Comparative Analysis of Performance of a Combined Power and Cooling System with Vapor Compression and Absorption Refrigeration System as Bottoming Cycle

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Abstract— In this paper, performance of a combined reheat regenerative steam turbine (ST) based power cycle and a vapor compression refrigeration system (VCRS) is presented and compared with the performance of a similar combined system (CS) integrated with a vapor absorption refrigeration system (VARS). R134a is used as refrigerant in theVCRS while water−LiBr is the working fluid pairs in the VARS. Net power and efficiencies (energy and exergy) of the topping power cycle, COP and exergy efficiency of the bottoming refrigeration system (RS), energy utilization factor (EUF), exergy efficiency of the CS and irreversibility of CS components are the various performance parameters that are compared. The comparative analysis indicates that the net power, EUF, efficiencies of the VCRS integrated CS are more than those of the VAR based system for the same operating conditions. Moreover, these are achieved in the VCR based CS with much lower total system irreversibility, higher COP and exergy efficiency of the bottoming RS.

Key words: Comparative analysis, Energy, Exergy, Vapor absorption refrigeration system, Vapor compression refrigeration system, Reheat regenerative power cycle

I. INTRODUCTION

Vapor compression refrigeration system (VCRS) has the advantage of high COP and large cooling capacity over the other refrigeration systems used in refrigeration and HVAC industry. Chlorofluorocarbons (CFCs) used in VCRS however have large degree of ozone depletion potential (ODP)and global warming potential (GWP). CFCs therefore nowadays are substituted with HCFCs, HCs and HFCs which have relatively less ODP and GWP. Use of refrigerants with high ODP and GWP will anyhow be restricted in the near future and low ODP/GWP refrigerants (HFCs and HCs) might be the solutions of the future even though the risk of toxicity or flammability is high with such refrigerants [1]. Continuous efforts are therefore being made by the research community to evaluate performance (both energetic and exergetic) of VCRS with low ODP/GWP refrigerants having superior thermo physical and heat transfer properties. Dalkilic and Wongwises [2] made a theoretical performance study on a traditional VCRS with refrigerant mixtures of R134a, R152a, R32, R290, R1270, R600, and R600a in various proportions and compared their results with R12, R22, and R134a. They recommended blends of R290/R600a (40/60 by wt. %) and R290/R1270 (20/80 by wt. %) as alternatives for R12 and R22 respectively. Padilla et al. [3] conducted experiments on a domestic refrigerator using R12 and R-413a refrigerants for comparative dynamic analysis of performance. The overall energy and exergy performance of the system working with R413A was found better than that of R12. Sagia and Paignigiannis [4] carried out exergy analysis of a single stage VCR cycle with refrigerant mixtures R-404A, R-410A, R-410B and R-507 as working fluids providing detailed information on the variation of cycle's exergy efficiency with evaporating and condensing temperatures.

Many such investigations have been done to analyze performance of VCRS with existing and alternate refrigerants not only from the first law (energy) point of view but also on the basis of second law (exergy) of thermodynamics. Qureshi et al. [5] studied experimentally the effects of employing a mechanical sub cooling cycle with a residential 1.5 ton simple VCRS with R22 refrigerant in the main cycle and R12 in the subcooling cycle. The results indicated improvement in evaporator cooling capacity and the cycle's second law efficiency with the use of mechanical sub−cooling (5◦C−8◦C). Llopis et al. [6] made performance comparison of R404A and R507A in the evaporating temperature range between −36◦C and −20◦C at a condenser temperature of 40◦C in a double-stage VCR cycle with and without a subcooler. The plant efficiency was higher for R404A than R507A without subcooler, especially at high evaporating temperatures. However with the subcooler incorporated, the COP values with R404A and R507A were almost similar. New design concepts of combined absorption–compression refrigeration systems are also developed. One such combined system is proposed in Ref. [7] which is obtained by coupling a single stage VCRS and a single effect vapor absorption refrigeration system (VARS) in the cascade condenser where refrigerant (R22) vapor of the VCRS condenses and rejects heat to evaporate water of the Water−LiBr VARS. They also made comparative analysis based on first and second laws between the hybrid and the independent VCRS for the same cooling capacity of 66.67 kW and found significant improvement in performance with the hybrid system. Han et al. [8] in his combined absorption–compression refrigeration system utilized waste heat of flue gas to generate high-temperature vapor from one stream of absorber leaving strong NH3–H2O solution which in the combined system is split into two streams. The vapor is then used for power generation in a turbine that drives the compressor of the VCRS. Low temperature flue gas heat is further utilized in a low temperature heat exchanger to generate NH3 vapor from the other stream of absorber leaving strong solution. The NH3 vapor from the evaporator is also split into two streams; one stream goes into the absorber while the other stream is compressed in the VCRS compressor which then mixes with vapor stream coming from a rectifier and then condensed in the condenser. The rectifier receives exhaust vapor from the turbine, strong NH3–H2O solution from the low temperature heat exchanger and generates pure ammonia vapor and weak solution. Zhao et al. [9] in an optimization study considered two such NH3–H2O combined absorption/compression refrigeration cycles with one and two solution circuits. The combined cycle with one solution circuit consists of gas engine exhaust driven absorption chiller, a condenser, an evaporator and a compressor which is also driven by the engine. In the combined cycle with two solution circuits, the condenser and evaporator of the one solution circuit combined cycle are replaced with a second absorber and a second generator. The study showed better performance in case of the combined cycle with two solution circuits.

Thus we come across several studies in the literature based on VCRS alone and in the combined mode with VARS. VCR uses high grade energy while VAR uses low grade thermal energy as driving force for refrigeration. Usually VCRS outperforms VARS, but the advantage with VARS is that it can be operated with waste heat stream, non-conventional energy sources such as solar or geothermal energy. Trygg and Amiri [10] made a comparison between vapor compression and absorption chillers for energy utility in district cooling and Swedish municipality industries. Elsafty and Daini [11] made a cost analysis to compare a vapor compression air-conditioning system with a solar powered water−LiBr vapor absorption system (single and double-effect). The analysis was carried out on the basis of present worth value (PWV) and equivalent annual cost (EAC) methods. The PWV and EAC were found minimum in case of the double-effect vapor absorption system. Similarly, many studies have been carried out to analyze VARS performance thermodynamically with the help of energy and exergy [12−15]. Combined power and VARS based refrigeration cycle have also been investigated [16-20].

II. RESEARCH ELABORATIONS

In most of the power industries/plants, cooling is mainly achieved through use of VCRS driven by high grade electric energy produced indigenously in the power plants. Therefore it may be relevant that a combined power and cooling plant be analyzed to evaluate its performance with a VCRS integrated to provide the cooling in the plant. In the present work, we provide thermodynamic analysis (energy and exergy) of a combined reheat regenerative type thermal power plant and a VCRS based cooling system. Since often comparison is provided between VCR and VAR systems [21-23]and although some of the facts are known, still a comparison between VARS and VCRS based combined vapor power−cooling system is possible considering that neither a thermodynamic analysis comparing two combined ST based power and cooling systems is available nor it was attempted before from this point of view. Although the advantages and disadvantages of the VCRS and VARS separately are known, but when they are integrated with the topping power cycle it will depend upon many other factors, most importantly the change in configuration and accordingly the performance of the power cycle will vary. Thus the main aim of the present work is to provide a comparative energy and exergy analysis between the combined power and cooling systems with VCRS and VARS as bottoming cycles. The study quantifies the difference in performance of the topping VPC and the bottoming cooling system for the same operating conditions.

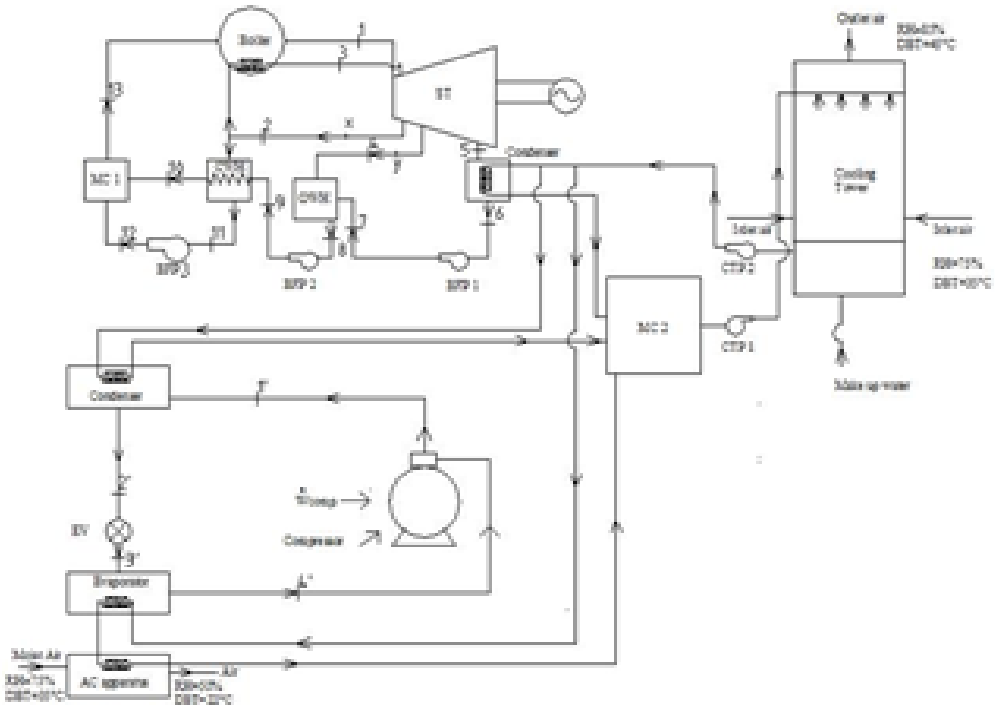

Fig. 1: Schematic of the combined reheat regenerative vapor power cycle and vapor compression refrigeration system

The configurations of the combined power and cooling systems integrated with VCRS and VARS are shown in Fig. 1 and Fig. 2 respectively. The topping vapor power cycle (VPC) of both the systems are more or less similar; however, in the VCRS based configuration in Fig. 1, no steam from the steam turbine (ST) is extracted. One mixing chamber (MC) and a boiler feed pump (BFP) have been removed from the topping VPC of the VARS based combined system (CS) in Fig 2 to obtain the new VCRS based CS (Fig.1). The piping network, which was required in the VARS based configuration (Fig. 2) for carrying cold water from cooling tower (CT) exit to the absorber and then hot water from the absorber to the mixing chamber, is not required in the VCRS based CS. The electricity produced by the steam power plant is the source of energy for the compressor of the VCRS. It may be noted that there were total three mixing chambers viz. MC1, MC2 and MC3 in the VARS based CS (Fig. 2) whereas in the present VCRS based combined configuration (Fig. 1), there are only two mixing chambers namely MC1 and MC2, which are identical of MC2 and MC3 in Fig. 2. Similarly, the total number of boiler feed pumps(BFPs) in the VCRS based CS (Fig. 1) is three compared to four numbers of BFPs in the VARS based configuration in Fig. 2.

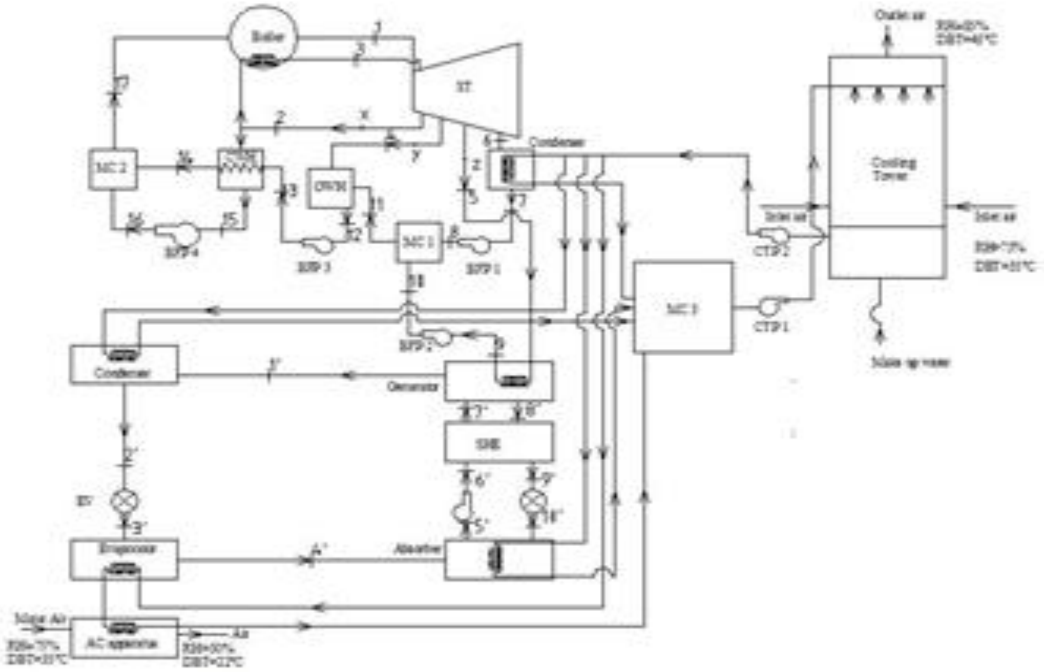

Fig. 2: Schematic of the combined reheat regenerative vapor power cycle and LiBr vapor absorption refrigeration system

III. CALCULATIONS: VAPOR POWER CYCLE

Coal is used as fuel in the boiler of the power plant and it has the following composition: Carbon (C) 60%, Hydrogen (H) 4%, Oxygen (O)3%, Nitrogen (N)2%, Sulphur (S) 3%, Moisture (H2O)4% and Ash content 24%.Fuel lower heating value is calculated from its composition using standard molar specific enthalpy of devaluation of reactants and products assuming complete fuel combustion. Thermodynamic properties such as specific enthalpy, entropy of water at the saturated liquid state, steam at saturated vapor and superheated state at various pressures and temperature are determined from International Associations for the properties of water and steam (IAPWS) formulation 1997. Fuel chemical exergy, energy and exergy lost with the boiler leaving flue gas at the exhaust temperature are calculated. Thermo-mechanical exergy of fuel and chemical exergy of air are assumed to be zero. The mass flow rate of steam generated in the boiler is calculated from energy balance applied to the boiler control volume. The various parameters assumed for system simulation are shown in Table 1.

Fuel flow rate -	20 kg/s
Boiler pressure-	150 bar
Reheat pressure-	30 bar
Condenser pressure-	0.1 bar
ST isentropic efficiency-	85%
Pump isentropic efficiency-	85%
CT exit water temperature-	25◦C
Flue gas temperature at boiler exit-	300◦C
RS condenser exit water temperature-	30◦C
RS evaporator exit water temperature-	10◦C
VARS absorber exit water temperature-	30◦C
Compressor isentropic efficiency-	85%
Chilled water temp. A/C apparatus inlet-	10◦C
CT inlet air temperature-	35◦C
CT inlet air relative humidity-	75%
CT exit air temperature-	45◦C
CT exit air relative humidity-	85%
Air flow rate though A/C apparatus-	4 kg/s

A/C apparatus inlet air temperature-	35◦C
A/C apparatus exit air temperature-	22◦C
A/C apparatus inlet air relative humidity-75%	
A/C apparatus exit air relative humidity-	50%
VARS generator temperature-	80◦C
VARS condenser temperature-	35◦C
VARS evaporator temperature-	5◦C
VARS absorber temperature-	35◦C
Evaporator cooling load-	14000 kW

Table 1: Assumed values of parameters

The power developed by the steam turbine (ST) is:(1)

Total pump work required for running the boiler feed pumps (BFPs):(2)

The pumping power required for running the two CT side pumps, (see Fig. 1) is calculated considering pipe dimensions, pipe roughness, static lift, water density, water velocity in pipe, frictional and other minor losses[24].

The energy efficiency of the power cycle is the ratio of net power developed to the fuel input energy. Similarly the exergy efficiency is the ratio between net plant power and total exergy supplied to the boiler (sum of fuel's chemical exergy and thermo mechanical exergy of air).

Total exergy of boiler leaving flue gas is the sum of chemical and thermo-mechanical exergy as explained in Ref. [25].

Boiler irreversibility:(3)

Turbine irreversibility(4)

Power cycle condenser (PCC) irreversibility:(5)

Irreversibility in BFP1:(6)

Irreversibility in BFP2:(7)

Irreversibility in BFP3:(8)

Total BFP irreversibility:(9)

Irreversibility in OWH:(10)

Irreversibility in CWH:(11)

Irreversibility in MC1:(12)

In the above equations x and y represents the fractions of steam extracted per kg of steam for the closed water heater (CWH) and open water heater (OWH) respectively. The irreversibility in the CT, cooling tower pumps (CTPs) and mixing chamber 2 (MC2) is calculated following the procedure described in Ref. [25].

IV. CALCULATIONS: VCRS

A single stage VCRS is coupled as a bottoming cycle with the reheat regenerative VPC.

In the VCRS, the refrigerant R−134a (1, 1, 1, 2-tetrafluoroethane) is used as working fluid. The CT in the combined system (CS) not only assists in supplying cold water to the power cycle condenser(PCC), certain amount of water is also routed through the VCRS condenser and the evaporator (also the air conditioning(AC) apparatus). The water circulated through the evaporator provides the latent heat of vaporization required for evaporation of R134a in the VCRS evaporator. Since the temperatures of water at evaporator inlet and outlet are specified, the mass flow rate of water can be calculated from heat balance in the evaporator. The chilled water thus produced in the evaporator flows through the AC apparatus to cool and dehumidify the hot and humid air. Mass flow rate of moist air, its dry bulb temperature (DBT) and relative humidity (RH) at inlet and exit of the AC apparatus are specified. Therefore, from known values of chilled water flow rate and its inlet temperature, the temperature of water at the AC apparatus outlet can be determined from energy balance in the AC apparatus. Similarly the water flow rates through the PCC and VCRS condenser are also determined from energy balance applied to these devices. The water from the PCC, VCRS condenser and the AC apparatus, all goes to the mixing chamber (MC2). The temperature of the mixed water stream is calculated from the SFEE applied to MC2 (see Ref. [24] for details).

The temperature and pressure dependent thermodynamic properties of R−134a are calculated using equations taken from the Ref. [26]. Density of R−134a at a given temperature is calculated using the Newton-Raphson iterative method with an initial guess value of density calculated from the ideal gas equation. The following values of gas constant(R), critical temperature (T_c) and critical density c are taken.

R= 81.488856 J/kg K; Tc =374.18K; c =508 kg/m^3

VCRS cooling load is specified as a model input parameter from which the mass flow rate of refrigerant is calculated as follows.

(13)

Compressor work is calculated using the following relation. (14)

The coefficient of performance (COP) of the VCRS is: (15)

The maximum COP and exergy efficiency of the VCRS are defined as follows

(16)

(17)

The energy utilization factor (EUF) of the combined power and cooling system is: (18)

Exergetic efficiency of the combined vapor power and VCRS is defined by the following equation. (19)

Same formulae as specified in Ref. [25] are used for calculation of exergy destruction/irreversibility in the VCRS condenser, expansion valve, evaporator and the AC apparatus. The irreversibility values in the VCRS condenser, expansion valve and the evaporator of the two systems are different due to the refrigerants and its properties because it is R134a in the VCRS and water in the VARS. The irreversibility in the compressor is calculated using the following equation. (20)

The total irreversibility of the combined power and VCRS based cooling system is the sum of irreversibility in all components of the combined system.

V. STUDIES AND FINDINGS

Table 2 shows the comparison of performance of the combined power and cooling system with VARS and VCRS. It is seen that compared to the VAR based system, the net power output of the VCR based system is more although the steam generation rate in the boiler is slightly less for the CS integrated with VCRS. The BFP pumping power of the VCR based CS is only marginally less compared to that of the VAR based CS although there is one less number of BFP in the VCR based CS in Fig. 1. However, the pumping power required for running the CTPs is significantly less in the VCR based CS compared to the VAR based CS. This is mainly due to water mass flow rate which is less in the VCR based CS because no water is required to be circulated through the absorber as required in the VAR based system in Fig.2. The frictional loss in the VCR based CS in Fig. 1 is also less due to removal of the pipe network connecting the CT exit, absorber and the MC3 in Fig.2.Obviously the compressor power requirement of the VCRS is more than the SP power requirement in the VARS, but overall the total pumping power requirement is less in the VCR based CS. Also the ST power W_{ST} the VCR based CS is more because no steam from ST is extracted here as it is required in the VAR based CS for supplying the heat for vapor generation in the generator Q_G. Therefore the net power output is more from the topping power cycle of the VCR based CS. Slight change in the steam generation rate in the two CSs is due to difference in values of enthalpy at state points 13 and 17 in Fig. 1 and 2 respectively. Since the net power is more in the VCR based CS for the same fuel input energy and exergy supplied to the boiler, hence the energy and exergy efficiencies are also more in case of the VCR based CS.It is seen that with VCRS as bottoming cycle, the energy losses in the PCC condenser is more. This is mainly due condensation of relatively more amount of steam in the condenser because no steam is extracted from the ST.

In the bottoming cooling cycle, it is observed that the COPs (both actual and Carnot) are significantly higher in case of the VCRS. Higher COP in case of VCRS is obvious due to lower magnitude of W_{COMP} compared to Q_G .In the VCRS, more refrigerant is required to achieve the same amount of cooling. VCRS exergy efficiency is also more compared to VARS. EUF of the VCR based CS is slightly more due to higher net power output from the power cycle. The exergy efficiency of the VCR based CS is more compared to the VAR based CS.

Table 3 shows irreversible losses occurring in various components of the CS integrated with VCRS and VARS. In the VCR based CS, the irreversible losses in the boiler, condenser, CT, BFPs of the power cycle components and expansion valve of the RS are more compared to losses in the respective components of the VAR based system. However, these losses in the ST, OWH, CTPs and refrigeration system condenser (RSC) are less, particularly in the CTPs. Irreversibility of the CWH, exhaust flue gas and the AC apparatus in the two systems are exactly the same. Evaporator irreversibility is also more or less the same for both the systems. Irreversibility of the VCRS compressor is also significantly less (298.023 kW) compared to that of the generator−solution heat exchanger (SHE)−SP−absorber assembly of the VAR based CS (total 4720.39 kW). The difference in total irreversibility also arises due to the irreversible losses of the mixing chambers. There are two and three mixing chambers respectively in Fig. 1 and Fig. 2; therefore, irreversible losses in the mixing chambers of Fig. 1 are less compared to irreversibility of mixing chambers in Fig. 2. This ultimately results in lower total system irreversibility in the VCR based CS.

Parameters	With VARS	With VCRS
Net power (MW)	176.752	180.196
Steam generation rate (kg/s)	170.498	169.609
BFP pumping power (MW)	3.154	3.138
CT side pumping power (MW)	6.127	5.028
Solution pump power (W)	202.5	−
Wcomp (W)	−	2075.188
Energy efficiency of VPC (%)	35.625	36.319
Exergy efficiency of VPC (%)	33.141	34.269
.VPC condenser loss (kW)	238208.483	253749.675
COP(Actual)	0.771	6.746
COP (Carnot)	1.181	9.270
mR (kg/s)	5.924	91.801
m_{LiBr} (kg/s)	38.018	−
Q_C (kW)	14829.0	16075.0
Exergetic efficiency of RS (%)	11.215	17.562
EUF of the CS	0.384	0.391
Exergy efficiency of CS (%)	33.008	34.203

Table 2: Comparison of component irreversibility of the CS with VARS and VCRS

Irreversibility (kW)	**With VARS**	**With VCRS**
^{I}boiler (kW)	115194.302	115680.002
I^{ST} (kW)	22040.911	19408.981
^{I}PCC (kW)	9778.663	10416.642
^{I}BFP (kW)	218.796	248.451
I_{OWH} (kW)	4372.586	4085.312
I_{CWH} (kW)	3602.665	3602.665
I_{MC1} (kW)	7706.578	0.000
^{I}MC 2 (kW)	0.000	1579.737
^{I}MC 3 (kW)	1780.766	
I_{fg} (kW)	179397.921	179397.921
I_{CT} (kW)	16171.771	16131.783
I_{CTP} (kW)	22226.994	14512.731
I_{RSC} (kW)	390.791	395.048
^{I}EVA (kW)	642.219	642.209
^{I}ExV (kW)	40.146	242.163
I_{COMP} (kW)	–	298.023
I_{SHE} (kW)	121.486	
I_{SP} (kW)	0.186	
^{I}GEN (kW)	4598.718	
I_{AC} (kW)	3.625	3.625

Table 3: Comparison of component irreversibility of the CS with VARS and VCRS

VI. Conclusion

The important conclusions which are made from this comparative analysis include the followings.

- Compared to the VAR based CS, the BFP pumping power is slightly less while the CTP pumping power is significantly less in the VCR based CS. This reduction in CTP pumping power in the VCR based CS is caused by the change in water mass flow rate which is significantly less in the VCR based system due to not having the absorber as in the VCRS where it requires cold water circulation to the absorb the heat released during exothermic reaction between water vapor and LiBr salt. The CTP pumping power also reduces in the VCR based CS due to lower frictional head loss caused by absence of the pipe network connecting the CT outlet, VARS absorber and the mixing chamber (MC3) as required in Fig. 2. Although the compressor power of the VCRS is more than the solution pump (SP) power of the VARS, but reduction in the BFP and CTP pumping power, particularly the CTP pumping power caused a significant reduction in the total negative power requirement in the VCR based CS. Power developed by the ST is also more in the VCR based CS due to the fact that steam is not extracted as it is done in the VAR based system to provide the heat required for vapor generation in the generator. Hence the net power output from the topping power cycle is more in the VCR based system compared to that of the VAR based CS.
- Accordingly the power cycle energy and exergy efficiencies of the VCR based CS are higher than those of the VAR based system. Moreover, a given cooling effect is produced with much higher COP and exergetic efficiency with VCRS and most importantly this is done with significant gain in the net power output from the topping power cycle.
- The total irreversibility of the power cycle components is less in the VCR based CS due to irreversibility difference caused by one extra mixing chamber (MC1 in Fig. 2) in the VAR based system.
- Irreversible losses in the components of the cooling systems are also less in case of the VCRS because of compressor irreversibility which is lower than the total losses in the generator−solution heat exchanger (SHE) −SP−absorber assembly of the VARS. Consequently, the total system irreversibility of the VCR based CS becomes less than that of the VAR based system at all operating conditions. Among the VCRS components the condenser produces the highest irreversibility followed by the evaporator, compressor and the expansion valve.
- Finally, it can be concluded that among the two combined power and cooling systems, the system with VCRS may be preferred if higher net power output and minimum total system irreversibility are the sole criteria. Definitely the cost of the VCR based CS will be less than that of the VAR based system. The VAR based combined configuration may also be useful in case when excess steam is produced and lost unused in the plant. If cooling is at all to be produced by using VARS without losing much power from the power cycle, the exhaust heat of the boiler leaving flue gas could be an option as a source of heat for operating the VARS.

Appendix

A. Nomenclature

c_p	specific heat (kJ kg^{-1}K^{-1})
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c_p	specific heat (kJ kg^{-1}K^{-1})
h	specific heat (kJ kg^{-1}K^{-1})
h	enthalpy (kJ kg^{-1})
I	Irreversibility rate (kW)
m	mass flow rate (kg s-1)
Q	cooling capacity (kW)
RS	refrigeration system
T	temperature (K)
T_0	reference temperature (K)
W	work rate (kW)
Ex	Exergy (kW)

B. Subscripts

a	air
AC	Air Conditioning
BFP	boiler feed pump
C	RS condenser
ch	chemical
COMP compressor	
CS	combined system
CT	cooling tower
CTP	cooling tower pump
CWH closed water heater	
ex	flue gas exhaust
Ei	evaporator inlet
Eo	evaporator outlet
E	evaporator
EVA	evaporator
ExV	expansion valve
f	fuel
fg	flue gas
GEN	generator
i	inlet
MC	mixing chamber
o	outlet
OWH	open water heater
PCC	power cycle condenser
r	refrigerant
RS	refrigeration system
RSC	RS condenser
s	steam
SHE	solution heat exchanger
SP	solution pump
ST	steam turbine
tm	thermo-mechanical
w	water

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