257. A novel water-heating system for utilizing waste heat contained in cooling water of a steam plant condenser

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Abstract

The utilization factor of a steam power plant can be enhanced by adopting suitable measures that are aimed to utilize the waste heat from the condenser. One such scheme is presented in this paper which uses the waste heat of condenser to produce hot water at steady conditions. This article evaluates all the possible conditions available for the implementation of the said scheme in a steam power plant and also provides the associated outcomes. As per the scheme provided, the low-grade heat occupant 'cooling water' from the condenser of the power unit is fed directly into the evaporator section of the heat pump. The power required for the operation of the heat pump is produced by a separate power production system which incorporates a turbine that runs on the super-heated steam. The super-heated steam is formed by heat transfer with the main steam that is generated in the central boiler or extracted from the main turbine. An additional heat exchanger, then arranged, provides hot water at the steady conditions. A detailed parametric study of the scheme is performed based upon the thermodynamic predictions of the states of the given system using Engineering Equation Solver (EES). The results outline the viability of two most important variables, called the Effectiveness and Ratio of comparison, defined later, being evaluated for the set of varying parameters including the refrigerants, compression ratio of the heat pump and the mass flow and inlet temperature of water supply. The study depicts the maximum performance, for the heat input equivalent to 1200 KW, with the refrigerant characteristics of R-600, the compression ratio of 3.35-3.85 for the heat pump and the mass flow and inlet temperature of cold water at 11-12.5 Kg/s and 08-13 ^{O}C respectively. This study will impose the significant ecological and economic benefits when acquired with the available power producing setup, particularly in cold environmental conditions.

Keywords: *Waste heat recovery; Utilization factor; Steam power plant condenser; Cogeneration plant; Back pressure plant; Absence of Cooling tower; Space heating; Water heating; Applications of Heat pump*

1. Introduction

The Kelvin-Planck statement of second law of thermodynamics limits a power cycle to obtain a 100 % efficiency as it necessitates the availability of two reservoirs for a system to produce the net amount of positive work. In other words, for a system to complete the power cycle, some of its energy must be rejected into the heat sink (either the atmosphere or a secondary coolant) to produce the positive work [1]. In a steam power plant, the working substance now called a system in this context, rejects its heat into the condenser, so named as the system condenses into the liquid water in this device. Neglecting the losses associated with the pipe's insulation, flow friction and flow leakages, the magnitude of heat discharge by the system depends on its state at the exit of low-pressure Turbine along with the mass flow rate of the system. While the mass flow rate of the system at the condenser is the function of the heat source temperature that is available to achieve the feasible temperature of the system at the inlet of the high-pressure Turbine, and the availability of regeneration, the state of the system at the exit of the low-pressure Turbine depends on the system's pressure and the dryness fraction at the corresponding state, which is then the function of the temperature of secondary substance (i.e. heat sink). The condition of a secondary substance depends on the geographical conditions at the site.

No matter, a considerable portion of the total heat that is discharged into the condenser is often wasted in the rivers, lakes, and atmosphere. A study on the evaluation of the extent of this energy loss from the condenser of the power plant yielded the mean energy loss of 78.3 % with the standard deviation of 7.898. This study was, however, based on the energy and exergy analysis of fewer power plants that are reported in the literature [2-9].

The energy loss from the condenser of the power plant is attributable to the low exergy of the system less than the minimal one that is required to match an appropriate utility, the lack of interest of the power plant owners due to the limited financial incentives and the lack of sufficient knowledge pertaining to the installation of such setups that can take the advantage of this low exergy of the waste heat to produce a significant outcome.

Of all the problems mentioned above, the low exergy of the system in the condenser is the common cause of the deficiency of any prolific exertion for the utilization of this energy loss. This was assessed by the same study depicted above, which concluded the mean exergy destruction of the system in the condenser to a mere 3.038 % of the total exergy loss with the standard deviation of 2.465.

The energy loss lessens the utilization factor of the steam power plant. The utilization factor is defined as the total fraction of the available heat input that is used to produce the mechanical work and fulfills heating or cooling loads [10]. A waste heat recovery scheme may even though, not produce any significant mechanical work; nevertheless, it possesses a potential to satisfy heating or cooling loads.

Furthermore, this energy loss has far-reaching impacts on waterborne species and even human health, if the condenser heat is discharged directly into the water bodies, as it disturbs the entire ecosystem of water bodies due to thermal pollution. The Thermal pollution in water bodies, for example, can be lethal to a wide array of sensitive fish populations, and it may also pose a potential risk to human health by way of increased infectious diseases such as dengue, malaria, etc. [11]. The cooling towers remains a remedy to avoid thermal pollution in water bodies, as they aid in discharging the condenser heat into the atmospheric air. This is mostly done by the evaporation of condenser cooling water which produces the cooling effect, thereby increasing the relative humidity of the atmospheric air and decreasing the temperature of secondary substance that can sustain the cyclic process of power production. However, this does not increase the utilization factor, and also raises the concern about the environmental consequences of the effluents that are discharged into the atmosphere from the cooling towers. These effluents include potentially harmful chemicals along with large quantities of water vapor [12].

In the direction to overcome the aforementioned problems, the heat recovery schemes have come under scrutiny in recent years. The extensive set of researches helped to evolve, what are now known as back pressure cogeneration plants due to their capability to produce two or more essential utilities by utilizing a common heat source, along with the recovery of heat from the cooling water of the condenser. Besides the back pressure cogeneration plants, another configuration of cogeneration plant uses the extraction of steam from the turbine to satisfy the thermal load in addition to the generation of mechanical power [1]. The latter one generally delivers medium pressure steam, although at the expense of mechanical power, and with the requirement of cooling towers or any other heat discharge equipment. Moreover, the steam extraction cogeneration plant also doesn't recover the waste heat from the condenser. For such reasons, the back pressure cogeneration plants are more efficient and feasible, both thermodynamically and economically on the criteria of waste heat recovery.

The back pressure cogeneration plants of today are available in various configurations, depending on the available power production setup and the mode of application for which they are acquired. The applications of back pressure plants are though limited, due to the fact that the condenser of the power plant generally operates in the pressure range of 5-15 KPa at most, and the temperature of the primary fluid (mostly water), which is allowed to get in the thermal contact with the cooling water of the condenser is thus constrained to a maximum of 54 ^{O}C for the available pressure range. An attempt to increase the temperature of the primary fluid might come from the increase in the condenser pressure, but that would adversely affect the power production. The various applications of back pressure cogeneration plants include low-temperature desalination, low-temperature cooling, hot water heating and space heating, and much more.

The low-temperature desalination processes have been adopted and studied extensively by various researchers [13-15]. The most notable technique as outlined in these researches is to utilize the temperature gradient between two water bodies to evaporate warmer water at low pressure and condense the resultant vapor with colder water to obtain fresh water. Ahmed, et. al. applied this technique to utilize the waste heat contained in cooling water of condenser for desalination purpose [16].

The low-temperature refrigeration can be achieved by exploiting the waste heat of the condenser through the cascade absorption-compression refrigeration systems. These systems offer more efficiency than the stand-alone counterparts [17] and that they can employ the waste heat of condenser at any subtle stage. For example, the condenser of the Compression refrigeration system can be cooled by the waste heat equipped cooling water to reduce the electricity consumption due to lower condensing temperatures, or the waste heat can be used as a heat source in the generator of vapor-absorption system.

A back pressure cogeneration setup for heating purpose typically include a heat exchanger that transfers the energy from the cooling water of the condenser to the water required for heating. However, the temperature of the heated water as constrained by the pressure of the condenser may not suffice for most of the heating applications.

In order to elude the inadequacy of the back pressure cogeneration plants to heat the water at an appropriate level without compromising for the magnitude of power production, a novel water heating scheme is presented in this paper to heat the water to a state so that it could be effectively used for the diverse range of applications. This paper evaluates all the possible conditions available for the implementation of the said scheme in a steam power plant and also provides the associated outcomes. For this objective, a model of the scheme is developed on EES software.

2. Waste energy recovery strategy

The heat recovery scheme discussed in this paper relates to a combined heat and power plant ‘CHP,' also known as cogeneration plant, operating upon the topping cycle. This cycle derives its name from the ability to, at first convert heat into power and then utilizes the excess energy to meet the thermal needs [18]. The present scheme does not relate any significance to power generation as it rather concentrates to satisfy the thermal load and provides the hot water at around 70 ^{O}C, however, the essence of this scheme can be effectively realized in a CHP cogeneration plant.

The schematic diagram of waste heat recovery strategy is shown in Fig. 1. In accordance with the plan, the waste heat equipped cooling water from the condenser of the power production setup is fed into the evaporative section of the heat pump. The waste heat of the cooling water is designated with the certain magnitude of 1200 KW, which is tantamount to a heat discharge of a small power plant with an average capacity of 1-4.5 MW, that is being contingent with the thermal efficiency of the power plant. The cooling water, after releasing the excess heat in the evaporator of the heat pump is recirculated in the condenser of the power plant. The heat pump operates on an organic Rankine cycle, utilizing one of the following refrigerants: R11, R12, R134a, R141b, R404a, R500 and R600. The choice of refrigerants is in accordance with the fact, that the saturation temperature of refrigerants must be smaller than the temperature of cooling water. The power required to run the compressor of the heat pump is produced by a Turbine that forms an integral part of a power unit, and it expands the superheated steam produced in the heat exchanger. The heat exchanger transfers the energy to the primary fluid of the power unit via the heat transfer from the main steam that is produced either in the main boiler or is extracted from the main turbine. The cold water stream at the certain mass flow rate and the temperature is directed towards the condenser of the heat pump and the power unit, where it absorbs the heat from the primary fluid of condensers, the primary fluid being refrigerant for the heat pump and the heated water for the power unit. The water stream then flows towards an auxiliary heat exchanger which primarily works on dual nature, supplying the necessary heat when the temperature of its inlet stream is less than the required one and absorbing the excess heat for the opposite. In this way, this auxiliary device controls the temperature of water and supplies the steady stream of water for heating applications. The various intensive properties and variables at different states are defined in the process diagram, Fig. 1; that narrates with the practical values available in the current modes of operation.

3. Thermodynamic model

A thermodynamic model is proposed to determine the effect of various parameters on the performance variables of the present scheme. The analysis is done on EES, which is a general equation solving program possessing the high accuracy of thermodynamic properties of different substances. The program can solve numerous algebraic equations, performs optimization and can even be used for regression analysis [19].

There are two performance variables set for the analysis, the effectiveness, and the ROC. The effectiveness of the scheme is defined as the ability of the proposed plant to utilize the heat of both condensers effectively to an extent as required for the application. This means that any increase or decrease in the temperature of water stream exiting the two condensers, relative to the desired temperature would curtail its value. The ROC as is known as Ratio of comparison is the ratio of the amount of energy, used by the proposed plant to heat the water to that of the energy that would have been utilized for a simple heat exchanger to accomplish the same task. A smaller value of ROC dictates the maximum effectiveness of the scheme.

Various assumptions are taken in the analysis, as deemed for easiness, but they are rather typical for most of the analysis that is performed on the same subject. These includes for both the heat pump and the power unit and are given below:

- Each component of the heat pump and the power unit is modeled as a control volume operating at the steady state. The stray heat transfer to the environment is also neglected from each component.
- There is no pressure drop along the heat exchangers, and they operate at the ideal effectiveness of 100%.
- The state of the system at the exit of the evaporator is saturated vapor state, while that of the system at the exit of the condenser is a saturated liquid state.
- The flow across the throttling valve is isenthalpic.
- The compressor of the heat pump and the Turbine of the power unit have the isentropic efficiency of 80 % and the power transmission between the two occurs without any losses.
- The fluid is pumped isentropically in the power unit.

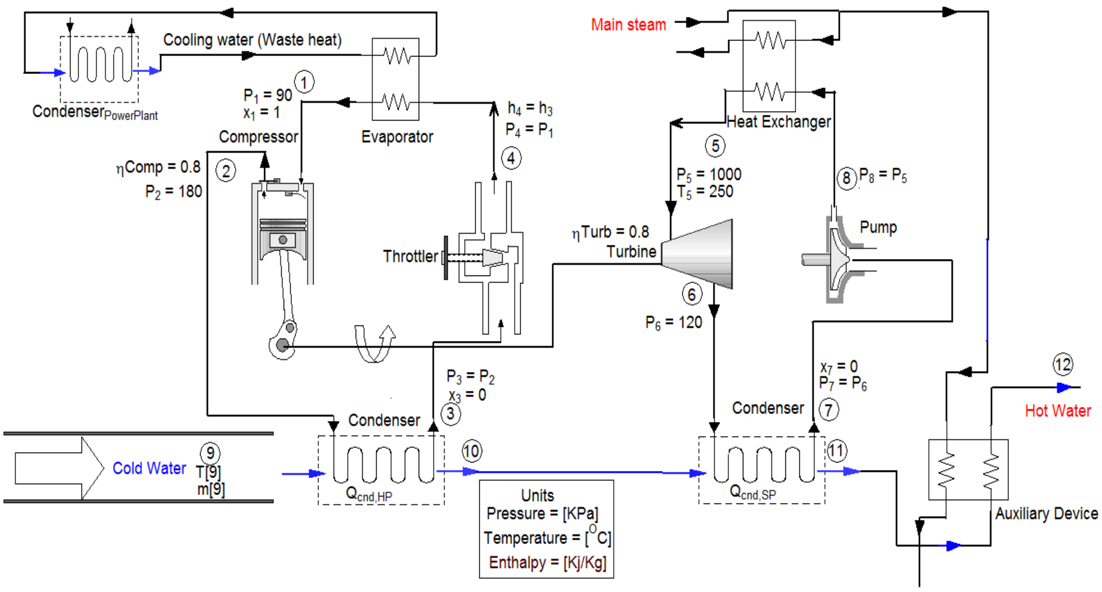

Fig. 1. Schematic diagram of heat recovery strategy.

The property functions and algebraic equations, used in the analysis to determine various intensive properties at different states of the system are listed in Table 1 in conjunction with the values of other properties known at the respective states. The property functions in Table 1 refers to compressible fluids, the states of which are defined in terms of only two independently variable properties. Amongst the set of refrigerants used in the analysis, R11, R12, R141b, R404a and R600 use the Martin-Hou (1995) equation of state to relate the thermodynamic properties while R134a and R600 employ Span's (2000) equation of state for the same [20].

Table 1. Property functions at different states of the system and algebraic equations for plant components.

State Point	Location	Known Properties	Property function and Equations
1	Ref., HP, Exit of Evaporator	$P_1 = 90$, $x_1 = 1$	$h_1 = Enthalpy\ (Refrigerant, x = x_1, P = P_1)$ $s_1 = Entropy\ (Refrigerant, x = x_1, P = P_1)$
2	Ref., HP, Exit of Compressor	$r = \frac{P_2}{P_1}, \eta_C = 0.8, s_{2s} = s_1$	$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1}$ $h_{2s} = Enthalpy\ (Refrigerant, s = s_{2s}, P = P_2)$
3	Ref., HP, Exit of Condenser	$P_3 = P_2, x_3 = 0$	$h_3 = Enthalpy\ (Refrigerant, x = x_3, P = P_3)$
4	Ref., HP, Exit of Throttle valve	$P_4 = P_1$	$h_4 = h_3$
5	Steam., SP, Exit of Boiler	$P_5 = 1000, T_5 = 250$	$h_5 = Enthalpy\ (Water, T = T_5, P = P_5)$
6	Steam., SP, Exit of Turbine	$P_6 = 120, \eta_T = 0.8, s_{6s} = s_5$	$W_{Turb,SP} = W_{Comp,HP}$ $W_{Comp,HP} = m_{refrigerant} * (h_2 - h_1)$ $\eta_T = \frac{h_5 - h_6}{h_5 - h_{6s}}$ $h_{6s} = Enthalpy\ (Water, s = s_{6s}, P = P_6)$
7	Steam., SP, Exit of Condenser	$P_7 = P_6, x_7 = 0$	$h_7 = Enthalpy\ (Water, x = x_7, P = P_7)$ $V_7 = Volume\ (Water, x = x_7, P = P_7)$
8	Steam., SP, Exit of Pump	$P_8 = P_5$	$W_{Pump} = V_7 * (P_8 - P_7)$ $h_8 = h_7 + W_{Pump}$
9	Coldwater, Inlet		
10	Coldwater, Exit of HP Condenser		
11	Coldwater, Exit of SP Condenser		
12	Hotwater, Exit of Auxiliary device	$T_{12} = 70$	

The thermodynamic analysis, consists of several criteria, with each of them accompanies a particular outcome that forms a part of developing the conclusion. However, surrounded by these criteria, are the number of equations that are similar, and are mostly derived from the energy balance of control volumes. The energy balance of control volume asserts that "the net amount of energy transferred into or out of the system equals the change in energy of the system, with the change in energy being zero for the steady state." These equations and the variables associated with them are all discussed below, as they form the framework upon which the conclusion is to be laid.

Refrigerant mass flow rate in heat pump:

For a fixed heat input to the evaporator of the heat pump, the energy balance equation determines the mass flow rate of the refrigerant that would be circulating along the device to fully recover the condenser waste heat.

$$Q_{evp,HP} = m_{ref} * (h_1 - h_4) \tag{1}$$

As can be seen from the above equation, the mass flow rate of the refrigerant is an inverse function of the enthalpy difference of it across the evaporator for the fixed heat input. The heat energy of the refrigerant at the exit of the evaporator as obtained from $\mathrm{m_{ref}} * \mathrm{h_1}$, is an important parameter to predict the performance of the plant. A higher value of this parameter for the same compression ratio of the heat pump would aid in discharging more heat in the condenser of the same.

Heat transfer across the condenser of Heat Pump:

The amount of heat transferred from the refrigerant flowing through the condenser of heat pump is given by

$$Q_{cnd,HP} = m_{ref} * (h_2 - h_3) \tag{2}$$

The inlet water stream is subjected to this heat of the condenser and the temperature of the water at the exit of this first stage is governed by

$$Q_{cnd,HP} = m_{water} * (h_{10} - h_9) \tag{3}$$

Work input to the compressor:

The total work input required for the operation of the heat pump is evaluated as

$$W_{comp,HP} = m_{ref} * (h_2 - h_1) \tag{4}$$

The compressor work is produced by the Turbine that runs on the super-heated steam. For the fixed enthalpy difference across the Turbine, as governed by the constant state at the inlet and exit of the Turbine, the mass flow rate of the steam is the direct function of the Turbine work as shown below

$$W_{Turb,SP} = m_{steam} * (h_5 - h_6) \tag{5}$$

Heat transfer across the condenser of power unit and auxiliary device:

The amount of heat absorbed by the water stream in the condenser of the power unit is proportional to the mass flow of the steam and the enthalpy difference of the same across the condenser.

$$Q_{cnd,SP} = m_{steam} * (h_6 - h_7) = m_{water} * (h_{11} - h_{10}) \tag{6}$$

For the heat energy of water, greater or less than the desired one as nominated by the temperature of 70 ^{O}C, the water has to be routed to the auxiliary device to maintain its steady condition. The amount of the heat loss or gain by the water stream as it flows through the auxiliary device is determined by considering the amount of heat energy required by the water flow and the amount of energy gained by it, where

$$Q_{required} = m_{water} * (h_{12} - h_9) \tag{7}$$

$$Q_{gain} = Q_{water,first} + Q_{water,second} \tag{8}$$

Therefore,

$$Q_{auxiliary} = Q_{required} - Q_{gain}, for\ Q_{gain} < Q_{required} \tag{8}$$

$$Q_{lost} = Q_{gain} - Q_{required}, for\ Q_{gain} > Q_{required} \tag{9}$$

System performance:

The performance variables, called as Effectiveness and ROC, are evaluated by following equations:

$$Effectiveness = \frac{Q_{gain}}{Q_{required}}, for\ Q_{gain} < Q_{required} \tag{10}$$

$$Effectiveness = 1 - \frac{Q_{lost}}{Q_{gain}}, for\ Q_{gain} > Q_{required} \tag{11}$$

$$ROC = \frac{Q_{boiler} + Q_{auxiliary}}{Q_{required}} \tag{12}$$

As can be perceived from the equations of performance variables, the effectiveness compares the amount of heat gained about the heat required. For the amount of heat gain greater than the heat required, the extra potential is rather wasted and is thus accounted in the formulation. The ROC is as previously defined, compares the energy used in this scheme to the energy used by conventional means.

The various criteria that are followed in the given analysis are highlighted below:

I. Finding the compression ratio for maximum performance.

The analysis begins by fixing the mass flow rate and temperature of the cold water stream with the magnitude of 12 Kg/sec and 10 ^{O}C to find a compression ratio that could give the maximum performance for each refrigerant. The refrigerants mentioned above were all kept as a part of the analysis where they replace the keyword 'ref' in the above equations.

II. Determining the mass flow of inlet water stream for maximum performance.

Once the optimal compression ratio is fixed for each refrigerant of the heat pump, the mass flow of the inlet water stream is investigated for the maximum performance for the constant temperature of 10 ^{O}C.

The value of compression ratio is seized at 3.5 for the heat pump as this value lie within the optimal range of compression ratio.

III. Investigating the temperature of inlet water stream for maximum performance.

For the constant mass flow of the inlet water stream at 12 Kg/sec and fixed compression ratio of 3.5, this criterion involves ascertaining the temperature of inlet water stream for the viability of the performance parameters.

4. Thermodynamic model results and discussion

Using the thermodynamic model presented above, the criterion (I) yields the compression ratio for each refrigerant for maximum performance, as listed in Table 2.

Table 2. Compression ratio for maximum performance when m_9 = 12 Kg/s, T_9 = 10 ^{O}C

Refrigerants	Compression ratio
R600	3.374
R500	3.575
R404a	3.655
R141b	3.525
R134a	3.807
R11	3.5
R12	3.472

Fig. 2 demonstrates the impact of compression ratio on the performance parameters of the plant for the constant mass flow and temperature of inlet water stream as mentioned in criterion (I). The optimal compression ratio is where the effectiveness is maximum, or ROC is minimum. It can be analyzed from the plot that the compression ratio ranges between 3.35-3.8 for the set of refrigerants for maximum performance.

It must be noted that any variation in the mass flow and temperature of cold water stream drifts the optimal compression ratio for each refrigerant. This can be analyzed from Table 3; that while the mass flow and temperature has been changed, the values of compression ratio change in a definite pattern relative to the earlier values, with the R600 again showing the maximum performance with the least compression ratio and R134a with the most. Therefore, for the conditions of inlet water stream other than those of criterion (I), the compression ratios should be chosen accordingly. R600 appears more efficient to be used in terms of low compression ratio; however, the total energy input for all refrigerants at their optimal compression ratio is same as discussed in the subsequent section.

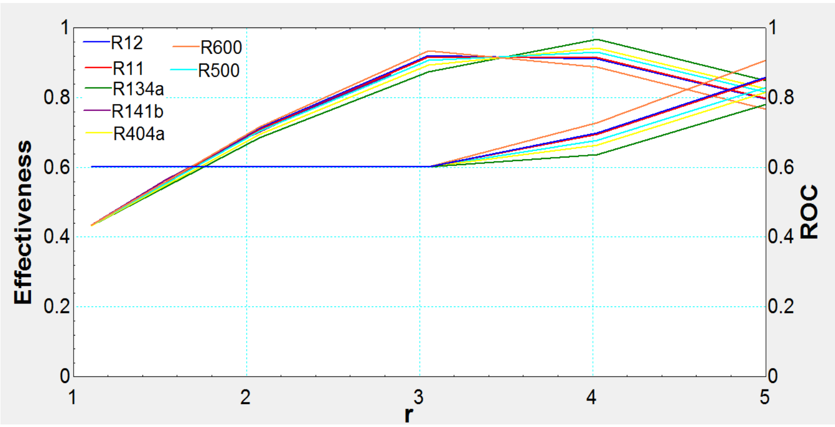

Fig. 2. Effect of compression ratio of Heat pump on the effectiveness and ROC of the plant.

Table 3. Effect of m_9 and T_9 on the optimal compression ratio of refrigerants.

Refrigerants	m_9 = 10 Kg/s	m_9 = 11 Kg/s	m_9 =12	m_9 = 13	m_9 = 14	m_9 = 15	m_9 = 16

	T_9 = 15 ^{O}C	T_9 = 12.5 ^{O}C	Kg/s T_9 = 10 ^{O}C	Kg/s T_9 = 7.5 ^{O}C	Kg/s T_9 = 5 ^{O}C	Kg/s T_9 = 2.5 ^{O}C	Kg/s T_9 = 0 ^{O}C
R600	2.275	2.786	3.374	4.037	4.77	5.566	6.414
R500	2.341	2.905	3.575	4.356	5.252	6.262	7.382
R404a	2.392	2.971	3.655	4.446	5.343	6.343	7.439
R141b	2.347	2.893	3.525	4.241	5.036	5.903	6.831
R134a	2.461	3.078	3.807	4.653	5.615	6.688	7.865
R11	2.318	2.862	3.5	4.237	5.073	6.005	7.028
R12	2.301	2.839	3.472	4.205	5.04	5.977	7.01

The variation of optimal compression ratio for different refrigerants as obtained in the analysis and shown on plots is on par the following explanation: For an optimal value of compression ratio for any refrigerant, there must not be any heat transfer in the auxiliary device. For the satisfaction of such case, the energy of the water stream at the inlet of the auxiliary device must equal the energy of the water desired for the heating application. Hence,

$$h_{11} = h_{12}\, for\ optimal\ performance\ of\ the\ plant.$$

For the fixed mass flow and temperature of inlet cold water stream, the energy associated with the same is constant. Hence the total energy gains of the stream for optimal performance is deterministic and equal to required heat. This can be envisaged from the following equation:

$$Q_{required} = m_{water} * (h_{12} - h_9) = m_{water} * \{(h_{10} - h_9) + (h_{11} - h_{10})\} \qquad Eq. 14$$

From above equation, the energy of water at the inlet of the second heating stage is constrained, once the required heat is affirmed. This would fix the amount of heat transfer in the condenser of the power unit and hence the flow of steam that has to circulate through the power unit. Since the state of the steam at the inlet and exit of the turbine is known for the given criterion, the work done by the turbine can be evaluated and which is same for any refrigerant. Further, the work done by the Turbine equals the work done on the compressor; hence the compression ratio for different refrigerants is such that the amount of heat discharged by them in the condenser of heat pump matches the amount of heat to be gained by water in the first heating stage.

$$Q_{water,first} = m_{water} * (h_{10} - h_9) = m_{ref} * (h_2 - h_3) \qquad Eq. 15$$

The corollary of above explanation is that the work done by the turbine, and the heat flow in the first and second heating stage remains the same for the configuration of each refrigerant at optimal compression ratio.

It can further be analyzed that, with the compression ratio more than the optimal range for the fixed states as per criterion (I), the refrigerant that exhibited maximum effectiveness at the least compression ratio now has the minimum effectiveness when compared with the other refrigerants, for the new value of compression ratio. This is due to the excess of heat absorbed by the water stream in the first stage, that is more than the desired one, as a result of which the same must be withdrawn to maintain the steady conditions. This withdrawal of excess heat represents a loss and hence a decrease in effectiveness. The effect of surplus compression ratio on the performance parameters is presented in Table 4.

Table 4. Performance parameters of the plant when m_9 = 12 Kg/s, T_9 = 10 ^{O}C with the compression ratio of heat pump = 4.

Refrigerants	Effectiveness	ROC
R600	0.7431	0.8673
R500	0.7778	0.8073
R404a	0.7881	0.7905
R141b	0.7672	0.8249
R134a	0.8088	0.758
R11	0.7657	0.8275
R12	0.762	0.8339

An intermediate value of 3.5 is then selected, as per criterion (II) to decide for the adaptability of the mass flow of water that can take full advantage of the proposed model. Table 5 lists the optimal mass flow rates of water. The variation of optimal mass flow rates is in accordance with the concept that a fixed compression ratio of heat pump limits the work required by the compressor and hence the work done by the Turbine. The repercussion of which is the fixed amount of heat loss in the condensers of the heat pump and the power unit. As previously elaborated that the maximum effectiveness of the plant comes with the value of heat gains equal to the heat required, the optimal mass flow of water stream is such that the sum of condensers heats equals the required heat. Unlike the criterion (I), the work required by the compressor and the heat loss in both stages in the present criterion are not equal for the same configurations of different refrigerants. This is because the states of the refrigerants as fixed by the conditions of the same at the exit of the evaporator and the compressor are different corresponding to their saturation and super-heated conditions. The effect of which is a variable compression power required for each and hence the different rate of heat loss in the condensers. Table 6. depicts the impact of optimal mass flow rates on the ROC of the plant, work required by the compressor and the heat loss in the condensers for each refrigerant.

Table 5. Optimal mass flow rates of water stream for r = 3.5 and T_9 = 10 ^{O}C

Refrigerants	Optimal mass flow rates
R600	12.29
R500	11.83
R404a	11.67
R141b	11.93
R134a	11.39
R11	11.99
R12	12.05

Table 6. Effect of optimal mass flow rates on the ROC of the plant, work required by the compressor and the heat loss in the condensers for criterion (II).

Refrigerants	ROC of the plant	Rate of Work required by compressor (KW)	Heat loss in condenser of Heat pump (KW)	Heat loss in condenser of Power unit (KW)
R600	0.6114	241.3	1441	1645
R500	0.5961	226.7	1427	1546
R404a	0.5905	221.5	1421	1510
R141b	0.5995	229.9	1430	1568
R134a	0.5805	212.6	1413	1449
R11	0.6013	231.7	1432	1580
R12	0.6034	233.7	1434	1539

Any value of mass flows more or less than the optimal range would curtail the effectiveness of the plant as shown in Table 7. This is due to the additional heat of the boiler to be transferred to the auxiliary device in the former case and the superfluous heat loss in the latter.

Table 7. Effect of m_9 = 13 Kg/s and m_9 = 11 Kg/s on the effectiveness of the plant with r = 3.5 and T_9 = 10 ^{O}C.

Refrigerants	Effectiveness of the plant for m_9 = 13 Kg/s	Effectiveness of the plant for m_9 = 11 Kg/s
R600	0.9461	0.8944
R500	0.9111	0.9287
R404a	0.8985	0.9418
R141b	0.9187	0.921
R134a	0.8771	0.9647
R11	0.923	0.9168
R12	0.9278	0.912

As from above tabulation, the maximum mass flow of water as obtained from R600 is most favorable as it involves acquiring the greater amount of energy with the maximum flow of water at the constant end conditions. This maximum flow will though be achieved at the expense of additional compressor power as presented in Table 6. Fig. 3 illustrates the effect of the mass flow of inlet water stream on maximum performance.

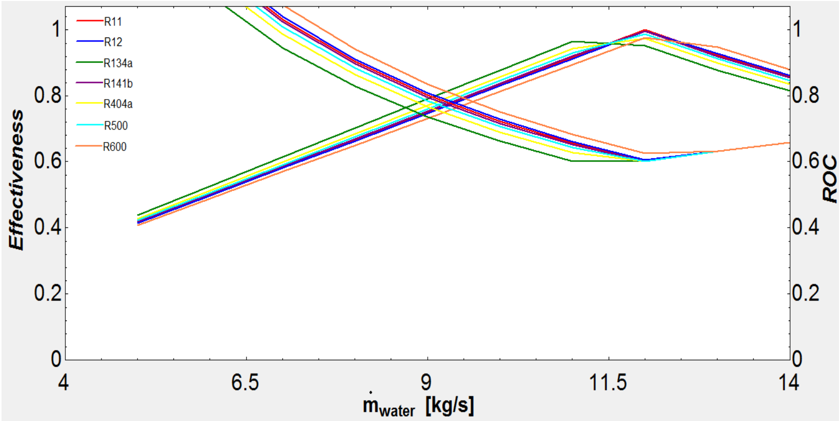

Fig. 3. Impact of mass flow of inlet water stream on the effectiveness and ROC of the plant for r = 3.5 and T9 = 10 C.

For criterion (III), the temperature of the inlet stream of water is investigated for the maximum performance, and the results are enumerated in Table 8. As from the theory presented for the criterion (II), the variation in the optimal temperature of the inlet water stream is such that the heat loss in the condensers is effectively utilized to an extent as required for the application. Moreover, the values of Table 6 remain valid for the current criterion as the tabulated values were calculated relative to the fixed compression ratio and states of the system at different sites, which continuous to be the same for the present measure.

Table 8. Optimal temperature of water stream for r = 3.5 and m_9 = 12 Kg/s

Refrigerants	Optimal temperature
R600	8.507
R500	10.78
R404a	11.6
R141b	10.28
R134a	12.98
R11	10.01
R12	9.694

The minimum optimal temperature of water stream as can be attained by R600 is most viable, as it involves acquiring the greater amount of energy with the least temperature. Fig. 4 displays the plot of maximum performance against the temperature of inlet water stream.

The order of refrigerants for maximum performance of the plant as per criterion (II) and (III) are tabulated in Table 9. The results of this tabulation are in accordance with the concept that the mass flow of refrigerant is inversely related to the change in specific energies of the same across the condenser. For the high value of this change, the mass flow is a small quantity. The product of mass flow and this change is, however, the actual energy that the water stream can absorb in its first stage. Hence the higher value of this allows the stream to absorb more energy, as a result of which it can either sustain low temperature or more mass flow of inlet water stream. The enthalpy values at different nodes of the heat pump are summarized in Table 10 to understand the above postulates better.

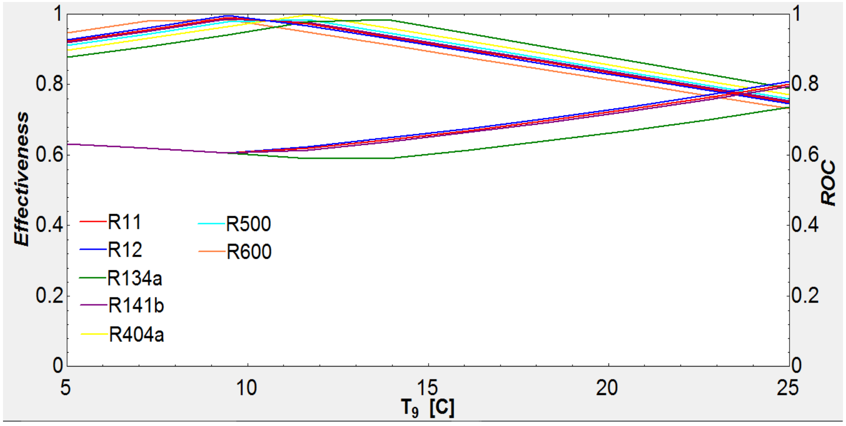

Fig. 4. Impact of temperature of inlet water stream on the effectiveness and ROC of the plant for r = 3.5 and m_9 = 12 Kg/s.

Table 9. Feasibility of the refrigerants in ascending order relative to the efficient performance of the refrigerants as per criterion (II) and (III).

Order of performance	Refrigerants
# 1	R600
# 2	R12
# 3	R11
# 4	R141b
# 5	R500
# 6	R404a
# 7	R134a

Table 10. Specific enthalpy values, mass flow of refrigerant and specific enthalpy changes as per criterion (II) and (III).

Parameters	R600	R500	R404a	R141b	R134a	R11	R12
h_1	579.4	205.7	338.7	296.9	233	233.2	173
h_2	627.6	231.3	363.1	323.9	258.3	255.8	194.3
h_3	280.3	36.33	173.9	120.7	54.57	86.82	36.64
h_4	280.3	36.33	173.9	120.7	54.57	86.82	36.64
m_{ref}	4.012	7.087	7.279	6.81	6.724	8.197	8.798
$h_2 - h_1$	48.2	25.6	24.4	27	25.3	22.6	21.3
$h_2 - h_3$	347.3	194.97	189.2	203.2	203.73	168.98	157.66
$m_{ref} * (h_2 - h_3)$	1393.37	1381.75	1377.19	1383.8	1369.88	1385.13	1387.1

5. Conclusion:

In this study, a novel water heating scheme has been presented. This scheme exploits the waste heat of condenser to improve the utilization factor of the available power production setup. The thermodynamic analysis of the proposed scheme is also performed to account for the viability of the performance parameters, called the effectiveness and ROC, for the set of fixed conditions. The refrigerants selected for the analysis includes R11, R12, R134a, R141b, R404a, R500 and R600. The thermodynamic analysis was split into three criteria, criterion (I) was accounted to determine the optimal compression ratio of the heat pump for the set of refrigerants. The criterion (II) and (III) were performed to determine the optimal mass flow and inlet temperature of cold water stream respectively, for the fixed compression ratio. The optimal range of compression ratio of heat pump was found as 3.35-3.85. The optimal mass flow and temperature of inlet cold water stream were calculated as 11-12.5 Kg/s and 08-13 ^{O}C respectively for the set of refrigerants. Furthermore, R600 shown the maximum performance for the given scheme and R134a was the least efficient. It was also shown that an attempt to increase the compression ratio, mass flow and inlet temperature of water stream relative to the optimal range would lower the effectiveness of the scheme.

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Nomenclature	
r	Compression ratio of Heat pump
x	dryness fraction
P	Pressure (KPa)
T	Temperature (^{O}C)
h	Specific enthalpy (KJ/Kg)
s	Specific entropy (KJ/Kg ^{O}C)
V	Specific Volume (Kg/m^3)
Q	Heat flow rate (KW)
W	Power (KW)
m or ṁ	mass flow rates (Kg/s)
Subscripts	
HP	Heat pump
SP	Power unit
cnd	Condenser
comp	Compressor of Heat pump
evp	Evaporator of Heat pump
Turb	Turbine of Power unit
ref	Refrigerant of Heat pump
Greek letter	
η	Isentropic efficiency

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