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Possible Waste Heat Recovery in the Condenser of a Regenerative Steam Cycle*

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Abstract

The present work is concerned with a new steam cycle proposed as a modification to the normal regenerative steam cycle. A steam ejector and an extra feed water heater are incorporated into the latter cycle. The motive steam of the ejector is bled from the steam turbine. The ejector entrains a portion of the saturated steam exiting the turbine and entering the condenser. Both amounts of motive and entrained steam are raised in pressure and used as heating steam in the first feed water heater. The saturated water getting out of this feed water heater is further heated in the second feed water heater and the cycle is then completed. Thermodynamic analyses of the proposed cycle are developed and used for predicting the characteristic performance of the proposed cycle. The results of the current work show clearly that the thermal efficiency of the proposed cycle is always greater than that of the normal regenerative steam cycle.

Key words: Steam Turbine, Feed Water Heater, Steam Ejector, Regenerative Cycle, Steam Cycle Efficiency.

1. Introduction

Steam power plants are responsible for production of most electric power in the world, and even small increases in thermal efficiency can mean large savings from the fuel requirements and great reduction in pollutants emitted to atmosphere. The steam power plant was invented 150 years ago by Scottish engineer William Rankine (1). The overall efficiency of the early power plants using coal combustion was approximately 15% (2). Early power plants were operated with low inlet steam conditions in the pressure range from 10 to 20 bar and in the temperature range from saturated steam, to slightly superheated steam (100 K).

Numerous efforts have been made to modify the steam power plant cycle and improve its thermal efficiency. The basic idea behind all the modifications is: Increase the average temperature at which heat is transferred to the working fluid (water) in the boiler, or decrease the average temperature at which heat is rejected from the working fluid in the condenser (3). Hence, the efficiency of the steam power plant cycle can be increased by increasing the boiler pressure and the degree of superheat and lowering the condenser pressure. Increasing either boiler pressure or the degree of superheat results in raising the inlet temperature to the turbine. The temperature to which steam can be heated is limited, however, by metallurgical consideration to maximally about 620°C. Any increase in this value depends on improving the present materials or finding new ones that can withstand higher temperatures (3). Accordingly, the increases in boiler pressure and degree of superheat are limited. As for lowering the condenser pressure, there is a lower limit on the

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condenser pressure. It can not be lower than the saturation pressure corresponding to the temperature of the cooling medium.

One of the advances in the steam turbine power plant that has led to improved efficiency was the introduction of the reheat steam turbine [4]. The reheat steam turbine made it possible to raise the main steam pressure without the concern of moisture deposit on the turbine blades. Reheating can improve efficiency by about 5% for single reheat and an additional 2.5% for double reheat [4].

To further improve the overall efficiency and performance of the steam turbine power plant the reheat steam turbine cycle was combined with the addition of feed water preheating [4], also referred to as regeneration. Modern steam turbine power plants utilize multiple feed water heaters that take advantage of turbine extraction steam. The amount of feed water heaters can range from 7 to 10 heaters and it is not uncommon for the addition of feed water heaters to account for up to 13% power plant efficiency improvement.

The current efforts being made for improving steam power plant thermal efficiency centers on improving the present materials and finding new ones that can withstand higher temperatures [5-10]. This entails a great deal of research work and immense amount of investment.

In the present work a new cycle is proposed as a modification to the regenerative steam power plant cycle. In this proposed cycle some of the low pressure steam in the condenser is entrained by a steam ejector whose motive steam is bled from the turbine. Both amounts of motive steam and entrained steam are increased in pressure and used to heat the condensate coming out of the condenser in a feed water heater. Thermodynamic analyses for the proposed cycle are presented in the current work. Based on these analyses the characteristic performance of the proposed cycle is predicted and compared with the normal regenerative cycle.

Nomenclature

h	specific enthalpy	kJ/kg
m_b	steam mass flow rate out of the boiler	kg/s
m_c	steam mass flow rate entering the condenser	kg/s
m_{cc}	condensate rate out of the condenser of the proposed cycle	kg/s
$m_{cc'}$	condensate rate out of the condenser of the normal regenerative cycle	kg/s
m_{ent}	steam mass flow rate entrained from the condenser by the steam ejector	kg/s
$m_{f,1}$	mass flow rate of heating steam in feed water heater (d1)	kg/s
$m_{f,2}$	mass flow rate of heating steam in feed water heater (d2)	kg/s
$m_{f'}$	mass flow rate bled from the turbine for heating in the feed water heater of the normal regenerative cycle	kg/s
m_{ms}	mass flow rate of the ejector motive steam	kg/s
p	Water pressure	bar
Q_b	rate of heat added to water in the boiler	kW
Q_c	rate of heat rejected in the condenser	kW
T	Temperature	^{o}C
W_t	turbine power	kW

Greek letters

Δt_s	degree of superheat	^{o}C
η_c	compression efficiency of the ejector diffuser	-
η_{ent}	entrainment efficiency of the steam ejector	-
η_n	efficiency of the ejector nozzle	-
η_t	turbine efficiency	-
η_{th}	thermal efficiency of the proposed cycle	-

$\eta_{th'}$	thermal efficiency of the normal regenerative steam cycle	-

Subscripts

b	boiler
c	condenser
eo	exit of the feed water heater (d1), Fig. 1
T	turbine
$1,2,$ ..	refers to the steam states (Figures 1,2)

2. Description of the Proposed Cycle

The process configuration of the proposed modified steam cycle and the associated temperature-entropy diagram are shown in Figures 1 and 2, respectively. The numeric data of Fig. 1 correspond to the points 1 to 15 given in Fig. 2. A mass flow rate m_b of elevated pressure and temperature steam outlet of the boiler (f) enters the steam turbine (a) at point 1. m_b expands in the turbine to point 5 where a mass rate of steam $m_{ms}+m_{f,2}$ with state 5 is bled from the turbine, whereas the rest (i.e. $m_c=m_b-m_{ms}-m_{f,2}$) expands further to the low pressure p_c of the condenser. m_{ms} serves as motive steam for the steam ejector (e). On expansion of the motive steam in the nozzle of the ejector, it entrains a rate m_{ent} of saturated steam at p_c (it is equal to p_2 and p_3) from the condenser. The pressure of both amounts of steam, i.e. the heating steam rate $m_{f,1}$ ($m_{ms}+m_{ent}$) of feed water heater (d1), is then raised in the diffuser of the ejector to the pressure p_{eo} (it is equal to p_4, p_{10} and p_{11}). $m_{f,1}$ is used then to heat the condensate rate $m_{cc,}$ coming out of the condenser and pumped to the pressure p_{eo} by pump (c1), in the feed water heater (d1) to the saturation temperature corresponding to the pressure p_{eo}. The saturated water rate m_b-$m_{f,2}$ exiting the feed water heater (d1) is pumped to the pressure p_{12} (equal to p_5 and p_{13}) by pump (c2) and is heated to the saturation temperature corresponding to the pressure p_5 by mixing it with the bled steam rate $m_{f,2}$ in the feed water heater (d2). The outlet rate of saturated water m_b of the feed water heater (d2) is then pumped to the boiler pressure p_b (equal to p_{14}, p_{15} and p_1), where it is heated to superheated state 1 and fed again to the steam turbine.

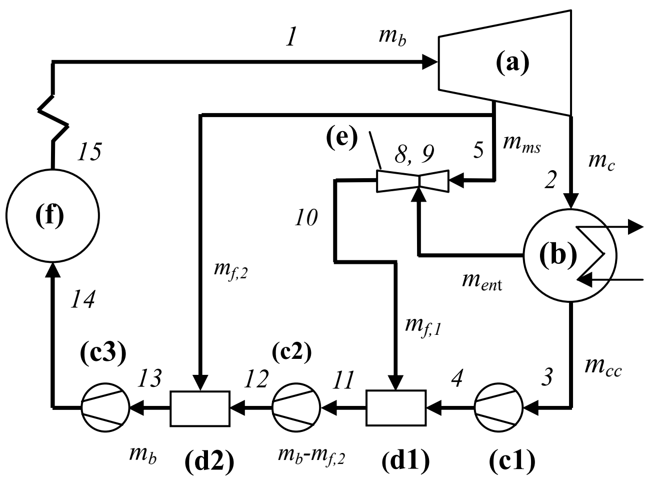

Fig. 1 Shematic diagram of the proposed regenerative steam cycle components

(a)	Steam turbine	(b) Condenser	(c1),(c2),(c3)	Pumps	
(d1),(d2)	Feed water heaters	(e) Steam ejector	(f)	Boiler	

1,2, ..,15 represent the points 1-15 on Fig. 2

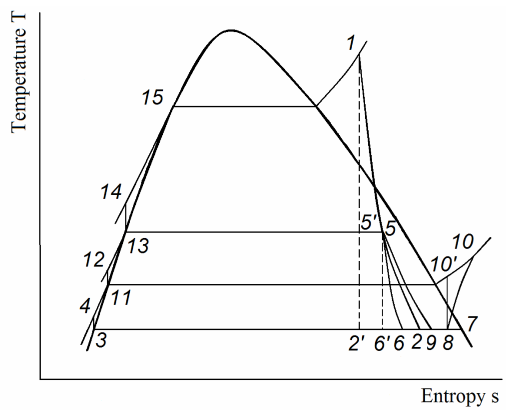

Fig. 2 Temperature - entropy diagram of the proposed modified steam cycle

The steam ejector (e), primary ejector, serves besides its main function to provide the feed water heater with heating steam, to keep the vacuum pressure in the condenser un-raised. When using open feed water heater, a smaller steam ejector (secondary ejector) is needed to remove dissolved gases in the mixture of water coming from the condenser and heating steam outlet from the primary ejector. In order to make the secondary ejector much smaller, a closed type feed water heater can be used. In this case the entrained steam from the condenser contains most of dissolved gases. As the outlet steam from the primary ejector is passed through the closed feed water heater, it condenses and leaves the dissolved gases that can be drawn out by the secondary ejector. In general, for either type of feed water heaters a special design is needed which enables removing most of the dissolved gases with possibly minimum amount of steam.

3. Thermodynamic Analyses of the Proposed Cycle

In this section, the processes of the proposed steam cycle are thermodynamically analyzed. These analyses enable the prediction of the cycle performance so that its goodness can be judged. It is assumed in these analyses that all processes occur at steady state and steady flow condition and the work of pumps are negligible in comparison to the turbine work. The last assumption implies that: $h_4=h_3$, $h_{12}=h_{11}$ and $h_{14}=h_{13}$, where h is the specific enthalpy and the numerical subscripts *1,2,3,…* and *15* stand for the points on the *T-s* diagram (see Figs. 1 and 2). Also, the feed water heaters (d1) and (d2) are assumed to be of the open type.

Referring to Figs. 1 and 2, heat rates Q_b and Q_c added in the boiler and rejected in the condenser, respectively are given by:

$$Q_b = m_b (h_1 - h_{13}) \tag{1}$$

and

$$Q_c = m_{cc} (h_2 - h_3) \tag{2}$$

Applying the continuity of mass equation to the steam turbine (a) and condenser (b), the mass rate m_{cc} may be given by:

$$m_{cc} = m_c - m_{ent} = m_b - m_{f,2} - m_{ms} - m_{ent} \tag{3}$$

Expansion in the steam turbines comprises two processes; i.e. expansion of steam rate m_b from state *1* to *5* and expansion of steam rate m_c between states *5* and *2*. Accordingly, the rate of work given to the turbine shaft is calculated by:

$$W_t = W_{1-5} + W_{5-2} = m_b (h_1 - h_5) + m_c (h_5 - h_2) \tag{4}$$

Assuming the turbine efficiency η_t is equal for both expansion processes, the irreversible adiabatic expansions *1-5* and *5-2* are related to the isentropic expansions *1-5′* and *5-2′* by:

$$\eta_t = \frac{h_1 - h_5}{h_1 - h_{5'}} = \frac{h_5 - h_2}{h_5 - h_{2'}} \tag{5}$$

In Combining Eqs. (3), (4) and (5), it follows that:

$$W_t = \eta_t \,[\, m_b (h_1 - h_{5'}) + (m_b - m_{f,2} - m_{ms})(h_5 - h_{2'})] \tag{6}$$

To be able to carry out the thermodynamic analyses of the feed water heaters (d1) and (d2), it is necessary to find out the values of the pressure inside them. Determination of bleed pressure p_{13} (equal to p_5 and p_{12}) is based on the assumption that bleed temperature t_{13} to obtain maximum efficiency for a regenerative cycle is approximately the arithmetic mean of the temperatures at states *15* and *3* [11]; i.e.

$$t_{13} = \frac{t_{15} + t_3}{2} \tag{7}$$

where t_3 and t_{15} are the water saturation temperatures at condenser pressure p_c and boiler pressure p_b respectively.

In the feed water heaters (d1) and (d2), water mass flow rates m_{cc} and $(m_b\text{-}m_{f,2})$ are mixed with steam rate $m_{f,1}$ $(m_{ms}+m_{ent})$ coming out of the ejector (e) and bleed steam rate $m_{f,2}$, respectively. It is assumed that the mixtures of water and steam get out of the feed water heaters (d1) and (d2) as saturated water at pressure p_{11} (p_{eo}) and p_{13} respectively. Hence, an energy balance for the feed water heater (d1) leads to:

$$(m_{ms} + m_{ent})(h_{10} - h_{11}) = m_{cc}(h_{11} - h_3) \tag{8}$$

Solving Eq. (8) in combination with Eq. (3) to find out m_{ent} results in:

$$m_{ent} = \frac{(m_b - m_{f,2})(h_{11} - h_3)}{\left(\dfrac{m_{ms}}{m_{ent}} + 1\right)(h_{10} - h_3)} \tag{9}$$

Similarly, an energy balance for the feed water heater (d2) leads to:

$$m_{f,2}(h_5 - h_{13}) = (m_b - m_{f,2})(h_{13} - h_{11}) \tag{10}$$

Rearranging the terms of Eq. (10) to find out $m_{f,2}$ results in:

$$m_{f,2} = \frac{m_b\,(h_{13} - h_{11})}{h_5 - h_{11}} \tag{11}$$

For completing the thermodynamic analyses of the proposed steam cycle, the steam ejector (e) is now considered. To better understand the following analyses, a section through a steam ejector is shown schematically in Figure 3. The pressure of steam flowing through the ejector is shown in Figure 4 corresponding to the stations marked in Fig. 3. The motive steam bled from the turbine at state *5* and with rate of m_{ms} is expanded in a convergent divergent nozzle and the exit condition is at state 6 where the steam pressure is equal to the pressure p_c of steam in the condenser and its velocity is supersonic. A steam mass flow rate of m_{ent} with pressure p_c is entrained from the condenser and mixed with m_{ms}. It is to be noticed in this figure that both the shock wave compression and diffuser compression is assumed to be replaced by simple compression and the efficiency of compression is modified to take into account this assumption. Thus, it follows for the nozzle efficiency η_n and compression efficiency η_c that:

$$\eta_n = \frac{h_5 - h_6}{h_5 - h_{6'}} \tag{12}$$

$$\eta_c = \frac{h_{10'} - h_8}{h_{10} - h_8} \tag{13}$$

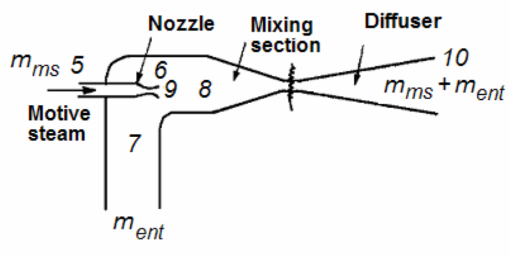

Fig. 3 Section through a steam ejector

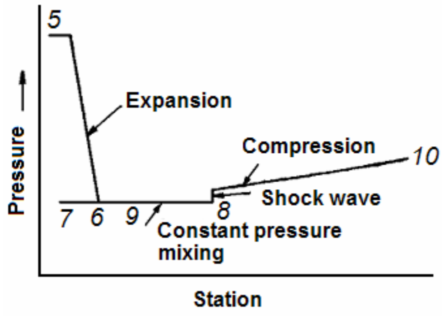

Fig. 4 Pressure at different stations of a steam ejector

The steam coming out of the nozzle is at high velocity while the steam coming out of the condenser is at low velocity. Giving the required momentum to steam exiting the condenser is called entrainment of steam. Thus the motive steam will lose some energy. This entrainment process is very inefficient. This is accounted for introducing a term called entrainment efficiency, which may be given by:

$$\eta_{ent} = \frac{h_5 - h_9}{h_5 - h_6} \tag{14}$$

where the subscript *9* stands for the state of steam just before mixing with the steam entrained from the condenser.

Applying the law of conservation of energy to the ejector results in:

$$m_{ms}\,(h_5 - h_9) = (m_{ms} + m_{ent})\,(h_{10} - h_8) \tag{15}$$

From Eqs. (12) through (15), the ratio of mass rates of motive steam to entrained steam can be given by:

$$\frac{m_{ms}}{m_{ent}} = \frac{(h_{10'} - h_8)}{\eta_n\ \eta_{ent}\ \eta_c\ (h_5 - h_{6'}) - (h_{10'} - h_8)} \tag{16}$$

For energy balance at entry to the mixing section of the ejector, it follows that:

$$m_{ms}\, h_9 + m_{ent}\, h_7 = (m_{ms} + m_{ent})\, h_8 \tag{17}$$

where the subscript *7* refers to the saturated steam at condenser pressure p_c.

Dividing both sides of Eq. (17) by m_{ent} and solving it to get h_8, it follows that:

$$h_8 = \frac{\frac{m_{ms}}{m_{ent}}\, h_9 + h_7}{\frac{m_{ms}}{m_{ent}} + 1} \tag{18}$$

Solution of Eqs. 1 through 18, which will be explained later in this section, enables the determination of the states of all points of the proposed regenerative steam cycle as well as the mass flow rates flowing through the different components of the cycle. Accordingly, the thermal efficiency of this cycle may be given by aid of Eqs. (1) and (6) by:

$$\eta_{th} = \frac{W_t}{Q_b} = \frac{\eta_t\,[\,m_b\,(\,h_1 - h_{5'}\,) + (\,m_b - m_{f,2} - m_{ms}\,)(\,h_5 - h_{2'}\,)\,]}{m_b\,(\,h_1 - h_{13}\,)} \tag{19}$$

For the sake of comparison with the normal regenerative cycle (without any waste heat recovery of the condenser), the thermal efficiency $\eta_{th'}$ of this cycle is to be determined. This can be found in any thermodynamic text book, e.g. [11]. Hence $\eta_{th'}$ is given by:

$$\eta_{th'} = \frac{\eta_t\,[\,m_b\,(\,h_1 - h_{5'}\,) + (\,m_b - m_{f'}\,)(\,h_5 - h_{2'}\,)\,]}{m_b\,(\,h_1 - h_{13}\,)} \tag{20}$$

Where $m_{f'}$ is the bleed steam rate required for heating process in the feed water heater of the normal regenerative cycle; it corresponds to feed water heater (d2) of the modified steam cycle. $m_{f'}$ is given by [11]

$$m_{f'} = \frac{m_b\,(h_{13} - h_3\,)}{h_5 - h_3} \tag{21}$$

Prediction of the states of the different points of the modified steam cycle can be performed by applying the methods explained in any thermodynamic text book, e.g. [11], except for the points *8*, *10′* and *10*. Finding out the states of these points necessitates knowing the values of the mass flow rates m_{ms} and m_{ent}. Calculation of these mass flow rates, and in turn, determination of the states of points *8*, *10′* and *10* is carried out through an iteration method developed in the current work and explained in the flow chart shown in Figure 5.

4. Results and Discussion

In order to test the advantages that the proposed regenerative steam cycle may have over the normal regenerative steam cycle, the foregoing analyses were applied to predict the thermodynamic characteristic performance of the proposed cycle. The results obtained hereafter are based on the basic design parameters of the cycle given in table 1. The pressures p_b and p_c of the boiler and condenser are set to 150 and 0.09 bar, respectively. The degree of superheat Δt_s amounts to 100 ^{o}C. The efficiencies η_t, η_n, η_{ent}, and η_c of the turbine, ejector nozzle, entrainment and diffuser compression have the values of 0.85, 0.85, 0.7 and 0.8 respectively. The thermal properties of water and steam flowing through the cycle were determined as a function of pressure and temperature by aid of the soft ware associated with the text book [12]. This soft ware gives water and steam thermal properties in exact agreement with the values existing in steam tables [13]. In the calculations carried out in this study, only the parameter/s, whose effect was to be tested, was/were changed while the other parameters were kept constant at their values given in Table 1.

Figure 6 shows the relation between the mass flow rate ratios m_{ms}/m_b, $m_{f,2}/m_b$ and $(m_{ms}+m_{f,2})/m_b$ and the steam pressure p_{eo} at the exit of the steam ejector for condenser pressure p_c of 0.03, 0.09 and 0.15 bar. For the sake of comparison with the normal regenerative steam cycle, the ratio $m_{f'}/m_b$ is plotted on the diagrams of Fig. 6. The ratio $m_{f'}/m_b$ is independent of p_{eo}, and in turn, it is shown in these diagrams as straight lines parallel to the p_{eo}- axis.

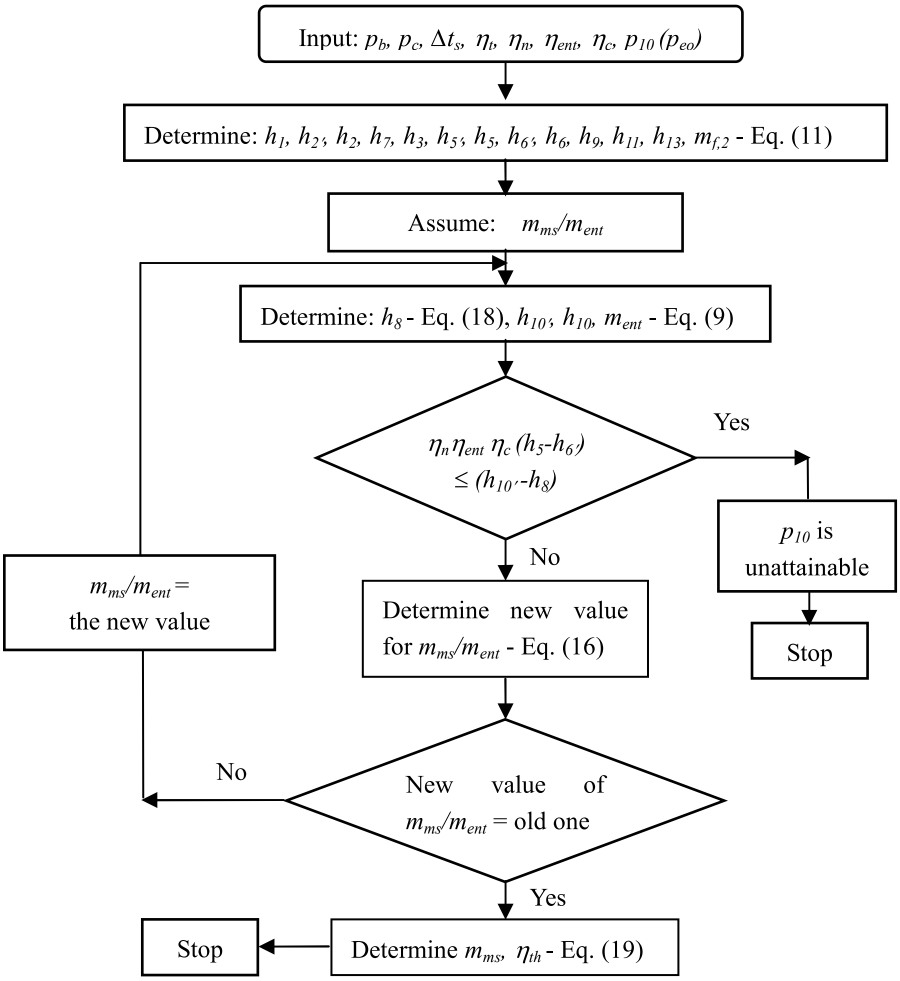

Figure 5. Flow chart of the iteration method

Table 1: Basic design data of the proposed modified steam cycle

Boiler pressure	p_b	bar	150
Condenser pressure	p_c	bar	0.09
Degree of superheat	Δt_s	°C	100
Turbine efficiency	η_t	-	0.85
Steam ejector nozzle efficiency	η_n	-	0.85
Steam ejector entrainment efficiency	η_{ent}	-	0.7
Steam ejector compression efficiency	η_c	-	0.8

It is clear from Fig. 6 that the steam ejector can only function in a limited range of p_{eo}. This range depends in essential on the pressure p_c of the condenser. As it can be seen from Fig. 6, this range is given by: 0.033 – 0.485 bar, 0.1 – 0.98 bar and 0.163 – 1.31 bar for p_c of 0.03, 0.09 and 0.15 bar respectively. Also, it is obvious from Fig. 6 that m_{ms} is increased linearly as p_{eo} is elevated. This is mainly ascribed to the increased energy needed to raise the pressure of steam entering the ejector diffuser from p_c to p_{eo}. Contrarily, $m_{f,2}$ is reduced with increasing p_{eo} due to the greater amount of heating steam entering the feed water heater (d1), see Fig. 1. The sum of m_{ms} and $m_{f,2}$ is always less than $m_{f''}$ along the whole range of the ejector working pressure p_{eo}. The minimum value of $m_{ms}+m_{f,2}$ occurs at a certain pressure p_{eo} whose value is dependent on the value of the condenser pressure p_c. This pressure amounts to 0.13, 0.31and 0.47 bar for p_c of 0.03, 0.09 and 0.15 bar respectively. The amount of steam mass rate representing the difference between $m_{f'}$ and $m_{ms}+m_{f,2}$ serves to increase the turbine power over that of the normal regenerative steam cycle, which results in higher power output of the proposed cycle, while the rate of heat added to the working fluid (water) in the boiler is kept the same for both cycles. Hence, the thermal efficiency of the proposed cycle is expected to be higher as it will be seen later in this section.

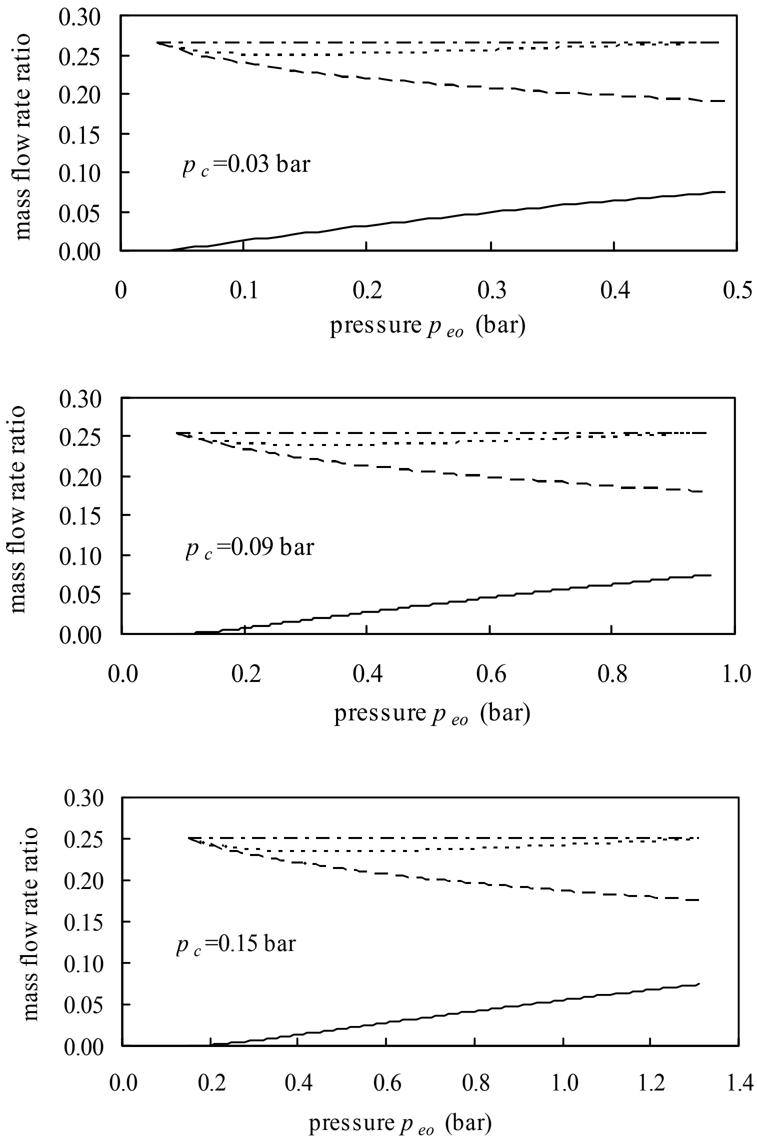

Fig. 6 Effect of steam pressure at outlet of the ejector on steam mass flow rates extracted from the turbine

————	m_{ms}/m_b	— — —	$m_{f,2}/m_b$
…………	$(m_{ms}+m_{f,2})/m_b$	—.—.—	$m_{f'}/m_b$

In Figure 7 the mass flow rate ratio m_{ent}/m_b is plotted versus the pressure p_{eo} for a condenser pressure p_c of 0.03, 0.09 and 0.15 bar. This figure reveals obviously that m_{ent}

grows up steeply with increasing p_{eo} till it reaches a maximum at a certain pressure whose value depends on the value of the condenser pressure p_c. Comparing the diagrams of Fig. 6 with Fig. 7, it can be inferred that the values of the pressure p_{eo} at which maximal m_{ent} occurs, are coincident with those at which minimum sum of m_{ms} and $m_{f,2}$ comes about.

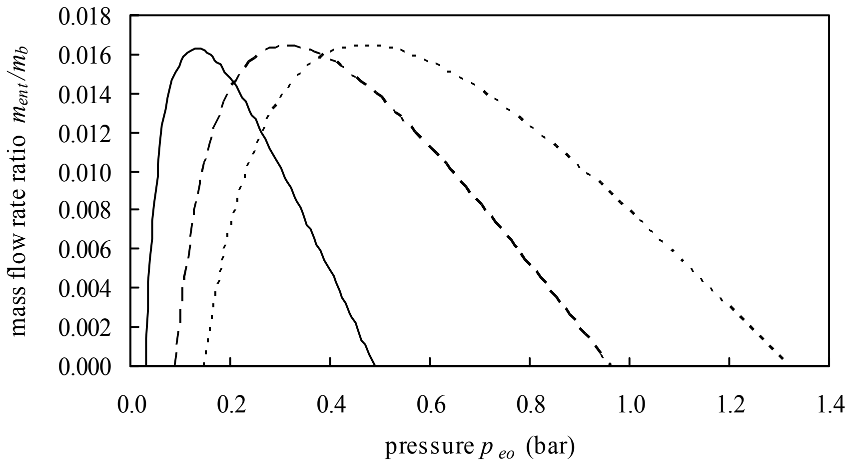

Fig. 7 Effect of steam pressure at outlet of the ejector on the entrained steam flow rate

——— p_c=0.03 bar　　— — — p_c=0.09 bar　　……… p_c=0.15 bar

The effect of the pressure p_{eo} on the thermal efficiency of the new cycle η_{th} is exhibited in Figure 8, where it is plotted against p_{eo} for condenser pressure p_c of 0.03, 0.09 and 0.15 bar. For the purpose of comparison with the normal regenerative cycle, $\eta_{th'}$ is shown in Fig. 8 too. As $\eta_{th'}$ is independent of p_{eo}, it is represented by lines parallel to the p_{eo}-axis. It is to be seen from this figure that η_{th} is always higher than $\eta_{th'}$ along the whole range of p_{eo} of the proposed cycle. This can be interpreted as explained above by the less amount of steam $m_{ms}+m_{f,2}$ used for heating purpose in the feed water heaters (d1) and (d2) of the proposed cycle than $m_{f'}$ used for the same purpose in the sole feed water heater of the normal regenerative cycle. Thus, more steam mass flow rate exerts work in the low pressure turbine in the case of the proposed cycle than in the case of the normal regenerative cycle. This results in greater power of the proposed cycle than that of the normal regenerative cycle while the heat rate added in the boiler of each cycle is the same.

Fig. 8 shows clearly that the values of the pressure p_{eo} at which maximal η_{th} occurs are in coincidence with those at which maximal m_{ent} and minimum sum of m_{ms} and $m_{f,2}$ takes place. Also, it is evident from Fig. 8 that the increase in the maximal efficiency of the proposed cycle over that of the normal regenerative cycle is independent of the condenser pressure and it amounts in average to 1.1%.

The mass flow rate m_{cc} of steam being condensed inside the condenser of the proposed cycle can be determined by aid of Eq. (3). The corresponding steam mass flow rate $m_{cc'}$ being condensed in the condenser of the normal regenerative steam cycle can be given similarly as in the case of Eq. (3) by:

$$m_{cc'} = m_b - m_{f'} \tag{22}$$

m_{cc} and $m_{cc'}$ were calculated by aid of Eqs. (3) and (22), respectively and they are plotted in Figure 9 versus the steam pressure p_{eo} at ejector outlet. As a matter of course $m_{cc'}$ is independent of p_{eo} and appears as straight lines parallel to p_{eo}-axis. It is clear from Fig. 9 that m_{cc} is always less than $m_{cc'}$. The minimum value of m_{cc} for a given value of the condenser pressure p_c occurs at the same pressure p_{eo} at which the maximum cycle thermal efficiency takes place. The minimum value of m_{cc} is less by 1. 0%, 1.03% and 1.05% than the value of $m_{cc'}$ for condenser pressure p_c of 0.03, 0.09 and 0.15 bar respectively. This leads to infer that the cooling load of the condenser would be a bit lower for the proposed cycle than that of the normal regenerative cycle.

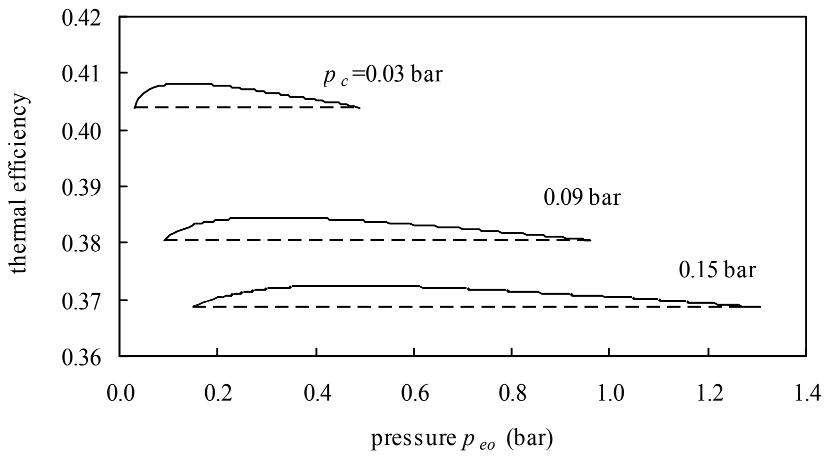

Fig. 8 Effect of steam pressure at ejector outlet on cycle thermal efficiency

——— η_{th} - - - - - - $\eta_{th'}$

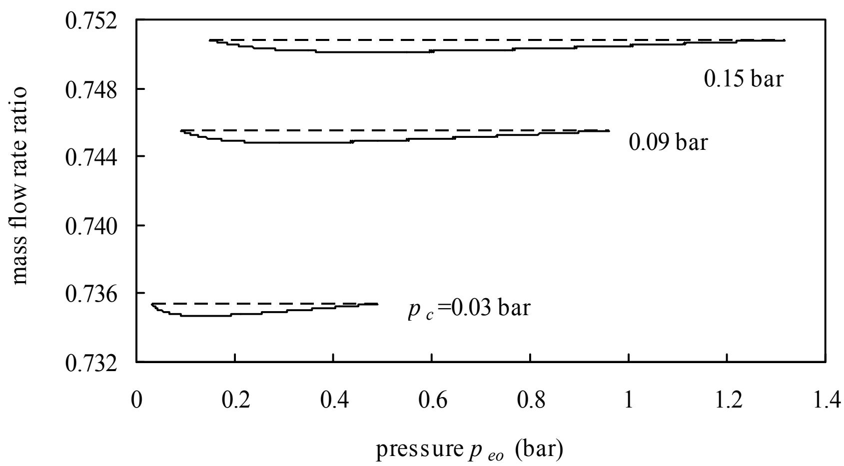

Fig. 9 Dependence of the condensate flow rate out of the condenser on the steam pressure at ejector outlet

——— m_{cc}/m_b - - - - - - - $m_{cc'}/m_b$

5. Conclusions

Thermodynamic analyses are presented for a proposed new regenerative steam cycle. In this cycle, a steam ejector and an extra feed water heater are inserted into the normal regenerative steam cycle with one feed water heater. The ejector is used to entrain some of the saturated steam entering the condenser to be used along with the ejector motive steam to heat the condensate outlet from the condenser. In this way a portion of the heat to be rejected in the condenser can be recovered and utilized in the cycle. The results of this study lead to draw the following conclusions:

1. The steam ejector has a working range of outlet steam pressure within which it can be put in operation. Outside this range the steam ejector does not work and in turn it has no effect on the cycle.
2. Thermal efficiency of the proposed cycle is always higher than that of the corresponding normal regenerative steam cycle throughout the working range of the steam ejector.
3. There is a maximal thermal efficiency for the proposed cycle which occurs at a certain value of the steam pressure at the outlet of the steam ejector.
4. The steam pressure at the ejector outlet for which the maximum efficiency occurs is dependent on the condenser pressure, where the former increases as the latter is raised.
5. The increase in the maximal efficiency of the proposed cycle over that of the normal regenerative cycle is independent of the condenser pressure and it amounts in average to 1.1%.

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References

(1) W. Kearton, Steam Turbine Theory & Practice,7th edition, Sir Isaac Pithman&sons, (1961).
(2) P. Hill, Power Generation: Resources, Hazards, Technology, and Costs, MIT Press, (1997).
(3) Y. Cengel, M. Boles, Thermodynamics: An Engineering Approach, McGraw-Hill, (1998).
(4) H. Termuehlen, 100 Years of Power Plant Development: Focus on Steam and Gas Turbines as Prime Movers, ASME Press, (2001).
(5) K.H. Mayer, W. Bendick, R.U Husemann, T. Kern and R.B. Scarlin, New Materials for Improving the Efficiency of Fossil-Fired Thermal Power Stations, VGB PowerTech No.1 pp22-27 (1998).
(6) R. Viswanathan and W. Bakker, Materials for Ultrasupercritical Coal Power Plants, Journal of Materials Engineering and Performance, 10 (1) (2001).
(7) A. Fleming, R.V. Maskell, L.W. Buchanan and T. Wilson, Material Developments for Supercritical Boilers and Pipework in Materials for High Temperature Power Generation and Process Plant Applications, Eds. A. Strang, Institute of Materials Book No. 728. pp32-77 (2000).
(8) R. Vanstone, Advanced 700°C Pulverised Fuel Power Plant, Proc. 5th Int. Charles Parsons Conf: Parsons 2000 Advanced Materials for 21st Century Turbine and Power Plants. IOM Communications, London, Book 736 (2000).
(9) K.H. Mayer, Advanced heat resisting steels, from Advanced Energy Conversion Materials, published by Nova Science Publisher Inc, USA, (2001).
(10) R. Viswanathan and W.T. Bakker, Materials for Ultra Supercritical Coal Power Plants Part 2: Turbine Materials, from Power Plant Chemistry, 3(8), pp446-451 (2001).
(11) T.D. Eastop and A. McConkey, Applied Thermodynamics For Engineering Technologists, 5th edition, Longman (1993).
(12) Sonntag, R.E., Borgnakke C., and VanWylen G.J., "Introduction to Classical Thermodynamics", 4th Edition, John Wiley & Sons Inc., (1994).
(13) VDI-Wasserdampftafeln, VDI Verlag, Duesseldorf, 6 Auflage, (1963).