Determination of optimum reheat pressures for single and double reheat irreversible Rankine cycle

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Determination of optimum reheat pressures for single and double reheat irreversible Rankine cycle

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This paper reports thermodynamic optimisations based on the maximum net cycle work and the maximum thermal efficiency criteria for an irreversible single and double reheat Rankine cycle, which includes the internal irreversibilities resulting from the adiabatic processes. The study encompasses the effects of boiler pressure and temperature on the general and optimal performances of the plant in terms of net work output and thermal efficiency. Therefore, the net work output and thermal efficiency of the plant are obtained by introducing the mechanical, boiler and isentropic efficiencies. The effects of boiler pressure and temperature on the optimal reheat pressures of the single and double reheat Rankine cycles are also investigated. It is shown that for the theoretical Rankine cycle, the optimal reheat pressures of single and double reheat Rankine cycle are both fairly affected by the boiler pressure and temperature.

Keywords: Rankine cycle, Performance analysis, Optimum reheat pressure, Thermal efficiency, Net work output

List of symbols

h	specific enthalpy, kJ kg^{-1}
P	pressure, bar
q	heat transfer per unit mass, kJ kg^{-1}
s	specific entropy, kJ kg^{-1} K^{-1}
T	temperature, °C
v	specific volume, m^3 kg^{-1}
w	work output per unit mass, kJ kg^{-1}
x	quality
δ_1	thermal efficiency increase rate for the single reheated plant
δ_2	thermal efficiency increase rate for the double reheated plant
η	efficiency
η_B	boiler efficiency
η_i	isentropic efficiency
η_m	mechanical efficiency

Subscripts

B	boiler
C	condenser
f	function
i	isentropic
in	input
m	mechanical
net	net
out	output
p	pump
th	thermal
X	first stage reheat pressure
Y	second stage reheat pressure

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Introduction

Basic steam power plants operate based on Rankine cycle and can be employed where both electricity and heat are required. Many techniques are being used to increase the efficiency of the steam cycle. Most outstanding of these techniques are reheating and/or reducing the irreversibilities. Reheating increases the efficiency by raising the mean temperature of the heat addition process, by increasing the steam temperature at the turbine inlet and also by increasing the efficiency of the expansion process in the steam turbine.[1] The irreversibility of the steam generator can be decreased by raising the steam temperature at the turbine inlet. Another technique to improve the efficiency is to enhance the quality of steam at the condenser inlet.[2]

Acar[3] conducted a research about reheat regenerative Rankine cycle, where he analysed a steam power cycle based on the second law by its implementation to the reheat regenerative Rankine cycle. Retzlaff and Ruegger[4] explained the efforts of advancement in the state-of-the-art in steam turbine technology, which is related to the improvement in the thermodynamic efficiency by increasing the temperature and pressure at which heat is added to the power cycle. They investigated steam turbines for ultrasupercritical power plants and reported heat rate improvement of single and double reheat steam cycles with ultrasupercritical steam conditions on their work. Habib *et al.*[5] studied the first and second law procedures for the optimisation of the reheat level in reheat regeneration thermal power plants.

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The procedure was used for a thermal power plant having two reheat pressure stages and open feed water heaters. He calculated and optimised the second law efficiency of the steam generator, turbine cycle and plant, taking into account the irreversibilities and some constraints, such as the steam qualities.

Dincer and Al-Muslim[6] performed a thermodynamic analysis of a single reheat Rankine cycle for the steam power plants based on the first and second laws of thermodynamic. He studied the energy and exergy efficiencies for different system parameters, such as boiler temperature, boiler pressure, mass fraction ratio and work output. In his study, the results of the efficiencies were compared with the real data and literature and found good agreement.

Bassily[7] designed and optimised the dual and triple pressure non-reheat combined cycles. In his model, the triple pressure cycle was essentially chosen to decrease the NO_x emission using steam injection. Another research carried out by Bassily,[8,9] which includes modelling and optimisation of a dual and triple pressure reheat combined cycle for many cases, is about the determination of constraints based on the minimum temperature difference for pinch points, the temperature difference for superheat approach, steam turbine inlet temperature and pressure, stack temperature and dryness fraction at the steam turbine outlet.

Although so many applied studies have been performed so far, there is a lack in theoretical studies about Rankine cycle in the literature. Accordingly, this paper presents the detailed and comprehensive analysis for the thermodynamic evaluation of the single and double reheated Rankine cycle.

Thermodynamic analysis of reheated irreversible Rankine cycles

Single reheat Rankine cycle

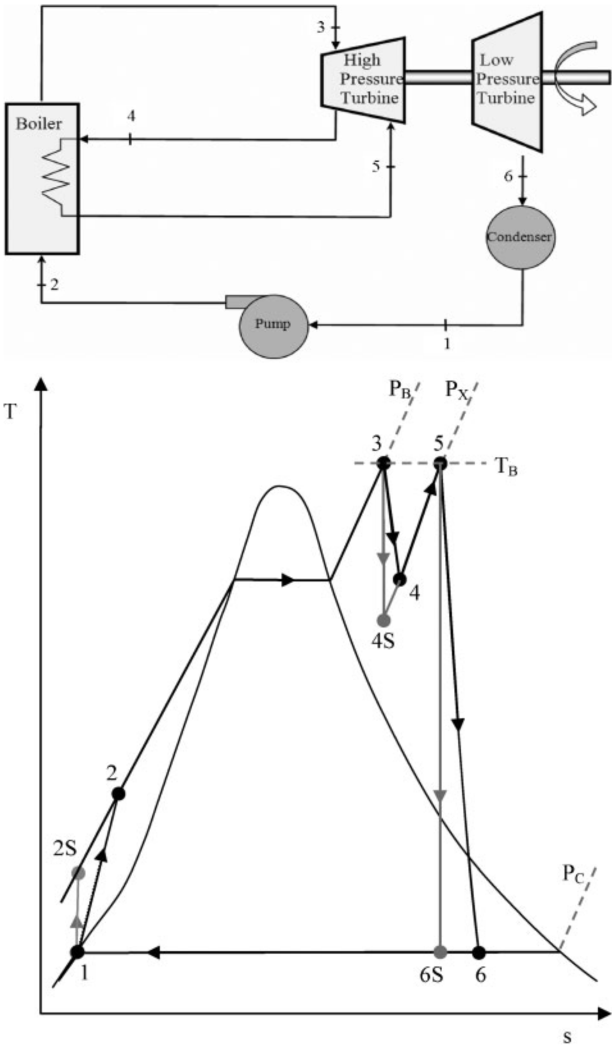
The flow[10] and temperature–specific entropy *T–s* diagrams for the thermodynamic processes of a standard irreversible single reheat Rankine cycle operating with water are shown in Fig. 1. In the diagram, process 1–2S is an isentropic (reversible adiabatic) work input of the pump, while process 1–2 is an irreversible adiabatic process that takes into account the internal irreversibilities in the real cycle process. The heat addition occurs in two steps. Processes 2–3 and 4–5 are heat additions in the boiler by heating and reheating respectively. Processes 3–4S and 5–6S are isentropic work outputs of high pressure turbine (HPT) and low pressure turbine (LPT), while processes 3–4 and 5–6 take into account internal irreversibilities in the real Rankine cycle. A constant temperature heat rejection process by the condenser 6–1 completes the cycle. The heat added by heating and reheating in the boiler during the constant pressure processes 2–3 and 4–5 is

$$q_{\rm in}=\frac{(h_3-h_2)+(h_5-h_4)}{\eta_{\rm B}} \tag{1}$$

and the heat rejection process in the condenser during the constant process 6–1 is

$$q_{\rm out}=h_6-h_1 \tag{2}$$

The net work output and thermal efficiency of the single reheated Rankine cycle can be written respectively as



1 Flow and *T–s* diagrams for single reheat irreversible Rankine cycle

$$w_{\rm net}=w_{\rm in}-w_{\rm out}=[(h_3-h_{4\rm S})+(h_5-h_{6\rm S})-w_{\rm pump}]\eta_{\rm i}\eta_{\rm m} \tag{3}$$

and

$$\eta_{\rm th}=\frac{w_{\rm net}}{q_{\rm in}}=1-\frac{q_{\rm out}}{q_{\rm in}}$$
$$=\left[\frac{(h_3-h_{4\rm S})+(h_5-h_{6\rm S})-w_{\rm pump}}{(h_3-h_2)+(h_5-h_4)}\right]\eta_{\rm i}\eta_{\rm m}\eta_{\rm B} \tag{4}$$

where the work input to the pump is given as

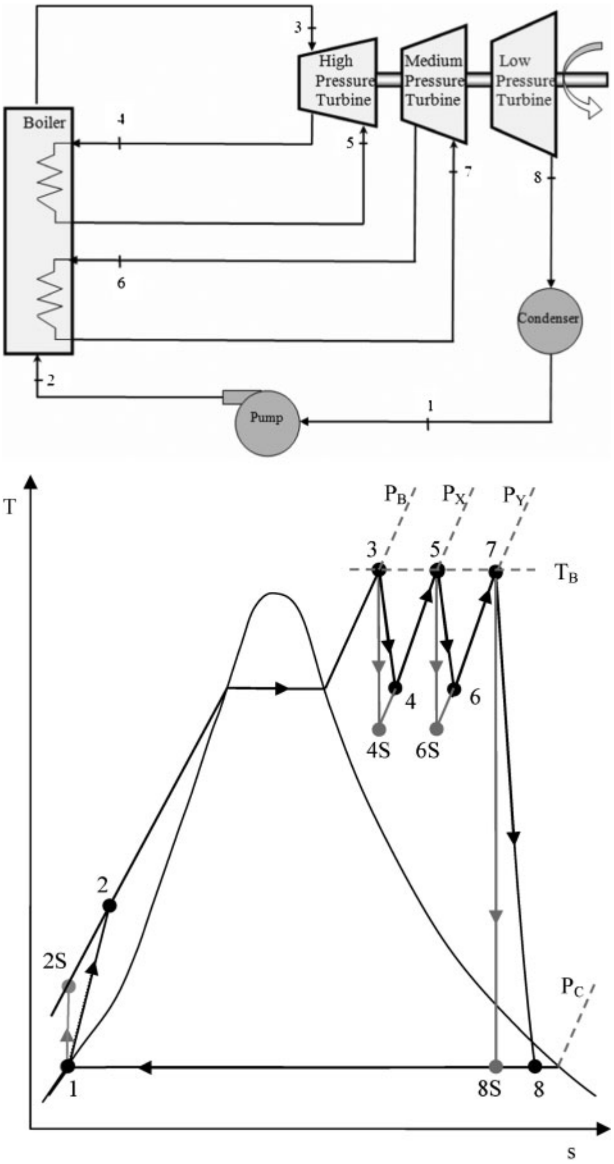
$$w_{\rm pump}=v_1(P_2-P_1) \tag{5}$$

The thermal efficiency increase rate for the single reheated plant is

$$\delta_1=\frac{\eta_{\rm th}-\eta_{\rm th,non\text{-}reheat}}{\eta_{\rm th,non\text{-}reheat}} \tag{6}$$

Double reheat Rankine cycle

The flow[10] and *T–s* diagrams for the thermodynamic processes of a standard irreversible double reheat Rankine cycle operating with water is shown in Fig. 2. In the diagram, the process 1–2S is an isentropic work input of the pump, while process 1–2 is an irreversible adiabatic process that takes into account the internal



2 **Flow and *T–s* diagrams for double reheat irreversible Rankine cycle**

irreversibilities in the real cycle process. The heat addition occurs in process 2–3 by heating and processes 4–5 and 6–7 by reheating in the boiler respectively. Processes 3–4S, 5–6S and 7–8S are isentropic work outputs of the HPT, middle pressure turbine (MPT) and LPT, while processes 3–4, 5–6 and 7–8 take into account internal irreversibilities in the real Rankine cycle. A constant temperature heat rejection process by the condenser 8–1 completes the cycle.

The heat added by heating and reheating in the boiler during the constant pressure processes 2–3, 4–5 and 6–7 is

$$q_{\mathrm{in},2}=\frac{(h_3-h_2)+(h_5-h_4)+(h_7-h_6)}{\eta_\mathrm{B}} \tag{7}$$

and the heat rejection process in the condenser during the constant process 8–1 is

$$q_{\mathrm{out},2}=h_8-h_1 \tag{8}$$

The net work output and thermal efficiency of the double reheated Rankine cycle can be written respectively as

$$\begin{aligned}w_{net,2}&=w_{in}-w_{out}\\&=\left[(h_3-h_{4s})+(h_5-h_{6s})=(h_7-h_{8s})-w_{pump}\right]\eta_i\eta_m\end{aligned} \tag{9}$$

and

$$\begin{aligned}\eta_{\mathrm{th},2}&=\frac{w_{\mathrm{net},2}}{q_{\mathrm{in},2}}=1-\frac{q_\mathrm{out}}{q_\mathrm{in}}\\&=\left[\frac{(h_3-h_{4\mathrm{s}})+(h_5-h_{6\mathrm{S}})+(h_7-h_{8\mathrm{S}})-w_\mathrm{pump}}{(h_3-h_2)+(h_5-h_4)+(h_7-h_6)}\right]\eta_\mathrm{i}\eta_\mathrm{m}\eta_\mathrm{B}\end{aligned} \tag{10}$$

The efficiency increase rate for the double reheated plant is

$$\delta_2=\frac{\eta_{\mathrm{th},2}-\eta_{\mathrm{th,non\text{-}reheat}}}{\eta_{\mathrm{th,non\text{-}reheat}}} \tag{11}$$

The isentropic efficiencies for the pump and turbines are

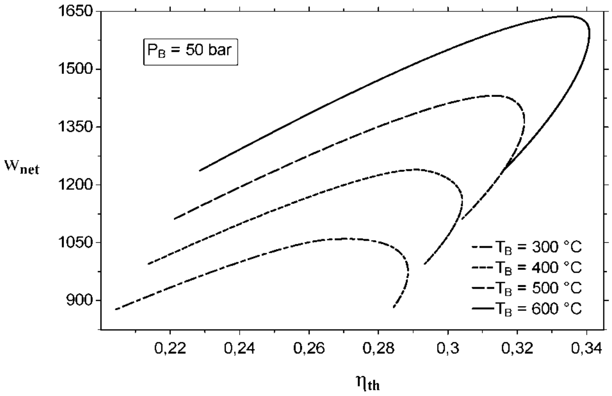
$$\eta_{\mathrm{i,P}}=\frac{h_{2\mathrm{S}}-h_1}{h_2-h_1} \tag{12}$$

and

$$\eta_{\mathrm{i,T_1}}=\frac{h_3-h_4}{h_3-h_{4\mathrm{S}}} \tag{13}$$

Table 1 Calculation procedures using engineering equation solver for double reheat Rankine cycle

State no.	Fluid	Phase	Quality	Temperature T/°C	Pressure P/kPa	Specific volume/ m^3 kg^{-1}	Specific enthalpy h/kJ kg^{-1}	Specific entropy s/kJ kg^{-1} K^{-1}
1	Water	Liquid	0	f(P_C, x_1)	P_C	f(P_C, x_1)	f(P_C, x_1)	f(P_C, x_1)
2S	Water	Comp. liquid	…	f($h_{2\mathrm{S}}$, P_B)	P_B	f($h_{2\mathrm{S}}$, P_B)	f(h_1, w_p)	f($h_{2\mathrm{S}}$, P_B)
2	Water	Comp. liquid	…	f(h_2, P_B)	P_B	f(h_2, P_B)	f(h_1, $h_{2\mathrm{S}}$, η_i)	f(h_2, P_B)
3	Steam	Superheated vapour	…	T_B	P_B	f(T_B, P_B)	f(T_B, P_B)	f(T_B, P_B)
4S	Steam	Superheated vapour	…	f(P_4, $s_{4\mathrm{S}}{=}s_3$)	P_X	f(P_X, $s_{4\mathrm{S}}{=}s_3$)	f(P_X, $s_{4\mathrm{S}}{=}s_3$)	f(T_B, P_B)
4	Steam	Superheated vapour	…	f(h_4, P_4)	P_X	f(h_4, P_X)	f(h_3, $h_{4\mathrm{S}}$, η_i)	f(h_4, P_X)
5	Steam	Superheated vapour	…	T_B	P_X	f(T_B, P_X)	f(T_B, s_5)	f(T_B, P_X)
6S	Steam	Superheated vapour	…	f(P_6, $s_{6\mathrm{S}}{=}s_5$)	P_Y	f(P_Y, $s_{6\mathrm{S}}{=}s_5$)	f(P_Y, $s_{6\mathrm{S}}{=}s_5$)	f(T_B, P_Y)
6	Steam	Superheated vapour	…	f(h_6, P_6)	P_Y	f(h_6, P_Y)	f(h_5, $h_{6\mathrm{S}}$, η_i)	f(h_6, P_Y)
7	Steam	Superheated vapour	…	T_B	P_Y	f(T_B, P_Y)	f(T_B, s_7)	f(T_B, P_Y)
8S	Steam	Saturated water vapour mixture	f(P_C,$s_{8\mathrm{S}}{=}s_7$)	f(P_C, $s_{8\mathrm{S}}{=}s_7$)	P_C	f(P_C, $s_{8\mathrm{S}}{=}s_7$)	f(P_C, $s_{8\mathrm{S}}{=}s_7$)	f(T_B, P_Y)
8	Steam	Saturated water vapour mixture	f(P_C, s_8)	f(h_8, P_8)	P_C	f(P_C, s_8)	f(h_7, $h_{8\mathrm{S}}$, η_i)	f(h_8, P_8)



3 Effect of T_B on $w_{net}-\eta_{th}$ characteristics

$$\eta_{i,T_2}=\frac{h_5-h_6}{h_5-h_{6S}} \tag{14}$$

$$\eta_{i,T_3}=\frac{h_7-h_8}{h_7-h_{8S}} \tag{15}$$

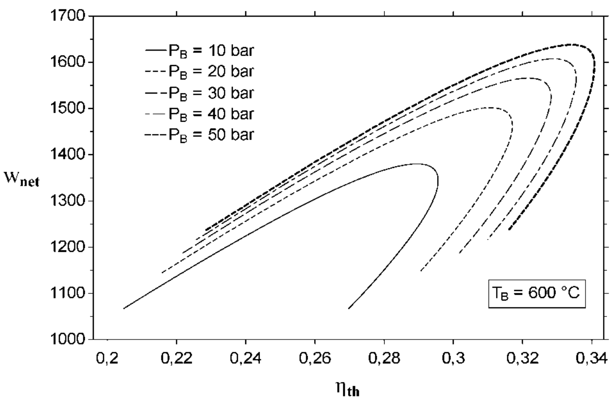
The enthalpy values of the state points can be found depending on their own T, P, x and s values using thermodynamic tables or diagrams. In order to find the thermodynamic properties of the state points, the engineering equation solver[11] software has been used, and the calculation procedure is also given in Table 1.

Results and discussion

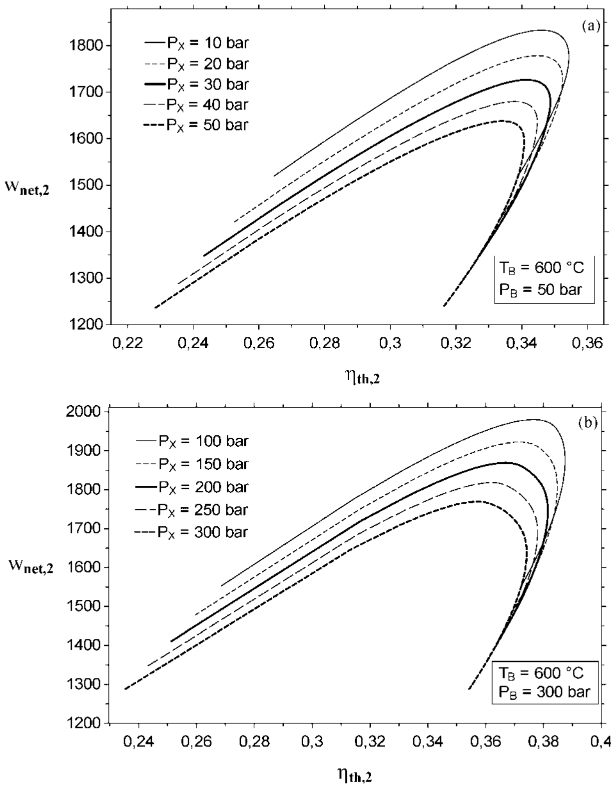
To investigate the single and double reheat effects on the performance of an irreversible Rankine cycle, a performance analysis has been carried out, and detailed numerical examples are provided. In this numerical analysis, the ranges for T_B and P_B are 300–600°C and 10–50 bar respectively. Additionally, the parameters selected are η_i=0·9, η_m=0·95 and η_B=0·9.

The curves in Figs. 3–5 are all in loop form, and thus, they have two different maxima for both axes, which are net work output and thermal efficiency respectively. The effects of T_B and P_B on the W_{net}–η_{th} characteristic curves for P_B =50 bar and T_B =600°C are indicated in Figs. 3 and 4 respectively. These figures show that the general and optimal performances in terms of net work output and thermal efficiency increase with increasing T_B and P_B.

In Figs. 6 and 7, the effects of T_B and P_B on the efficiency increase rate δ_1 are shown with respect to P_X.



4 Effect of P_B on $w_{net}-\eta_{th}$ characteristics

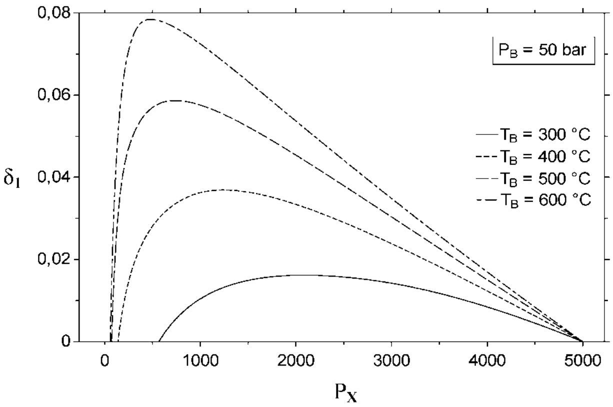


5 Variation of net work output with respect to thermal efficiency for double reheat irreversible Rankine cycle for *a* P_B=50 bar and *b* P_B=300 bar

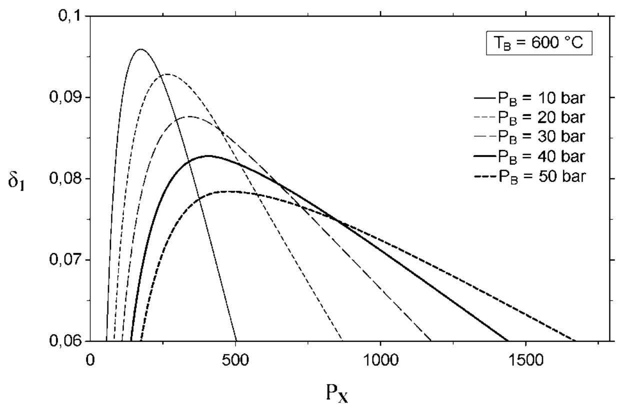
It is seen from the figures that the efficiency increase rate rises for increasing T_B and decreasing P_B, and also there is an optimum value for P_X that maximises the efficiency increase rate. This optimum value increases for decreasing T_B and increasing P_B.

The curves in Figs. 5 and 8 are derived from the analyses of the double reheated Rankine cycle. The variations of net work output $W_{net,2}$ with respect to the thermal efficiency $\eta_{th,2}$ for different first stage reheat pressures P_X with T_B=600°C are shown in Fig. 5*a* and *b* for P_B=50 bar and P_B=300 bar respectively. It is found that the general and optimal performances in terms of net work output and thermal efficiency decrease as the first stage reheat pressure increases.

The effects of first and second stage reheat pressures P_X and P_Y on the efficiency increase rate δ_2 for the double reheated Rankine cycle with T_B=600°C are



6 Effect of T_B on δ_1-P_x characteristics

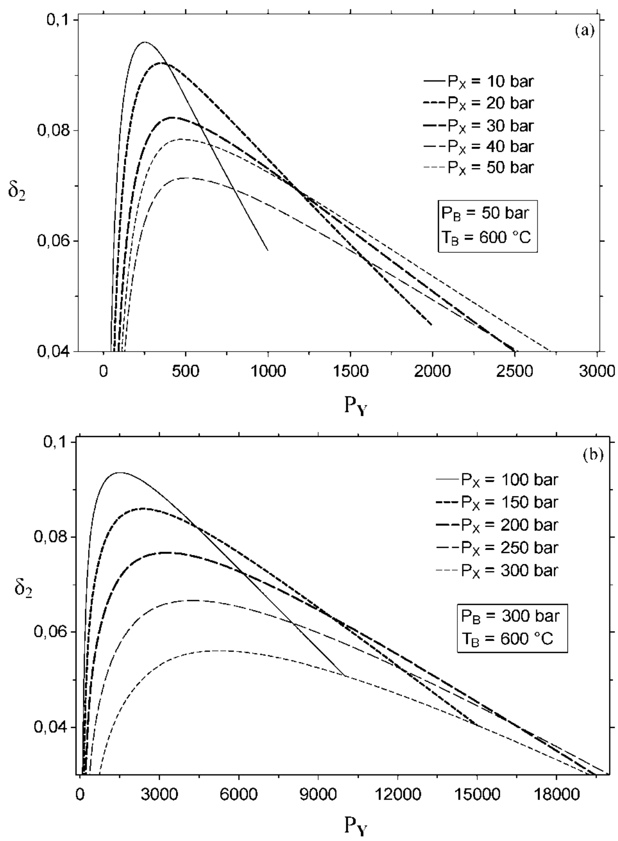


7 Effect of P_B on $\delta_1 - P_x$ characteristics

shown in the Fig. 8*a* and *b* for P_B=50 bar and P_B=300 bar respectively. It can be seen from these figures that there is an optimum value for P_Y that maximises the efficiency increase rate, and this optimum value increases for increasing P_X.

Conclusions

In this paper, the effects of boiler temperature and pressure and the irreversibilities on the performance of single and double reheat irreversible Rankine cycles are investigated. The relations between net work output, thermal efficiency and efficiency increase rate with reheat pressures are derived by taking irreversibilities into account, which are isentropic, mechanical and boiler thermal loses. The effects of first and second stage reheat pressures on the general performance and efficiency increase rates for the single and double reheated Rankine cycles are also investigated. It is obtained from these results that the general and optimal performances in terms of net work output and thermal efficiency decrease as the first stage reheat pressure increases, and also there is an optimum value for the second stage reheat pressure that increases with increasing first stage reheat pressure. The analysis helps us to understand the effects of boiler temperature and pressure on the performance of single and double reheat irreversible Rankine cycles. Furthermore, the results clearly depict how the optimal reheat pressure can be determined according to the thermal efficiency criterion. The results are of importance to provide guidance for the design of real steam power plants.



8 Variation of δ_2 with respect to P_Y for different P_X values for *a* P_B=50 bar and *b* P_B=300 bar

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