Power Plant Waste Heat Recovery for Household Heating Using Heat Pumps

Dosa, Ion

University of Petrosani

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Dosa Ion
Lecturer, Ph. D. at the
University of Petrosani
Romania

1. INTRODUCTION

As known, the major source of loss for a thermal power plant is the heat rejected by the condenser which can be up to 48.9% of thermal energy at turbine inlet as presented in [1] and other papers provide values between 33% [2] and 45% [3] of fuel input depending on technology used for power generation.

Heat rejected by the cooling water of condenser will end up in the environment through the use of a cooling tower or directly in rivers depending on the cooling scheme employed (open loop or closed loop cooling), producing thermal pollution.

In paper [3] energy balance for a 210 MW unit was completed. This unit, functioning at 94% load, rejected 274.1 MW through the condenser cooling water, representing 46.63% of the overall heat input. This is a huge amount of heat, and heat recovery must be considered.

In recent years heat recovery from low temperature sources is a constant area of interest as highlighted in informative articles of various organizations [4] and papers published [5][6][7][8].

2. PROBLEM FORMULATION

Condenser is a necessary component of the steam cycle as it converts the used steam into water for boiler feed water

A condenser serves the following purposes:

- maintains a very low back pressure on the exhaust side of the turbine. As a result, the steam expands to a greater extent and available heat energy increases. Lower cooling water temperatures assure a low condenser pressure;

- condensate is free from impurities and is used as feed water for the steam generator. A thermal power plant using regenerative Rankine cycle with closed feed water heating, will work at higher thermal efficiency and power generation will be increased.

In order to use cooling water in closed loop, warm water at the outlet of condenser must be cooled, usually using cooling towers. As laws become more restrictive, for example Clean Water Act requires the regulation of water thermal discharge from cooling water systems in order to protect aquatic wildlife [9], cooling of discharged water must be considered.

Conclusively, given the importance of condenser, cooling it is mandatory, so heat rejection is inevitable, but recovering heat from cooling water can turn heat loss in something useful.

From the above it follows than any solution of heat recovery from condenser cooling water should ensure at the same time cooling of outlet water for reuse or discharge into environment. In order to design a flexible and easy to implement solution, the technology involved should be a mature one, but used in an innovative way.

3. PROPOSED SOLUTION

Since the analyzed secondary heat source is one of low temperature, between 15 °C in winter and 30 °C in summer, next temperature levels we can think of is represented by temperatures needed for household heating and hot water that are in a range from 45 to 65 °C. Technology to produce 65 °C water from a heat source with only 5 °C also exist and is a very well known and widely used one, namely the heat pump technology.

All heat pumps perform the same three functions: receipt of heat from the waste-heat source; increase of the waste-heat temperature; delivery of the useful heat at the elevated temperature. Waste heat is delivered to the heat-pump evaporator in which the heat-pump working fluid is vaporized. In case of water source heat pumps, the heat for evaporator is drawn from surface water or pond using loops. As known, in ponds even if their surface is icy, temperature at the bottom can be only as low as 4 °C.

Given the above, the proposed solution presented in Fig. 1, is to use heat pump to recover heat from condenser cooling water while achieving its cooling.

The heat at elevated temperature can be used for household heating or similar. As the amount of available heat is huge, several heat consumers can be fed, depending on their heating requirements.

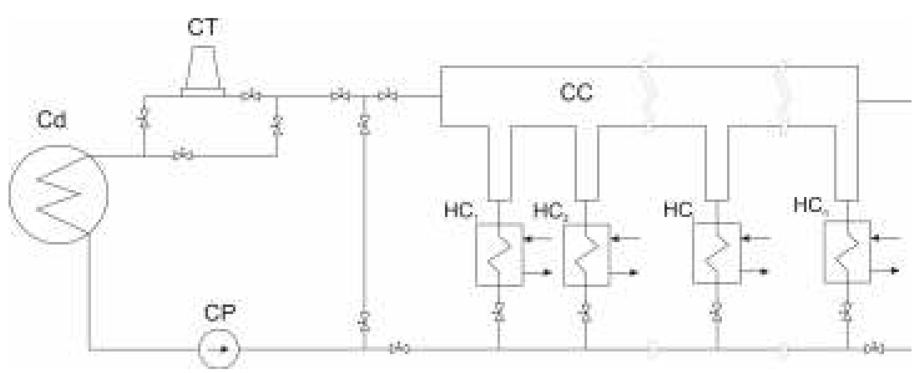

Fig. 1
Heat recovery and water cooling system.
Cd – condenser, CP – circulation pump, CT – cooling tower, CC – cooling canal, HC_i – heat consumer (evaporator of heat pump)

The evaporator can be a surface heat exchanger similar to those employed coolers.

Worst case scenario for the solution proposed is that no heat is required. In that case, a cooling canal can be used that rejects heat through a combination of convection and radiation heat transfer as well as evaporation of canal water [10].

A cooling canal can be an alternative way of heat rejection for newly built power plants, with nearly no negative visual impact since it can be integrated in the natural environment as a small man-made river.

Solution of cooling canal was suggested by the existence of several heat consumers who are likely to be situated at considerable distance from power plant, so large surface area required for cooling canal can be easily achieved given the length of them.

For existing power plants, cooling towers can be integrated in the recovery installation as backup in case that climate conditions prevents cooling canal rejecting heat properly and no heat is required.

4. CASE STUDY

In order to study the effect of heat recovery using power plant waste heat as heat source for heat pumps, data available for a 210 MW unit of Deva thermal power plant is used. In this case cooling water is taken from Maros River.

The heat pumps suitable for harnessing energy resources with low thermal potential are mechanical vapor compression, water-water type [11][12]. They can use waste heat from water with temperatures under 40 °C, and deliver heat at temperatures between 50 to 80 °C.

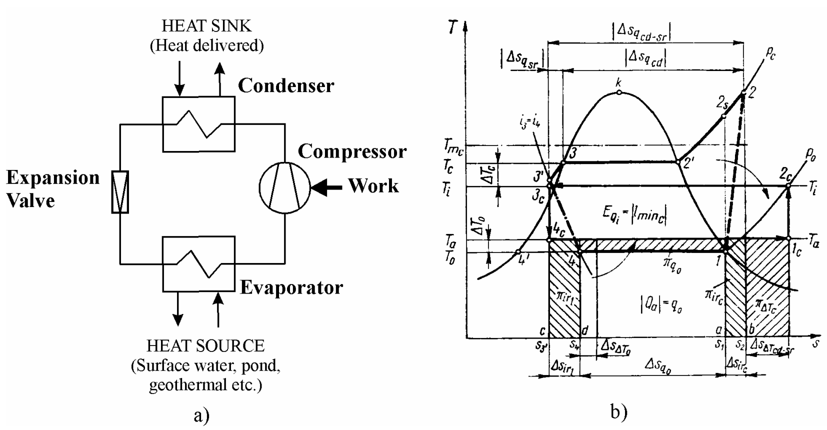

Fig. 2.
Schematic of Mechanically Driven Heat Pump (a) and Temperature–Entropy diagram of the vapor-compression cycle (b).

First climatic conditions for Deva region should be determined, and in Table 1, the average values of climatic factors are presented (for year 2004).

Table 1
Average values of climatic factors

Month	Temperatures of water in Maros River [ºC]	Water flow rate [$m^3 \cdot h^{-1}$]	Atmospheric temperature [ºC]	Atmospheric pressure [mmHg]	Air relative humidity [%]
January	7.82	601,200	-1.66	741.7	87

February	6.29	320,400	1.10	738.1	85
March	7.94	522,000	5.68	737.8	75
April	13.82	802,800	14.26	741.1	59
May	17.62	558,000	15.91	741.8	68
June	20.82	270,720	19.77	738.9	74
July	23.60	249,840	21.90	740.1	74
August	23.52	234,720	22.37	741.8	77
September	19.98	183,960	18.19	743.0	68
October	13.74	360,000	11.06	740.9	82
November	8.72	333,000	6.58	740.7	91
December	5.81	248,040	1.96	737.0	86

Data of average atmospheric temperature in Table 1 suggest that household heating can be considered for months starting from October until April.

In order to highlight the amount of available heat, it is assumed that heat consumers are typical households. Most companies that sell central heating equipment indicates that a 28 kW heating system that could meet the needs for heating and hot water for a property having a gross built area of 150 m^2, two bathrooms and more then 3 residents. Therefore this value will be used in further calculations.

Data regarding condenser cooling water outlet temperature are presented in Table 2 and are computed based on the following assumptions:

- the cooling water used for condenser is taken directly from the river, in an open loop cooling system;
- data for cooling water temperature at outlet are computed for a turbine load of 180 MW, main steam parameters of 130 bar at 545 °C and a flow rate of 540 $t \cdot h^{-1}$, to which corresponds the exhaust flow rate of 400 $t \cdot h^{-1}$.

Equations required for calculations can be found in literature [13][14], and they are well known, therefore are not presented here.

In order to compare the efficiency of using waste heat as heat source, a heat pump using heat source a body of water is calculated first.

Data required for calculation are: Q_i=28 kW – required heat delivery, T_i=65 °C – temperature of delivered hot water, T_a=5 °C – ambient temperature, ΔT_c=5 °C – temperature difference required for heat transfer in condenser (heat delivery), ΔT_0= 5 °C – temperature difference required for heat transfer in evaporator, ΔT_{sr}=10 °C – temperature difference for sub-cooling, η_{em}= 0.9 mechanical efficiency, employed refrigerant R717 (ammonia).

Table 2
Condenser data for open loop cooling operating system

Month	Inlet water temperature [ºC]	Outlet water temperature [ºC]	Overall heat transfer coefficient $[W \cdot (m^2 \cdot {}^oC)^{-1}]$	Saturation temperature [ºC]	Condenser pressure [bar]
January	7,82	16,99	2224,5	26,24	0,0341
February	6,29	15,46	2125,3	25,31	0,0323

March	7,94	17,10	2231,1	26,31	0,0343
April	13,82	22,96	2574,1	30,47	0,0436
May	17,62	26,75	2762,9	33,50	0,0517
June	20,82	29,93	2902,4	36,19	0,0599
July	23,60	32,70	3009,1	38,61	0,068
August	23,52	32,62	3006,2	38,54	0,0678
September	19,98	29,10	2867,5	35,48	0,0576
October	13,74	22,88	2569,8	30,41	0,0435
November	8,72	17,92	2283,4	26,84	0,0353
December	5,81	14,99	2094	25,05	0,0318

In addition to data calculated for characteristic points of reversed cycle, exergy analysis of heat pump cycle is carried out in order to highlight losses and their variation in different working conditions. Results are presented in Table 3 and plotted in Fig. 3 and 4.

For heat pump using heat source waste water, data required for calculation are the same, excepting the ambient temperature that will be the average temperature in the evaporator, taking into account that at outlet of evaporator water temperature must be over 5 °C to prevent freezing, and the inlet temperature is the condenser outlet temperature.

The theoretical number of households that can be supplied with heat and hot water are calculated from the energy balance of evaporator for 2% heat loss.

Data in Table 3 and Fig. 3 and 4 highlights some aspects of using condenser cooling water as heat source for a heat pump.

Table 3
Heat pump working at different heat source temperatures

Nom.	Reference	October	November	December
Condenser outlet temperature [°C]		22,88	17,92	14,99
Heat source temperature [°C]	5	13,94	11,46	10,00
Cycle compression work **l** [$kJ \cdot kg^{-1}$]	439,643	351,655	374,231	387,359
Work supplied to compressor $\mathbf{P_e}$ [kW]	9,861	8,361	8,763	8,992
Evaporator heat transferred $\mathbf{q_0}$ [$kJ \cdot kg^{-1}$]	947,413	956,883	954,378	952,859
Condenser heat transferred $\mathbf{q_c}$ [$kJ \cdot kg^{-1}$]	1.387,06	1.308,54	1.328,61	1.340,22
Ideal Carnot COP $\boldsymbol{\mu_C}$	5,640	6,620	6,320	6,150
Theoretical COP $\boldsymbol{\mu}$	3,160	3,720	3,550	3,460
Practical COP $\boldsymbol{\mu_e}$	2,840	3,350	3,190	3,110
LOSS due to irreversibility of:				
Compression $\boldsymbol{\pi_{irc}}$ [$kJ \cdot kg^{-1}$]	57,490	42,970	46,850	48,610
Expansion $\boldsymbol{\pi_{irl}}$ [$kJ \cdot kg^{-1}$]	33,690	25,060	27,320	28,690
Heat transfer in evaporator $\boldsymbol{\pi_{q0}}$ [$kJ \cdot kg^{-1}$]	17,340	16,960	17,070	17,130
Heat transfer in condenser $\boldsymbol{\pi_{\Delta Tc}}$ [$kJ \cdot kg^{-1}$]	85,320	69,630	73,110	75,380
Work of Ideal Carnot cycle $\mathbf{l_{minC}}$ [$kJ \cdot kg^{-1}$]	246,114	197,587	210,361	217,986
Exergetic efficiency $\boldsymbol{\eta_E}$	**55,980**	**56,190**	**56,210**	**56,270**
Duty of evaporator $\mathbf{Q_0}$ [kW]		20,475	20,113	19,907
Required water flow rate $\mathbf{m_{apa}}$ [$t \cdot h^{-1}$]		1,005	1,366	1,748

Households that could be supplied		24.881	18.303	**14.298**

Table 3 continued	January	February	March	April
Condenser outlet temperature °C	16,99	15,46	17,1	22,96
Heat source temperature	11,00	10,23	11,05	13,98
Cycle compression work **l** [kJ·kg^{-1}]	379,356	384,961	378,152	351,556
Work supplied to compressor $\mathbf{P_e}$ [kW]	8,852	8,950	8,832	8,359
Evaporator heat transferred $\mathbf{q_0}$ [kJ·kg^{-1}]	953,903	953,100	953,954	956,923
Condenser heat transferred $\mathbf{q_c}$ [kJ·kg^{-1}]	1.333,26	1.338,06	1.332,11	1.308,48
Ideal Carnot COP $\boldsymbol{\mu_C}$	6,260	6,170	6,260	6,630
Theoretical COP $\boldsymbol{\mu}$	3,510	3,480	3,520	3,720
Practical COP $\boldsymbol{\mu_e}$	3,160	3,130	3,170	3,350
LOSS due to irreversibility of:				
Compression $\boldsymbol{\pi_{irc}}$ [kJ·kg^{-1}]	48,020	48,040	47,350	43,400
Expansion $\boldsymbol{\pi_{irl}}$ [kJ·kg^{-1}]	27,750	28,480	27,700	25,020
Heat transfer in evaporator $\boldsymbol{\pi_{q0}}$ [kJ·kg^{-1}]	17,090	17,120	17,080	16,960
Heat transfer in condenser $\boldsymbol{\pi_{\Delta Tc}}$ [kJ·kg^{-1}]	74,060	75,050	73,950	69,300
Work of Ideal Carnot cycle $\mathbf{l_{minC}}$ [kJ·kg^{-1}]	212,911	216,725	212,530	197,423
Exergetic efficiency $\boldsymbol{\eta_E}$	**56,120**	**56,300**	**56,200**	**56,160**
Duty of evaporator $\mathbf{Q_0}$ [kW]	20,033	19,944	20,051	20,477
Required water flow rate $\mathbf{m_{apa}}$ [t·h^{-1}]	1,466	1,673	1,454	1,000
Households that could be supplied	17.053	14.943	17.194	24.991

For the reference case (T_a= 5 °C), exergetic efficiency η_E= 55.98 %, smaller than other values. Maximum exergetic efficiency for temperature ranges considered in calculations is 56.30 % but variation for this parameter is rather small, 0.32%.

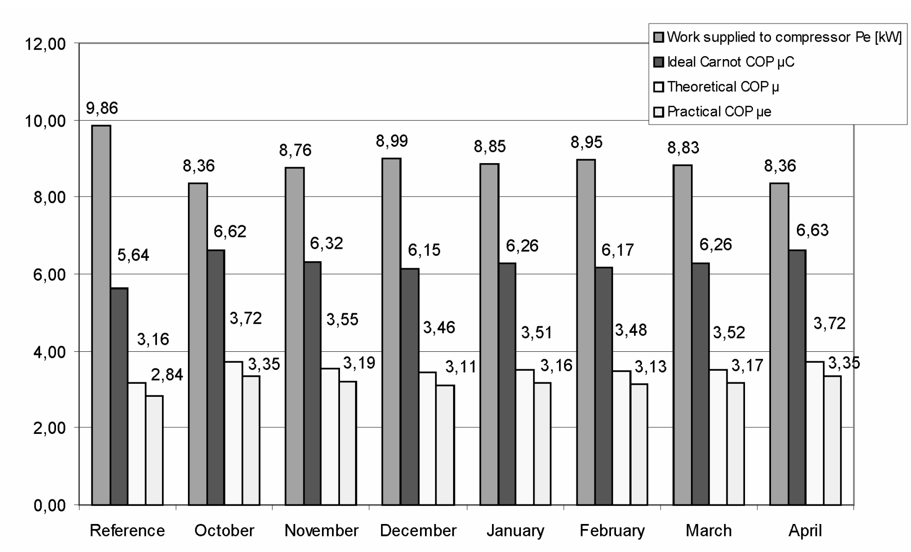

Fig. 3
Work and Coefficients of Performance at different heat source temperatures

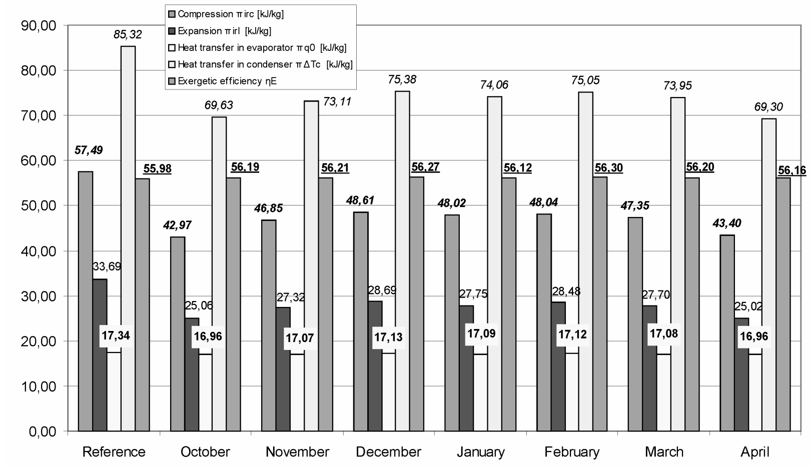

Fig. 4
Losses and exergetic efficiency (underlined)

The highest amount of heat 956.923 kJ·kg^{-1}, delivered from heat source to heat pump in the evaporator, corresponds to highest temperature of heat source which is 22.96 °C in April. As expected this corresponds to the highest practical COP which is 3.35 comparative to 2.84 for the reference case. Accordingly the smallest amount of work that needs to be supplied to the heat pump is 8.359 kW comparative to 9.861 kW for the reference case, resulting 1.502 kW savings.

For the heating period from October to April the worst case scenario is for the temperatures in December when temperature of water at the outlet of condenser was minimal 14.99 °C. Even in this case, practical COP is 3.11 higher than 2.84 for reference case, and the amount of savings with the work supplied to heat pump is 0.869 kW.

5. CONCLUSIONS

A large amount of heat is wasted by the condenser cooling water, and urgent measures must be taken in order to recover that heat.

Analyzing data in Table 3 and Fig. 3 and 4, some conclusions can be drawn regarding the proposed heat recovery system:

- using waste water as heat source for heat pumps can raise heat pump COP and lower the work that needs to be supplied, resulting energy savings;
- exergetic efficiency of heat pump remains basically the same, around 65% pointing that temperature of heat source has no major influence on overall exergy losses in heat pump;
- the amount of heat that could be recovered this way, is highlighted by the number of households that can be supplied, even in worst case scenario is 14.298;
- in closed loop operation, the temperature of returned water at the steam condenser inlet can be theoretically 5 °C (temperature at the outlet of heat pump evaporator) with all the benefits resulting from proper condenser cooling.

In order to apply proposed heat recovery system heat consumers must be found with constant demand of heat all year round (cafeterias, swimming pools, hospitals etc.), as high COP and savings are achieved in April and October, corresponding to higher water and atmospheric temperatures, and lower demand of heating.

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