Performance Analysis and Simulation of a Combined Rankine-Absorption

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ABSTRACT:

Combined cooling, heating and power (CCHP) systems including technologies, provide an alternative for the world to meet and solve energy related problems. A simulation of a combined Rankine-absorption cycle is presented in which the boiler flue gases of the Rankine cycle are used as an energy source to the absorption cycle that is used to cool the Rankine cycle condenser to offset the performance degradation of the Rankine cycle when operating under high ambient temperature conditions. The results indicate that there is insufficient energy in the flue gases for a single effect absorption unit to improve the overall cycle performance to that of an ISA day case. The analysis concludes by indicating the level of coefficient of performance (COP) that would be needed to achieve the desired objective.

Notation

A	coefficient in equation
A_f	heat exchanger face area
As	heat transfer surface area
B	coefficient in equation
Cf	friction coefficient
c_p	specific heat at constant pressure
C	$C2/C_1$
C_1	constant in Eq. (5)
C_2	constant in Eq. (6)

h convective heat transfer coefficient
j Colburn factor
l air side path length
m exponent
n exponent
nf number of fins per unit length
p power ratio parameter P_{fan}/ P_{net}
p pressure
P_g gross power Rankine cycle
P_f power requirement of the flue gas fan
P_{net} net power output of the total system
Pr Prandtl number
Qh heat content of the flue gases
Qgen additional heat supplied to the refrigeration unit generator
Q+ heat added to the cycle at the boiler
Q_ heat rejected by the cycle at the condenser
Re Reynolds number
t temperature (°C)
T absolute temperature (K)
U cooling air velocity
v heat exchanger volume
εHE flue gas exchanger effectiveness
ηc Carnot efficiency of the refrigeration unit
ηf fin efficiency
ηth Rankine cycle thermal efficiency
p density
τ shear stress
v kinematic viscosity

Subscripts

abs absorber
con condenser
evap evaporator
gen generator
ex flue gas
i inlet
o outlet
1 ambient conditions

Superscript

/ dimensionless parameters

1. Introduction

Combined cooling, heating and power (CCHP) is derived from combined heat and power (CHP), also it called cogeneration [1]. It is well known that the performance of a Rankine cycle is degraded as the ambient temperature increases. Condensing plants commonly operate more effectively in the winter than in summer because of the lower temperature of the condenser coolant. If the condenser temperature is lowered, the heat rejected is less, the work greater and therefore the efficiency will be increased. A particular plant reports a 10% greater power production in winter (condenser pressure 0.033 bar) than in summer (condenser pressure 0.087 bar). To condense the steam a natural available sink is used and we must be satisfied with a condenser temperature somewhat (15-20 °C) above that of the condenser coolant be it ambient air or water from rivers or lakes. The best direct water cooled condensers are very well designed thermally so it is not expected to improve the thermal

performance of this part of the cycle but efforts at improvements are continually being made because in very large plants even a small improvement involves a significant financial saving. The majority of steam plant in use today has been designed for operation in a temperate climate thus if such a plant is sited in a hot climate it is customary to accept the down rating of the cycle and to use it in a combined cycle mode, with a desalination plant for instance, to recover some of the lost efficiency.

The purpose of this paper is to explore the possibility of utilizing the boiler flue gases of a Rankine cycle to drive an absorption refrigeration unit that is in place to reduce the condenser coolant temperature to its design level and thus offset the overall cycle performance degradation due to a high ambient temperature.

The concept of combined cycles is not new but the combination of power producing cycle with absorption refrigeration plants has received little attention in the past. A search through the literature has brought to light several plants consisting of Diesel or Brayton cycles combined with refrigeration units either absorption or vapour compression [1-6] but there is no evidence of the combination being made with a Rankine cycle and absorption refrigeration unit. This may be because the low coefficient of performance (COP) of an absorption unit excluded it from consideration or it may be that the detailed insight into the operating characteristics of an absorption unit that is necessary to perform an analysis of such a combination, is generally not available in the literature although manufacturers of such units will have this data to hand. This work has been possible because a detailed study of the operating performance of an absorption unit with respect to ambient day temperatures has been undertaken by Alaktiwi and reported in [7].

2. Description of the complete system

A schematic view of the overall system is shown in Fig. 1. It consists of a basic Rankine cycle with boiler, expansion unit and condenser. To this has been added an absorption refrigeration unit between the boiler flue gasses and the condenser coolant. The absorption unit is in place to utilize the waste heat in the boiler flue gasses to cool the condenser coolant and thus offset the

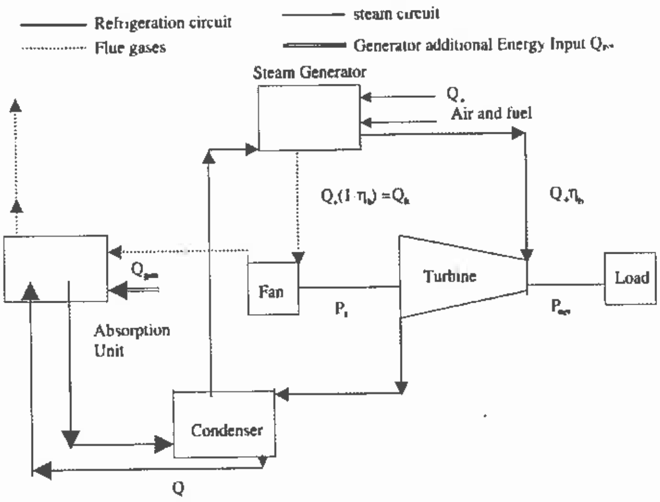


Fig. I. Schematic diagram of the combined Rankine-absorption cycle.

A fan, driven by the Rankine cycle, has been provided to overcome the flow restriction produced by the necessary heat exchangers in the flue gas line. Additional energy can be provided at the absorption unit generator when the COP of the refrigeration unit and the energy available in the flue gasses are insufficient to provide the necessary cooling requirement. The output power of the Rankine cycle is thus decreased by the requirement to drive the flue gas fan, this and the additional energy input must be taken into account when the overall efficiency of the complete system is considered.

3. Characterizing the absorption unit

bromide absorption unit has been undertaken by Alaktiwi [7]. This study provides the data that is used to integrate the absorption unit into the overall system proposed here. It is well known that the COP of an absorption unit is adversely influenced by high ambient temperatures through the impact it has on the evaporator temperature. This study has shown that the cop is independent of generator temperature if this is high enough. It has also shown that the practice of describing the COP as being the product of the ideal cycle COP and the second law efficiency is sufficiently accurate for most purposes. It is then possible to account for the COP by referring to the equivalent Carnot cycle temperatures within the absorption unit and the second law efficiency.

$$COP_{ideal} = \frac{T_{evap}}{T_{gen}}\left(\frac{T_{gen}-T_l}{T_l-T_{evap}}\right) \quad (1)$$

The influence of ambient temperature can then be determined by including the effects this has on the evaporator temperature in the ideal system and on the second law efficiency. These studies indicated, for the absorption unit examined, the evaporator temperature was related to the ambient temperature by the following equation:

$$T_{evap}=\frac{t_l}{2}-0.8+273 \quad (2)$$

And the second law efficiency could be expressed in terms of the ambient temperature as shown below:

$$=0.289-0.004tl \quad (3)$$

It can be seen that the effect of increased ambient temperature will be twofold. Firstly it will increase the absorber temperature which will lower the cop and secondly this will result in the condenser temperature rising which will reduce the efficiency of the Rankine cycle. Thus whilst the capacity of the absorption unit will decrease the heat removal rate required at the condenser will increase.

4. Characterising the absorption unit generator heat exchanger

Optimisations of heat exchanger based systems usually concentrate upon the optimisation of the heat exchanger when all other aspects of the system are optimised or visa versa. It is very unusual because of the large number of co-related parameters to be able to fully optimise all the system variables at anyone time. Indeed when optimising a heat exchanger it is possible to optimise single parameters individually, to indicate the direction of beneficial changes, but it should not be expected that selection of individual optimised parameters will lead to a fully optimised design, Smith [8]. It is also the case that heat exchanger manufacturer can produce only a limited range of fin-tube configurations in terms of tube diameter and spacing, fin type, fin spacing and thickness, thus any optimisation of these parameters is of academic interest to them. The optimisation method adopted here is similar to that of Lau et al. [9], It does not focus on the internal structure of the heat exchanger but is concerned with the interaction of the heat exchanger with the basic thermodynamic cycle, in this case the Rankine cycle. To this end the optimised heat exchanger is related through non-dimensional parameters (defined below) to a reference heat exchanger that satisfies the required heat transfer duty but may be unsatisfactory in other aspects that are related to the overall plant performance.

In analysing the gas side of the heat exchanger it is assumed that the flow paths are in the form of rectangular passages of large aspect ratio, the extended surfaces are thin compared to the spacing between the fins and the gas velocity is constant throughout the length of the path. The internal surface temperatures are at the temperature of the internal working fluid and only frictional work is expended in moving the flue gases through the heat exchanger. If these conditions are met then the equations shown below can be considered to describe the thermo-fluid state with some confidence.

The fan power required to provide the flue gases to the heat exchanger is described by the following equation:

$$P_f = A_s U_T = \left(c_f \left(\frac{pU^2}{2} \right) A_s U \right) \qquad (4)$$

and the friction coefficient and Colburn factor are related to the flow Reynolds number by the following general expressions:

$$c_f = C_1 Re^{-m} \qquad (5)$$

$$j = C_2 \, \mathrm{Re}^{-n} \qquad (6)$$

Where

$$\mathrm{Re} = \left(\left(\frac{4A_f l}{A_s} \right) \frac{U}{v} \right) \qquad (7)$$

If it is assumed that $m = \mathrm{n}$ and $(C2/C_1) = C$ then

$$c_f = \left(\left(\frac{h}{pUc_p} \right) \frac{\mathrm{Pr}^{\frac{2}{3}}}{C} \right) \qquad (8)$$

The thermal efficiency of the power cycle is usually defined as the ratio of the gross power output to the rate of heat input. This relationship can be rearranged to express the ratio of the rate of heat rejection by the power cycle to the power output as shown below:

$$\frac{Q_{-}}{P_g} = \left(\frac{1-\eta_{th}}{\eta_{th}}\right) \quad (9)$$

In the above equation ηth is the thermal efficiency of the Rankine cycle determined from the boiler energy input not the heating value of the fuel. The overall efficiency of the steam cycle is equal to this efficiency multiplied by the boiler efficiency. The rate of heat addition to the generator heat exchanger is related to the mass flow rate and the temperature change of the flue gases and the COP of the absorption unit

$$Q_{-} = A_f \rho U c_p \left(T_{exi} - T_{exo}\right)\left(\frac{1}{COP}\right) \quad \textbf{(10)}$$

The net useful power produced by the Rankine cycle is the gross power out-put minus the power required to operate the flue gas fan, or,

$$P_{net} = P_g - P_f \quad \textbf{(11)}$$

This then leads to the power ratio parameter p defined as shown below:

$$p = \frac{p_f}{P_{net}} \quad \textbf{(12)}$$

Introducing the non-dimensional velocity *UI* as U / U_{ref} where :

$$U_{ref} \left(Cc_p T_{exi} \Pr^{\frac{2}{3}}\right)^{\frac{1}{2}} \quad \textbf{(13)}$$

and using Eqs. (4)-(12) leads to the following expression:

$$P = \frac{U'^2 f(T_{exo})}{(2-\varepsilon_{HE})COP U'^2 f(T_{exi})} \quad (14)$$

where

$$f\left(T_{xi}\right)=\frac{1-\eta_{th}}{\eta_{th}\left(1-\frac{T_g}{T_{exi}}\right)} \tag{15}$$

The net efficiency of the Rankine cycle can be described as shown in Eq. (15).

$$\eta_{net}=\frac{\eta_{th}}{1+p} \tag{16}$$

The above equation can also be expressed as shown below:

$$\eta_{net}=\eta_{th}-\frac{U^2(1-\eta_{th})}{COP(2-\varepsilon_{HE})\varepsilon_{HE}\left(1-\frac{T_{gen}}{T_{exi}}\right)} \tag{17}$$

In the above equation εHE is the effectiveness of the flue gas heat exchanger.

5. Linking the absorption unit to the Rankine cycle

The energy to drive the absorption unit is obtained via a heat exchanger from the boiler flue gasses. This can be related to the energy input to the boiler and the boiler efficiency ηb

$$Q_h = Q_+(1-\eta_b)$$

This can also be equated to the cooling requirement of the condenser. Under the majority of operating conditions the energy available in the flue gasses will be insufficient to provide the energy requirement of the evaporator. This will depend upon the boiler efficiency, the effectiveness of the heat exchanger and the COP of the absorption unit. In this event it will be necessary to provide some additional energy, Qgen, which can be expressed as a proportion of the heat input y.

$$y = \frac{Q_{gen}}{Q_+} \tag{19}$$

to produce

$$Q_+(1-\eta_b + y)\varepsilon_{HE}COP = (1-\eta_{th})Q_h \tag{20}$$

The limiting condition or the minimum value of COP when no additional energy is required will be specified by the following requirement:

$$COP \geq \frac{(1-\eta_{th})\eta_b}{(1-\eta_b)\varepsilon_{HE}} \tag{21}$$

Alternatively the value of y can be determined from the following equation

$$y = \eta_b - 1 + \frac{\eta_b(1-\eta_{th})}{COP_{\varepsilon HE}} \tag{22}$$

The additional energy supply will influence the cycle overall thermal efficiency which will now be as shown below:

$$\eta_o = \frac{P_{net}}{Q_+(1+y)} = \frac{\eta_b \eta_{net}}{1+y} \tag{23}$$

Typical boiler efficiency is between 70% and 80%, the higher figure being appropriate for the case of feed water heating. Substitution of the expression for y in the above equation however shows that the overall efficiency of the combined cycles is independent of the boiler efficiency. Thus in practice it would be possible to offset the cost of a feed water heater against the cost of an absorption refrigeration unit. To proceed further with the analysis requires values of the Rankine cycle thermal efficiency 11th' The Rankine cycle efficiency is a function of boiler pressure, condenser temperature and ambient temperature. For a specified piece of plant this will be known. Data taken from [4] indicate that a Rankine cycle efficiency can be represented by the following equation when the boiler pressure is 34 bar:

$$\eta_{th} = 1.44 - 0.85475\frac{T_{con}}{T_1} \quad (24)$$

The generator of the absorption unit can be considered to be a two-phase heat exchanger in which the water vapour that is the refrigerant is driven out of solution with the lithium bromide. In the case of this type of heat exchanger the effectiveness is a very weak function of NTU. Over the range of NTU from 2.5 to 5 the value of effectiveness will change from 0.917 to 0.993 therefore little error will be produced by assuming that the effectiveness of the flue gas heat exchanger, eHE, is constant and for simplicity it has been given the value of 1.

It is also necessary to specify additional system parameters to complete the analysis. All the assumed values of the input parameters are shown in the following table:

System parameter	Value
Flue gas heat exchanger effectiveness e	0.8
Flue gas temperature at inlet to the heat	300°C
exchanger	1.0
Condenser effectiveness εcon Boiler pressure	34 bar
Reference velocity U_{ref}	220 m/s

6. Discussion of the results

The performance of the combined cycle will be examined in the first instance by considering the individual cycles separately.

The influence of the power take off to drive the flue gas fan is illustrated in Figs. 2 and 3. It can be seen that the power factor, p, is influenced by the ambient temperature and the flue gas velocity.

The largest influence is produced by the gas velocity as a result of increased friction losses in the flue gas heat exchanger. In a practical situation increasing the gas velocity corresponds to reducing the size of the heat exchanger that could result in lower initial costs and lower maintenance costs over the life time of the unit. The influence of the power factor p on the net efficiency of the Rankine cycle is shown in Fig. 3. The net efficiency is that produced when the fan power requirements are taken into consideration. Also shown in this figure is the efficiency of the basic Rankine cycle based on the ambient temperature. The only combined cycle of those examined that performed better than the unmodified cycle is that with U equal to 0.1.

The performance parameters of a single effect absorption unit are shown in Fig. 4. It can be seen that the COP decreases significantly with an increase in ambient temperature. The effect of the low COP is to increase the external additional energy requirement, y, such that the combined cycle overall efficiency falls below that of the unmodified cycle based on ambient day conditions and is significantly worse than the ISA day performance of the unmodified cycle. The COP also has an inverse influence on the power factor p such that at high ambient temperatures the net

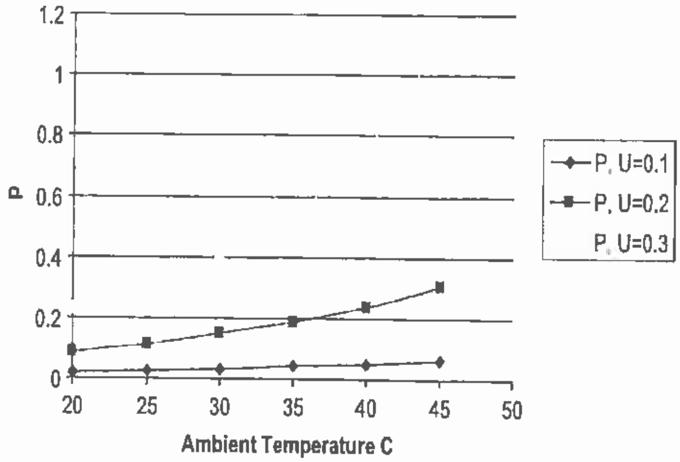


Fig. 2. Variation of power factor, p, with ambient temperature and flue gas velocity.

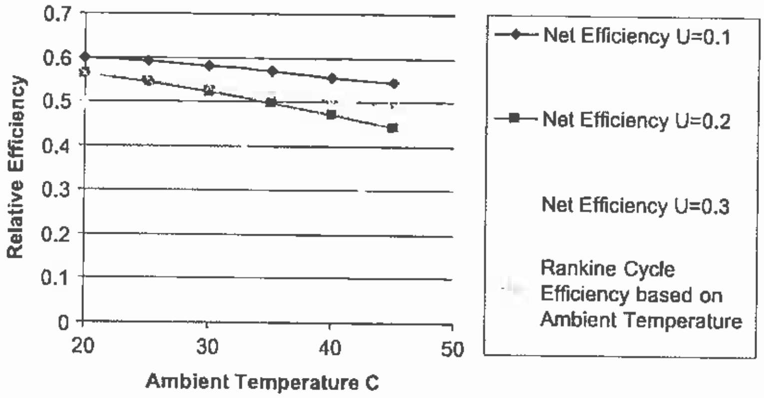


Fig. 3. Variation of Rankine cycle efficiencies with ambient temperature and flue gas velocity.

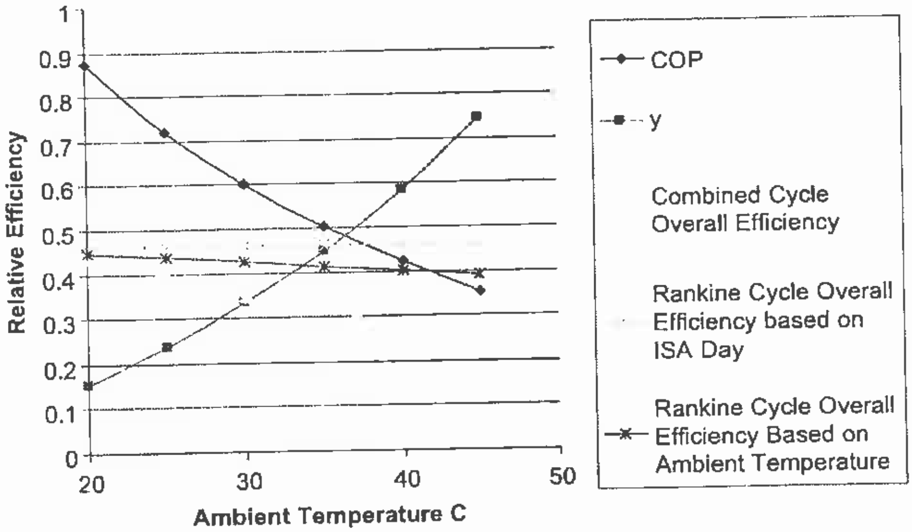


Fig. 4. Variation of system parameters for a single effect absorption refrigeration unit $U = 0.1$.

efficiency of the Rankine cycle falls because of this effect and thus reduces the system overall efficiency (Fig. 5).

The remaining graphs indicate how the performance of the combined cycles is influenced by the performance of the refrigeration unit. They serve to indicate the performance required from a triple or multi effect refrigeration unit to restore the ISA day performance for a specified ambient temperature. This requirement can be met, over the ambient temperature range considered, by a hypothetical unit with a COP of 1.8. This is difficult to achieve with a conventional arrangement but has been achieved with some experimental and multieffect units [10] although at some considerable cost. The importance of COP is illustrated in Fig. 6 that shows the large reduction in the external additional energy requirement with increased COP.

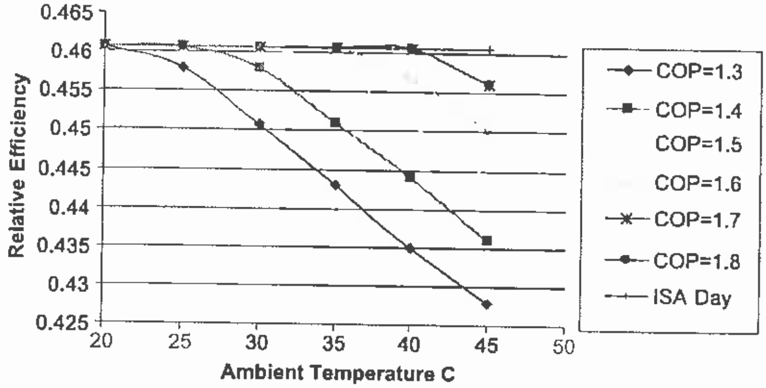
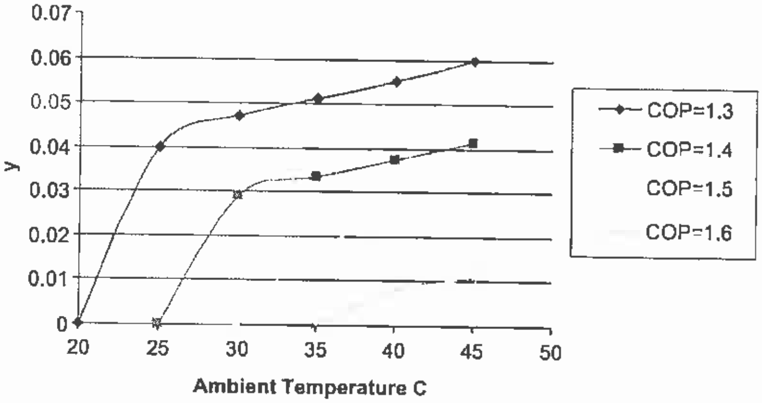


Fig. 5. Influence of COP and ambient temperature on the overall efficiency $U = 0.1$.



7. Conclusions

The analysis presented in this paper indicates that in the case of a single effect absorption refrigeration unit the combined cycles system will not perform in a satisfactory way. The ISA day operating efficiency can be obtained however if it would be possible to utilise a multieffect refrigeration unit with a COP of 1.8 for the ISA day efficiency to be achieved for an ambient temperature of 40°C.

There is clearly a theoretical thermodynamic advantage in utilising an appropriate absorption refrigeration unit as proposed in this work but thermo-economic aspects especially the impact of high initial and subsequent maintenance costs of the additional equipment may make this unattractive. This is the type of problem that cannot be answered by thermodynamic considerations alone as economic costs must also be considered.

The complexity of the interaction of the different elements of the combined system and the benefits of developing higher efficiency refrigeration units is apparent from this simple analysis.

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