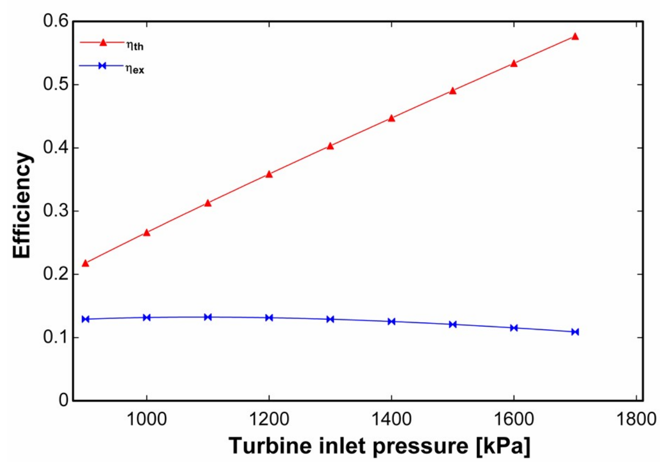

Figure 7 Effect of turbine inlet pressure on energy and exergy efficiency

Fig. 6 shows the effect of turbine inlet pressure on cooling capacity of EORC, ARC and VCRC and on combined cooling capacity. It is found that the cooling capacity of ARC increases considerably with the increase in turbine inlet pressure while cooling capacity of EORC and VCRC decreases with the same. The increase in the turbine inlet pressure lowers the mass flow rate of refrigerant (R141b) vapours generated in the HRVG. This results in reduction in absorption of thermal energy from the flue gas through the HRVG, leading to higher flue gas temperature at HRVG exit (2). This allows higher heat supply to the generator of ARC, causing considerable enhancement in the cooling capacity of ARC due to increased mass flow rate of refrigerant (water) through evaporator-2. The reason for the reduction of refrigeration output of ejector refrigeration cycle is the decrease in the mass flow rate of refrigerant (R141b) vapours through the turbine and the decrease in the turbine exit temperature. The decrease in turbine exit temperature reduces the velocity of primary fluid at ejector nozzle, resulting in reduced mass flow rate of secondary refrigerant through evaporator-1. Also, the reduced mass flow rate through the turbine causes reduction in power input to the compressors and thus decrease in flow rate of N_2O across evaporator-3. Hence, refrigeration output of both EORC and VCRC decreases with the increase in turbine inlet pressure.

The variation of energy and exergy efficiency with the increase in turbine inlet pressure is shown in fig. 7. With the increase in turbine inlet pressure, the energy efficiency is found to increase significantly while exergy efficiency decreases after initial increase. The rate of increase in refrigeration output of ARC is much higher than the rate of reduction in refrigeration outputs of EORC and VCRC with the increase in turbine inlet pressure. Therefore, variation of energy efficiency follows the pattern of variation of ARC refrigeration output. Initially, the rise in exergy output of ARC surpasses the drop in exergy output of EORC and VCRC, but later, the trend gets reversed. Consequently, the exergy efficiency increases initially and then starts decreasing with the hike in turbine inlet pressure.

CONCLUSIONS

The main conclusion drawn from the energy and exergy analysis of combined ARC, EORC and VCRC triple effect refrigeration cycle are as follows:

✓ Highest irreversibility occurs in HRVG followed by ejector and condenser of ejector Rankine cycle. Therefore, more attention is required to be paid on these components while designing and optimization.

✓ Around 21.79% of the total energy input is available as useful refrigeration output while only 12.91% of the exergy input is available as useful exergy output.

✓ Exergy output of VCR cycle is significantly higher than that of the absorption and ejector refrigeration cycles. It is of the order of 11%, 1% and 0.5%, respectively for VCRC, EORC and ARC.

NOMENCLATURE

ARC	absorption refrigeration cycle
COP	coefficient of performance
C_p	specific heat (kJ/kg.K)
$\dot{E}$	exergy flow rate (kW)
$\dot{ED}$	exergy destruction rate (kW)
EORC	ejector organic Rankine cycle
ERC	ejector refrigeration cycle
EV	expansion valve
h	specific enthalpy (kJ/kg)
HRVG	heat recovery vapour generator
HTC	high temperature cycle
LTC	low temperature cycle
$\dot{m}$	mass flow rate (kg/s)
ORC	organic Rankine cycle
P	pressure (kPa)
$\dot{Q}$	heat transfer rate (kW)
s	specific entropy (kJ/kg.K)
SHE	solution heat exchanger
TV	throttle valve
T	temperature (°C or K)
VCRC	vapor compression refrigeration cycle
$\dot{W}$	work transfer rate (kW)

Subscripts

0	reference state
1, 2…	operating points
a	absorber
c	condenser
cc	cascade condenser
comp	compressor
d	diffuser section
e	evaporator
ej	ejector
ex	exergetic
g, gen	generator
i	inlet, inside
k	any component
lvhe	liquid vapour heat exchanger
max	maximum
m	mixing section
n	nozzle section
o	outlet, outside
p	pump
she	solution heat exchanger
T	turbine
tv	throttle valve
u	useful

Greek symbols

η	efficiency
μ	entrainment ratio
Σ	sum of
ψ	specific exergy

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