Modeling and Simulation of the Cogeneration Plant Equipped with Back-Pressure Turbine Operates at Various Control Programs of Exit Steam Temperature

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ABSTRACT

Cogeneration represents one of the main ways for increasing the efficiency of primary energy use. This paper deals with a theoretical analysis of cogeneration plants equipped with various back-pressure turbines type R. These plants are studied during their operation with conventional method (water injection) and the suggested method (sliding live steam temperature) to regulate steam temperature supplied to the industrial consumers. A computer program had been written to work under MathCad software to simulate cogeneration plant with each back-pressure turbine under design and off design conditions. The performance of the different schemes is analysed in view of the first and second laws. In this analysis entropy method (second law) in addition to more conventional energy analysis (first law), are employed to evaluate overall and component efficiencies, fuel consumption and to identify the thermodynamic losses. The results show that, using the suggested method leads to increase the overall efficiency of the cogeneration plant for all types of back-pressure turbine and can reduce the fuel consumption. Finally, the results show that increasing back-pressure leads to improve the performance of cogeneration plant regardless of the method used.

Keywords: Cogeneration; Combined heat and power plant; Back pressure turbine; Live steam temperature.

نمذجة ومحاكاة لمحطة مشتركة مزودة بتوربين ذو ضغط خلفي تعمل على برامج سيطرة مختلفة لدرجة حرارة البخار الخارج من التوربين

الخلاصة

تعتبر المحطات المشتركة واحدة من الطرق الأساسية لزيادة كفاءة إستخدام الطاقة. يتناول البحث دراسة نظرية لمحطات مشتركة مزوده بتوربينات ذات ضغط خلفي مختلفة من نوع R. تمت دراسة هذه المحطات من خلال تشغليها بالطريقة التقليدية (طريقة حقن الماء) والطريقة المقترحة (تغيير درجة حرارة البخار الداخل للتوربين) للسيطرة على درجة حرارة البخار المزود للمستهلك الصناعي. وتم وصف نموذج محاكاة لمحطات توليد مشتركة تعمل على ضوء مواصفات التوربينات

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التي تم اختيارها ودراسة ظروف عملها (التصميمية وغير التصميمية) من خلال الموديل الرياضي المكتوب ببرنامج يعمل على (MathCad software). تم تحليل المحطات المشتركة وفقاً للقانون الأول للثرموداينمك (Energy balance) وكذلك استناداً الى القانون الثاني (Entropy method) لمقارنة أداء كل محطة عندما تعمل بالطريقة التقليدية والطريقة المقترحة. وبينت النتائج بأن إستخدام الطريقة المقترحة يؤدي إلى زيادة الكفاءة الكلية للمحطة المشتركة ولجميع انواع التوربينات التي تم دراستها وكذلك يؤدي إلى تقليل معدل أستهلاك الوقود في المحطة. وأخيراً بينت النتائج بأن زيادة مقدار الضغط الخلفي للتوربين يسبب تحسين في أداء المحطة المشتركة بغض النظر عن الطريقة المستخدمة للسيطرة على درجة حرارة البخار المزود للمستهلك الصناعي.

NOMENCLATURE

symbol	Description	unit
C.V.	Fuel calorific value	kJ/kg
H	Total heat drop	kJ/kg
h	Specific enthalpy	kJ/kg
m	Mass flow rate	kg/s
p	Pressure	bar
P	Power	MW
PHR	Power to heat ratio	kWe/kWh
Q	Rate of Heat Energy	MW
s	Specific entropy	kJ/kg K
v	Specific volume	m^3/kg
w	Work	kJ/kg

Greek symbols

symbol	Description	unit
Φ	Irreversibility	kJ/kg
Ω	Irreversibility coefficient	-
η	Efficiency	%
μ	Relative cross section area of valve	-
Δ	Difference	-
α	Relative steam consumption in the regenerative extraction	-

Subscripts

symbol	Description
cw	Cooling water
f	Saturated liquid
g	Saturated vapour
II	2^{nd} law of thermodynamic
j	Number of feed water heater
o	Design condition
p	Heat process
pro.	Proposal method
conv.	conventional method
W	water
e	Electric
h	Heat
d	Design, developed

Abbreviations

symbol	Description
CHP	Combined Heat and Power
HRSG	Heat Recovery Steam Generation

INTRODUCTION

Cogeneration is the simultaneously production of electrical energy and useful thermal energy from the same energy source. In conventional electricity generation, only a small portion of fuel energy is converted into electricity and the remaining is lost as waste heat. Cogeneration reduces this loss by recovering part of this. Principal applications of cogeneration include industrial sites, district heating and buildings [1]. The two types of cogeneration plant most widely used are the back-pressure and the extraction-condensing types. The choice between back-pressure turbine and extraction-condensing turbine depends mainly on the quantities of power and heat, quality of heat, and economic factors. The extraction points of steam from the turbine could be more than one, depending on the temperature levels of heat required by the processes [2]. For back-pressure steam turbine cogeneration plant, only the pressure of steam extracted from back-pressure turbine to industrial purposes is regulating and its temperature varies greatly with heat demand. So, the superheating rate of the supplied steam reaches 150-200°C, at the same time some chemical plants required also saturated steam. To regulate this temperature, desuperheated is located behind the turbine in which injection water is used [3].

Some researchers investigated the sliding live steam temperature method (changing the inlet steam temperature at turbine inlet) to control electric load covered by cogeneration plant [3].

Analysis of power generation systems are of scientific interest and also essential for the efficient utilization of energy resources. The most commonly-used method for analysis of an energy-conversion process is the first law of thermodynamics. However, there is increasing interest in the combined utilization of the first and second laws of thermodynamics [5]. According to the second law of thermodynamic different criteria are defined for analysis the performance of power plants based on the concept of exergy (availability). If all of these criteria are used, they must all give the same results. Although availability pinpointed the real losses of a steam power plant, it is difficult, complex and cannot give direct relationship between component losses and overall efficiency of plant. Thus, the criteria for selecting the best procedure to evaluate thermodynamic analysis should be best ease of use, best degree of correspondence with the viewpoint and background of intended users and greatest breadth of application. On these basis, the entropy method (lost work) approach was believed to be superior to other approaches in common use [6]. This paper objective is sliding live steam temperature to regulate the steam temperature at exit of back-pressure turbines. The present study is aimed at quantification of the influence of the suggested method on the first and second law parameters and irreversibility losses in different components of cogeneration plant and compares the results with those for conventional method (water injection). Based on the mathematical model simulation of cogeneration plants with four types of back-pressure turbine was developed. It admitted the analysis of different operating regimes (design and off design) for these plants.

DESCRIPTION OF CASE STUDIES

Schematic diagrams of the four back-pressure turbines, considered for the present study are shown in Figures (1, 2, 3&4). Figure(1). Represents the flow diagram of back-pressure turbine type (R-12/90-18) which is a single-cylinder of 12MW capacity. The steam pressure and temperature at inlet are p_o=90bar, t_o=535ºc respectively and the back pressure is limited between 15 and 21bar.This type has single-row control stage (regulating stage) followed by 5 pressure stages. One extraction point is provided for the regenerative feed water heating system. Figure (2) illustrates the flow diagram of back-pressure turbine type (R-40/127-31) which is a single-cylinder of 40MW capacity .For this type, the live steam pressure and temperature are p_o=127bar, t_o=565ºc respectively and the back-pressure is limited between 29 and 31bar.This turbine has two-row control stage followed by 5 pressure stages. Two extraction points are provided for the regenerative feed water heating system. Figure (3) shows the flow diagram of back-pressure turbine type (R-50/130-13) which is a single-cylinder of 50MW capacity. The live steam pressure and temperature are p_o=130bar, t_o=565ºc respectively and the back-pressure is limited between 7 to 21bar. The turbine has single-row control stage followed by 16 pressure stages and three extraction points provided for the regenerative feed water heating system. Fig.4 shows the flow diagram of back-pressure turbine type (R-100/130-15) which is a single-cylinder of 100MW capacity .Also, this turbine has single-row control stage followed by 12 pressure stages and three extraction points provided for the regenerative feed water heating system. All case studies used closed cascaded backward heater type. The three case studies R-12, R-40 and R-100 have nozzle control governing, steam being supplied to the control stage through four regulating valves. The first two valves are opened simultaneously, while the others are opened in succession while the case study R-50 contains additional valve which opens simultaneously with the fourth valve.

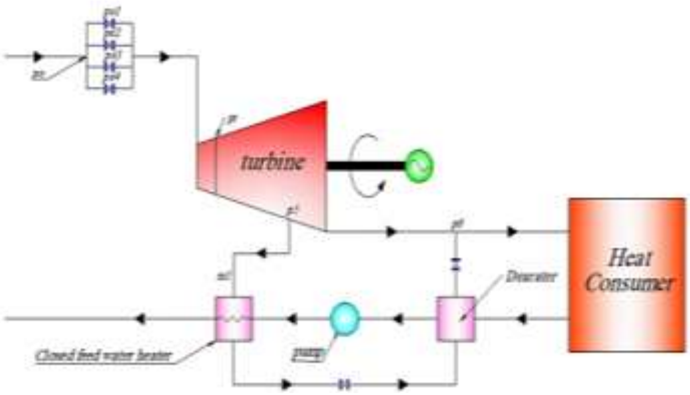


Figure (1) Flow diagram for Back-pressure turbine type (R12-90-18).

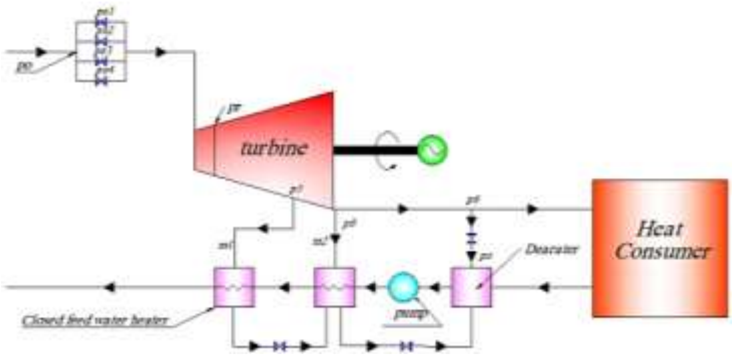
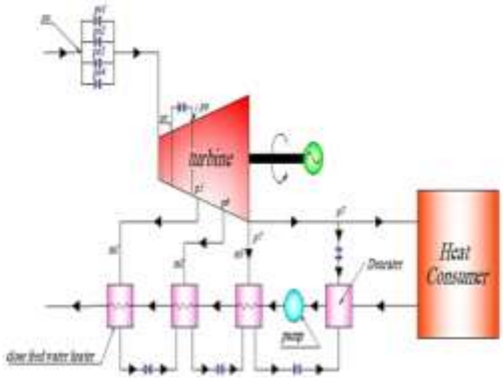


Figure (2) Flow diagram for Back-pressure turbine type (R40-127-31).



Figure(3) Flow diagram for Back-pressure turbine type (R50-130-13).

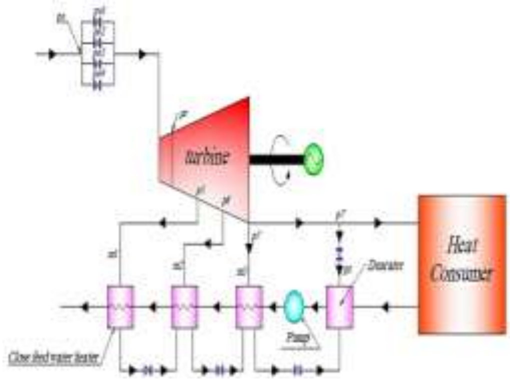


Figure (4) Flow diagram for Back-pressure turbine type (R100-130-15).

MATHEMATICAL MODEL AND THERMODYNAMICS PARAMETERS

Mathematical model

Simulation model of cogeneration plant is built to allow system simulation over a rather wide range of operation (non-linear model) and this based on the representation of plant components and of their inter connections. This model is referred to as dynamic, that is, it is able to predict transient response, even for large process variations [7]. The model deals with many types of back pressure turbine depend on constructive and functional characteristic of cogeneration steam turbine stages. For the purpose of power plant simulation, turbines are general modeled as lumped parameters. When accurate modeling is required, it is usual to split a turbine into number of cascaded sections. A section being in turn composed of number of cascaded stages [7]. The turbine stages belong to two categories.

1. Control stage

2. Pressure stage

Actual enthalpy drop in the control stage for any mass flow rate can be found by using [3]

$$\Delta \mathrm{h} = -316.98 \cdot \left(\frac{\mathrm{p_g}}{\mathrm{p}}\right)^{1.25} + 273 \qquad \dots (1)$$

Where $\mathrm{p_g}$ is the governing stage pressure and p is the steam pressure behind any valve. While the isentropic efficiency of pressure stages is defined by [3].

$$\eta_s = \left[0.915 \cdot \frac{0.3}{m_s \cdot v}\right] \cdot \left[1 + \frac{\Delta h_{is} + 1200}{25000}\right] - 0.03 \qquad \dots (2)$$

Where $m_s$the mass flow rate of steam through the group of stages is v is the specific volume at inlet of group of stages and Δh_{is} isentropic enthalpy drop through the group.

An important step into the method is pressure distribution calculus. As a basis for steam turbines pressure calculus is design and off design flow distribution. With the assumption that medium temperature variation is ignorable, pressure through turbine stages can be calculated by using Stodola equation [8].

$$\frac{m}{m_o} = \sqrt{\left(\frac{{p_1}^2 - {p_2}^2}{{p_{1_o}}^2 - {p_{2_o}}^2}\right)} \qquad \dots (3)$$

Where p_{1_0}, p_1 are steam pressures before the first or any other stage at design load and at the load under consideration respectively, p_{2_0}, p_2 are steam pressures after the first or any other stage at design load and at the load under consideration respectively.

For the feed water heaters the model equations are derived from the mass and energy balances to each type [3].

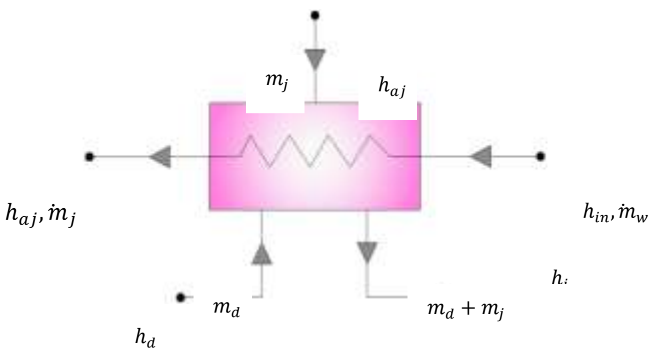


Figure (5) flow diagram for closed feed water heater.

$$m_j = \frac{m_{wj}\cdot(h_{out}-h_{in})-\sum m_d\cdot(h_d-h_j)}{h_{aj}-h_j} \qquad \dots (4)$$

Where m_j is the mass flow rate of the extracted steam, m_{wj} is the mass flow rate of the water entering the heater, h_{in} and h_{out} are the enthalpies of feed water that enters and leaves the heater, h_{aj} is the enthalpy of extracted steam, m_d is the mass flow rate of condensing steam and h_j is the enthalpy of condensing steam where j is the number of extraction point.

Based on the mathematical model, computer program has been written to work under MathCad software. The program allows analyzing cogeneration plants with different types of back-pressure turbine under design and off design conditions and different control programs of exit steam temperature. According to the first and second laws of thermodynamics the following parameters are studied:

Thermodynamics parameters

1- Irreversibility coefficient

According to the second law of thermodynamics, irreversibility coefficient (Ω) for each component of the plant is given by [6]:

$$\Omega_i = \frac{\Phi_i}{m_f.C.V.} \qquad \dots (5)$$

Where, $\Phi_i = T_o \cdot [\sum_i^n (m_i \cdot s_i)_{out} - \sum_i^n (m_i \cdot s_i)_{in}]$ … (6)

And the overall irreversibility coefficient of plant is,

$$\Omega_{total} = \frac{\sum_{i=1}^n \Phi_i}{m_f.C.V.} \qquad \dots (7)$$

$$\Omega_{total} = \Omega_1 + \Omega_2 + \cdots\cdots + \Omega_n = \sum_{i=1}^n \Omega_i \qquad \dots (8)$$

Then the thermal efficiency is defined as,

$$\eta_{II} = 1 - \sum_{i=1}^n \Omega_i \qquad \dots (9)$$

2- Rate of heat process

The rate of process heat (industrial heat demand) is,

$$Q_p = \dot{m}_p \cdot (h_{exit} - h_{cond.}) \qquad \dots (10)$$

Where: $\dot{m}_p$ is the steam mass flow rate supplied to heat consumer, h_{exit} is the enthalpy of exit steam turbine and $h_{cond.}$ is the enthalpy of condensated water from process system.

3- Power developed

The power developed of steam turbine is given by:

$$P_d = H_o - \sum_0^j \alpha_j \cdot (H_o - H_{oj}) \qquad \dots (11)$$

Where: H_o: is the total heat drop of turbine, α_j: is the relative steam consumption in the regenerative extraction and H_{oj}: is the total heat drop from extraction point.

4- Power to heat ratio

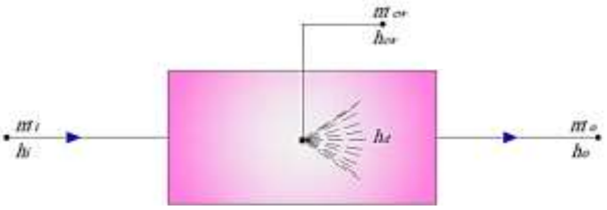
It is defined as the ratio of electricity to rate thermal energy (heat demand)

$$PHR = \frac{P_d}{Q_p} \qquad \dots (12)$$

5- Injection water flow rate

The cooling water requirement to desuperheater (injection water flow rate) can be determined as [4],

$$\dot{m}_{cw} = \frac{\dot{m}_s\left(h_{s_i}-h_{s_o}\right)}{\left(h_{cw_o}-h_{cw_i}\right)} \qquad \dots (13)$$



h_{so}, m_s

Figure (6) flow diagram for desuperheater.

Where h_{s_i} and h_{s_o} are the enthalpies of steam that enters and leaves the desuperheater, h_{cw_i} and h_{cw_o} are the enthalpies of cooling water that enters and leaves the desuperheater

6- Heat added to cogeneration plant

At the same power output and rate of heat process for the conventional and the suggested methods, the heat added to the cogeneration plant can be calculated according to the first law of thermodynamic as in following;

(a) Conventional method

$$Q_{add} = power\ output + rate\ of\ process\ heat + \frac{(power\ output)_{sug.} - (power\ output)_{conv.}}{\eta_c} \quad \text{If} \quad (power\ output)_{sug.} > (power\ output)_{conv.}$$

η_c: Efficiency of condensing plant ≈ 0.41

$Q_{add} = power\ output + rate\ of\ process\ heat$

If $(power\ output)_{conv.} > (power\ output)_{sug.}$

(b) Suggested method

$$Q_{add} = power\ output + rate\ of\ process\ heat + \frac{(power\ output)_{conv.} - (power\ output)_{sug.}}{\eta_c} \quad \text{If} \quad (power\ output)_{conv.} > (power\ output)_{sug.}$$

$Q_{add} = power\ output + rate\ of\ process\ heat$

If $(power\ output)_{sug.} > (power\ output)_{conv.}$

7- Fuel saving

Then the percentage of fuel saving (at the same boiler efficiency and fuel) is simply calculated by the following equation,

$$\Delta\dot{m}_f = \frac{Q_{add)sug.} - Q_{add)conv.}}{Q_{add)conv.}} * 100\% \qquad \ldots (14)$$

RESULTS AND DISCUSSION

Steam temperature at exit turbine

To satisfy the process heat demand steam flow rate through the turbine has changed. According to any flow rate (regime), the control valve position has changed, so the exit steam temperature also changed. Figure (7) shows the variation of steam temperature behind the turbines respective to the heat process demand (for third and fourth case studies). From Figures, it was shown that the temperature increases when the valve is partially opened (low heat demand) due to throttling losses, which causes the expansion to move to the right side on the h-s diagram (superheated region). While the process behaves completely in an opposite way when the valves are fully opened.

Since the back-pressure turbine operates according to heat demand it was expected that the steam has supplied to industrial consumers with variable temperature. For this reason, nowadays, back-pressure turbine operates with desuperheater (injection water) to keep constant steam temperature regardless of the heat demand.

Figure (8) represents the relation between mass flow rate of injected water and the relative heat process demand at several back pressures for four case studies. For a certain back-pressure, the variation of mass flow rate of injected water to keep constant steam temperature after turbine is not uniform, it depends on the temperature difference between the exit turbine temperature (before desuperheater) and the steam temperature required by the heat consumer (after desuperheater). Generally as the relative heat demand increases, the mass flow rate of injecting water in the desuperheater will decrease until reaching zero whereas the relative heat demand equal to 1.

Also, increasing the exit back-pressure leads to decrease the rate of injecting water at the same relative heat process demand. This can explained as the difference between exit temperatures when all the control valves are opened and any another regime decreases as the back-pressure increases especially at low relative heat demand. The figure also shows that this effect diminishes with increasing turbine capacity.

Figure (9) shows the relation between sliding live steam temperature (steam temperature at the outlet of the boiler at design conditions minus steam temperature at the outlet of the boiler required to keep constant temperature behind the back-pressure turbine) and the relative heat process demand for first and second case studies. From figures, it is clear that as the relative heat process demand increases (for a fixed back pressure), the sliding live steam temperature decreases. This is because increasing heat demand leads to an increase in the steam flow rate through turbines and consequently the throttling losses through control valves decrease and as a result the exit steam temperature decreases. When all the valves are fully opened (relative heat demand=1), these losses disappear and the sliding live steam temperature becomes equal to zero.

As well as, the increasing of exit back-pressure of the turbine leads to a decrease in the sliding live steam temperature

Power to heat ratio

To satisfy the process heat demand, steam flow rate through the turbine must be changed and consequently, the power developed also varies. From Fig(10), it is shown that the power to heat ratio is increasing with an increase in the relative heat demand, for first case study. This figure also shows that power to heat ratio increases with decreasing back pressure. This is because decreasing back pressure leads to a decrease in the thermal energy supplied to the consumer and at the same time the power developed increases due to an extend expansion through the turbine. Complete results obtained in the present study have shown that similar behaviors were also observed for other case studies.

Figure (11) shows a comparison of power to heat ratio between the conventional method and the suggested method. At low relative heat demand the power to heat ratio for the suggested method is greater than that for the injecting water although this effect is less pronounced when Q_p/Q_{pd} >0.5. Power to heat ratio for the conventional method decreases because steam flow rate through turbine decreases as a result of injected water to keep constant heat supplied to the consumer. Although the suggested method leads to a decrease in the power due to decreasing live steam temperature but its effect is less than that for the conventional method and as a result, the power to heat ratio for the sliding live steam temperature is greater.

Saving of fuel

According to the first law of thermodynamics, the percentage of fuel saving was determined. The fuel consumption for sliding live steam temperature and water the injection methods is calculated at the same power and heat demand. It was shown from Figure (12),it is clear that, the fuel consumption of suggested method is less than that for the conventional method. The value of percentage of fuel saving varies with heat demand and increases as the heat demand decreases. Increasing fuel consumption of the conventional method can be explained as decreasing the power developed from the turbine. To substitute this defect of power, another condensing power plant with efficiency = 41% was used. It is known as power plant with back-pressure turbine has the highest efficiency compared with all types of thermal plants (there is no rejected heat) and as a result the fuel consumption for water injection method increases. From these figures, it was shown that the effects of variation of the back pressures and the turbine capacity on the percentage of fuel consumption are nearly diminished.

Efficiencies and irreversibilities of combined heat and power plant

Figures (13, 14, 15 and 16) represent the comparison of the second law plant efficiency of each case study, between conventional and suggested method with various relative heat process demand. These figures show that the suggested method is more efficient than the conventional method and the improvement value of the efficiency for the cogeneration plant operates with turbine type (R-12/90-18) is about (0-0.54%) and this improvement is about (0-0.33%), (0-0.52%) and (0-0.78%) for cogeneration plants operate with turbine (R-40/127-18), (R-50/130-13) and (R-

100/130-15) respectively depending on heat demand. This is because sliding temperature method rearranges the irreversibility losses through the plant components. Although this method increases boiler losses due to a decrease in steam temperature (increase heat transfer losses) at the same time it leads to the removal of the desuperheater component and lifts its additional losses.

Figures (17, 18,19 and 20) illustrate the simplified process diagrams of exergy for the cogeneration power plants operate with the back pressure turbine (R-12/90-18),(R-40/127-31), (R-50/130-13) and (R-100/130-15) respectively with the conventional method (water injection). The performance of the plant was estimated according to the results of entropy analysis method. The exergy of the heat released upon combustion of fuel in the boiler furnace is assumed to be 100%; the diagrams show how the fraction of the exergy flow is spent as the loss of availability in each particular element of these cogeneration power plants. These diagrams indicate that the boiler is the major source of irreversibilities, where the exergy destruction occurred in the boiler was about 60.7%, 58.5%, 58.3% and 58.3% of fuel exergy input respectively. The second source of irreversibilities is the turbine about 2.6%, 2.6%, 2.9% and 2.7% of the fuel exergy input respectively. This agrees well with the analysis given by ref. [5] and ref. [10] .The exergy (entropy) efficiencies of the plants are 34.8%, 36.11%, 36.087% and 35% respectively.

The sliding live steam temperature method is the key aspect in improving the exergy efficiency of the system, where the element (desuperheater) is removed. Figure (21, 22, 23 and 24) represent the simplified process diagram of exergy for cogeneration power plant with the suggested method) with back-pressure turbine (R-12/90-18), (R-40/127-31), (R-50/130-13) and (R-100/130-15) respectively. A comparison between the suggested method and the conventional method show that the exergy losses for each component, approximately, are equal except the losses that occur in the desuperheater in conventional method (about 0.136%,0.155%, 0.053% and 0.227% of the fuel exergy input for CHP with back pressure turbine (R-12/90-18),(R-40/127-31), (R-50/130-13) and (R-100/130-15) respectively).Also, these Figures specify that the exergy efficiencies are about 35%, 36.27%, 36.135% and 36.63% respectively, so the suggested method is more efficient than the conventional method.

Mathematic model verification

In order to verify the mathematical model, the design data was used as the source of information of modelling. Validation versus design data is a based step towards certification [12]. A comparison between the developed powers obtained from simulation for R-40/127-31 and for R-50/130-13 with reference [11] is shown in Fig. 25 and Fig. 26 where maximum deviation is around 2%. For designing regime, the power developed for the turbine (R-12/90-18),(R-40/127-31), (R-50/130-13) and (R-100/130-15) obtained from the simulation have maximum deviation from design capacities equal to 1.5%, 2%, 0.6%, 1% respectively.

CONCLUSIONS

The following conclusions are drawn from the present work:

1. The good agreement of the calculated results with the design parameters confirms the correctness of the developed mathematical model

2. According to the operation regime (heat demand), the steam temperature behind the back-pressure turbine changes and its value becomes the highest when the control valves are partially opened.
3. In order to keep constant heat supplied to consumer, the conventional method can be used to decrease the steam flow rate through the turbine, so the power to heat ratio decreases.
4. Although the suggestion method leads to a decrease in the power developed, its effect is less than the conventional method and as a result the power to heat ratio for this method is greater.
5. Sliding live steam temperature doesn't cause any extra cost and can be used without any additional arrangement.
6. The suggested method causes fuel saving in comparison with conventional method. Depending on heat demand, the percentage of fuel saving was about (0-15%), (0-18%), (0-21%) and (0-24%) for back-pressure turbine (R-12/90-18), (R-40/127-31), (R-50/130-13) and (R-100/130-15) respectively.
7. According to the second law analysis, cogeneration plant with the suggested method is more efficient.
8. The capacity variation of back-pressure turbine doesn’t cause any change of the character of control methods.

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[11]. Steam turbine plants, “Steam Turbines Types R-40/127-31 and R-50/130-13 Catalogue,” LMZ company, sheet No.13 (in Russian).
[12].Flynn, D., “Thermal Power Plant Simulation and Control,” The Institution of Electrical Engineers, London, United Kingdom, 2003.

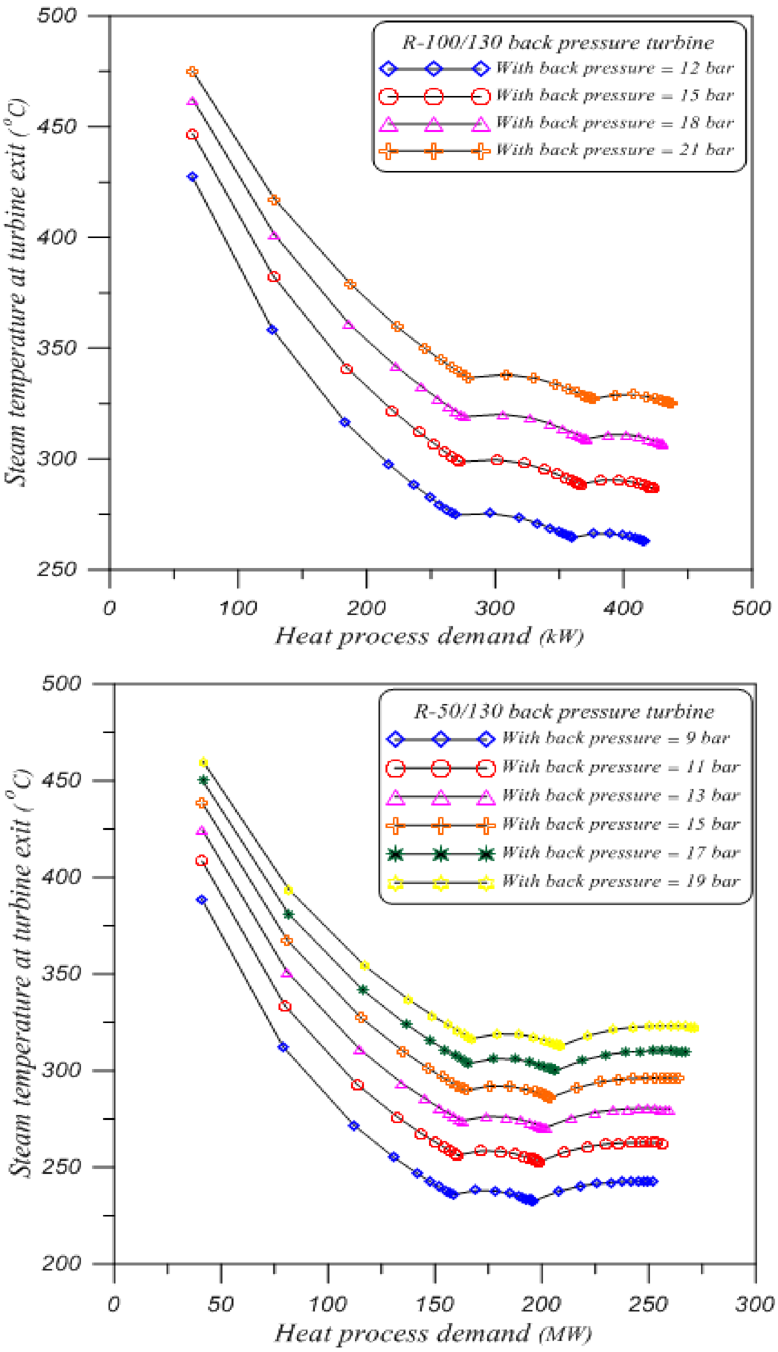
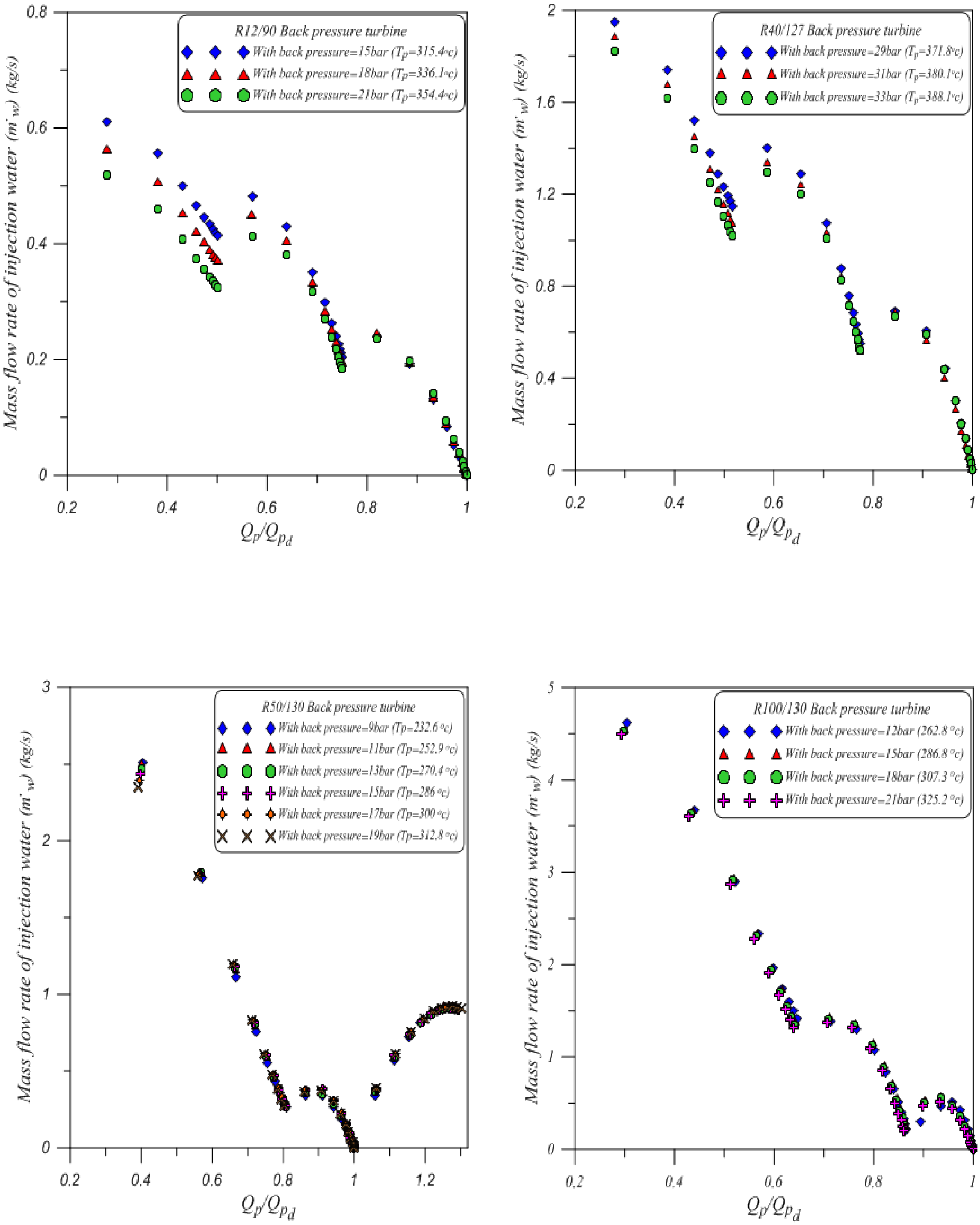


Figure (7) Variation of steam exit temperature with heat process Demand at third and fourth case studies.



Figure(8) Variation of injection water with relative heat process demand to keep constant steam temperature at various back pressures.

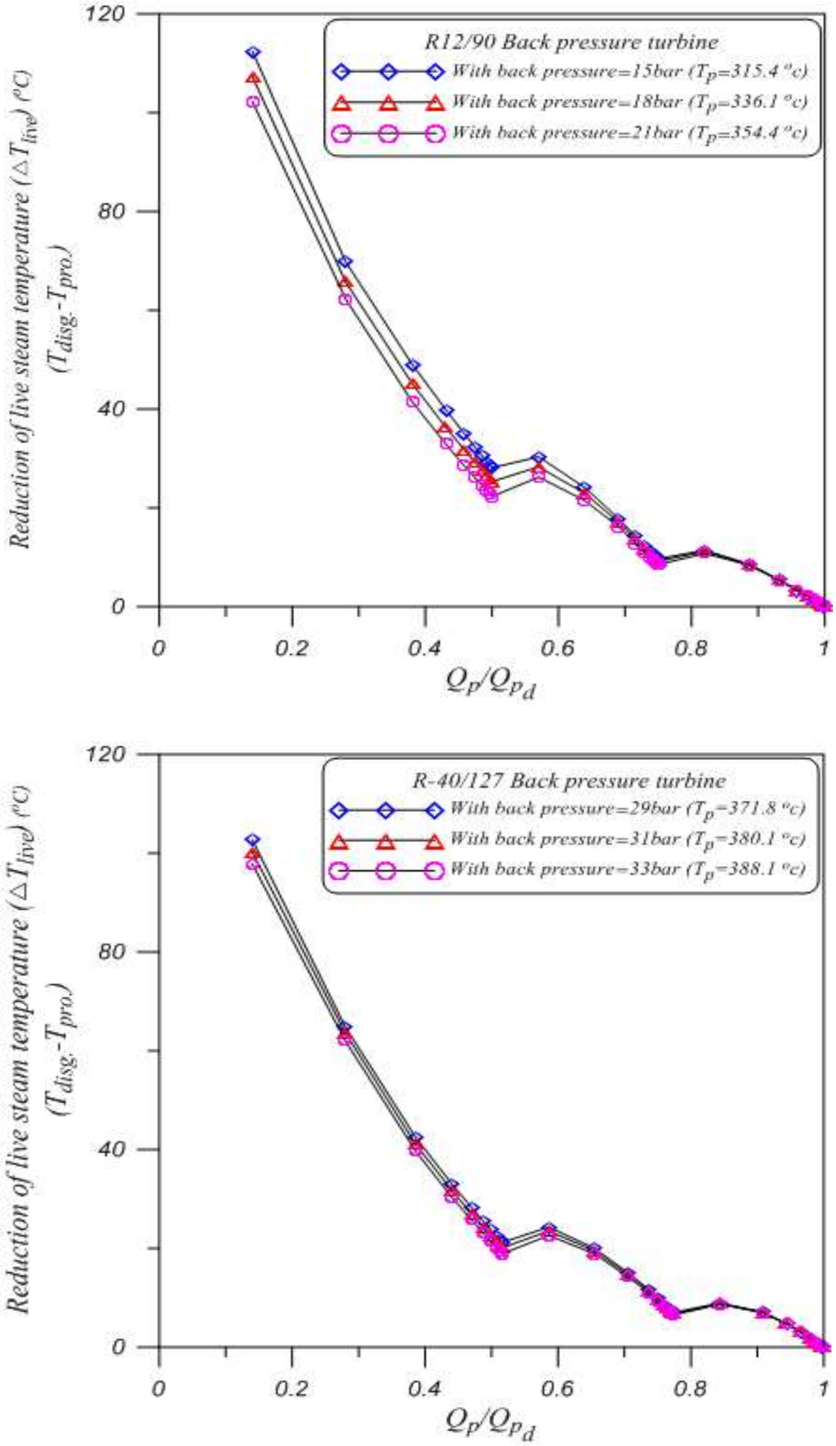
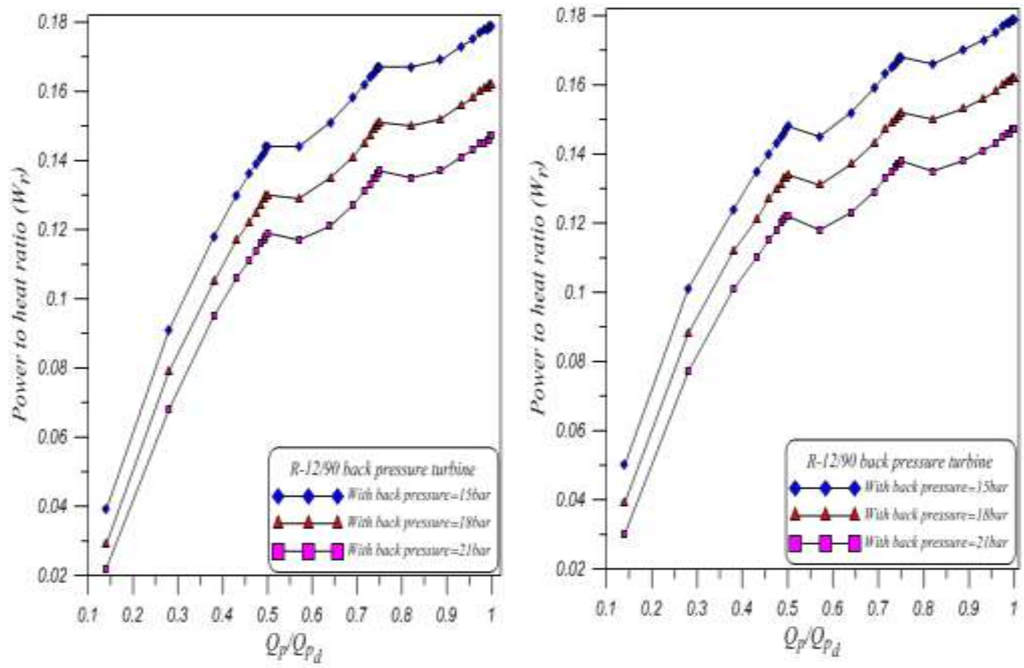


Figure (9) Reduction of live steam temperature versus relative industrial heat demand to keep constant steam temperature at first and second case studies.



(a) (b)

Figure (10) Power to heat ratio versus relative heat process demand for various back pressures for first case studieswith conventional control method (b) With suggested control method.

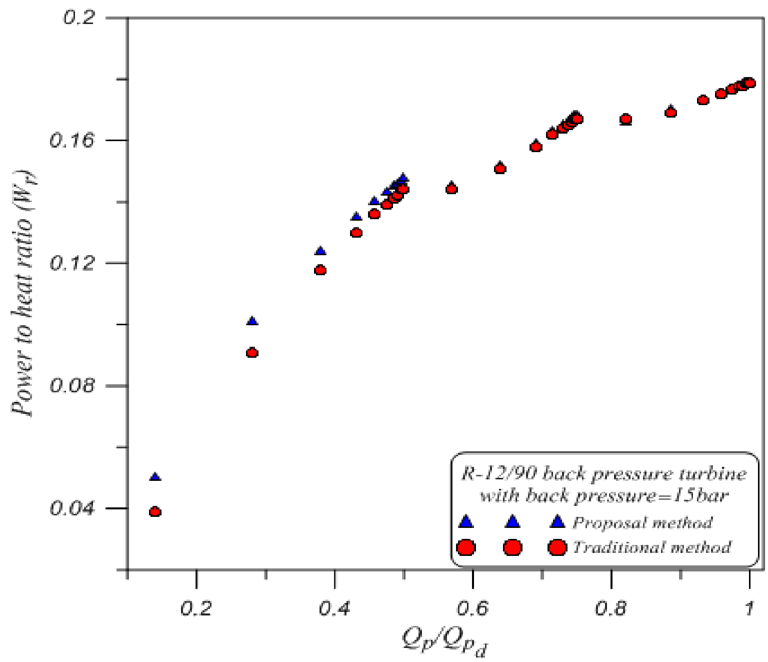
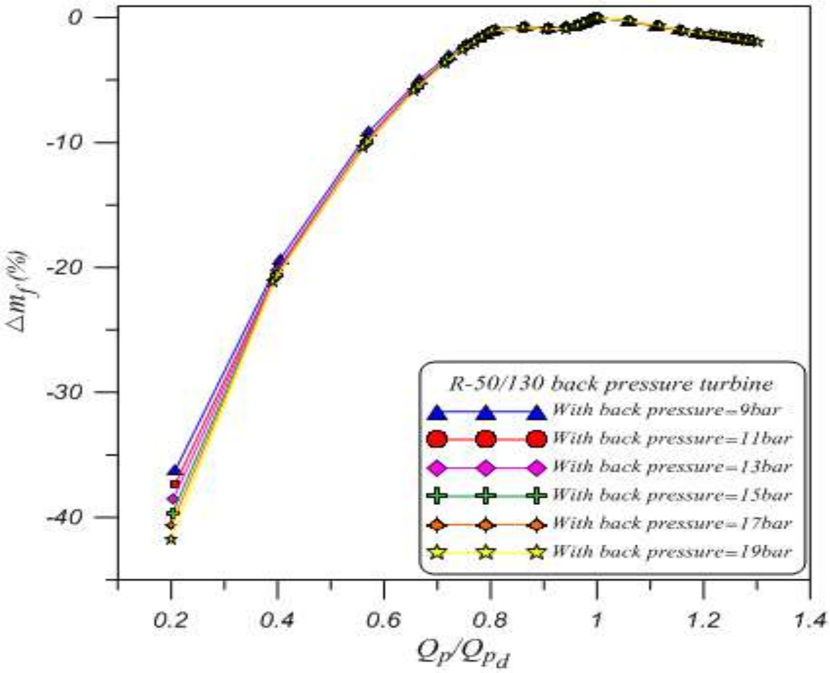


Figure (11) Comparison of Power to heat ratio versus relative heat Demand for conventional and suggested control method for first case study at back pressure 15bar .



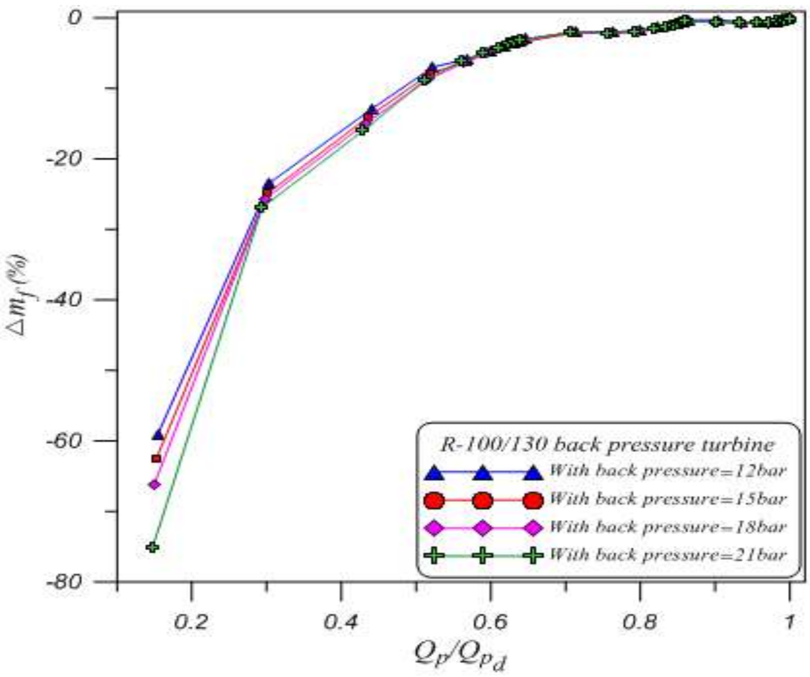


Figure (12) Fuel saving due to use suggested method at various Relative heat demands and various back pressures at third and second case studies.

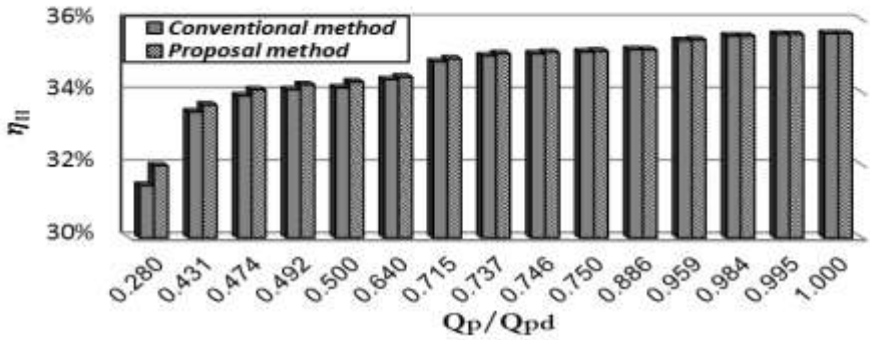


Figure (13) Second law efficiency with various relative heat process Demands for first case study.

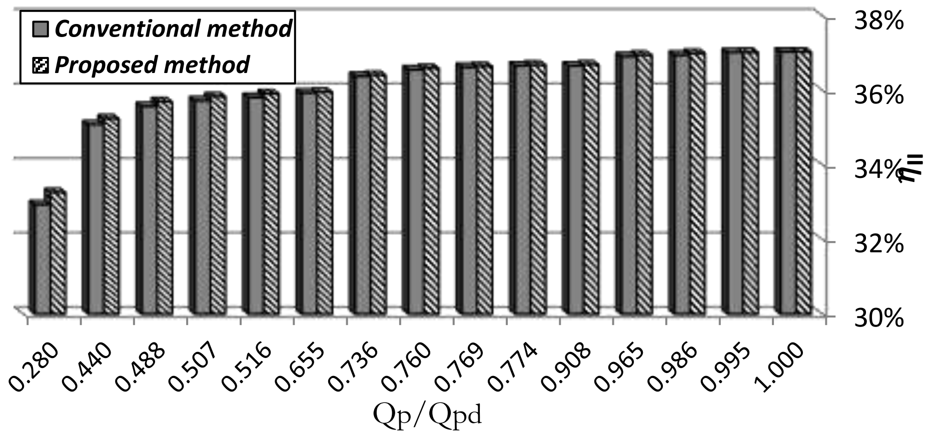


Figure (14) Second law efficiency with various relative heat processes Demands for second case study.

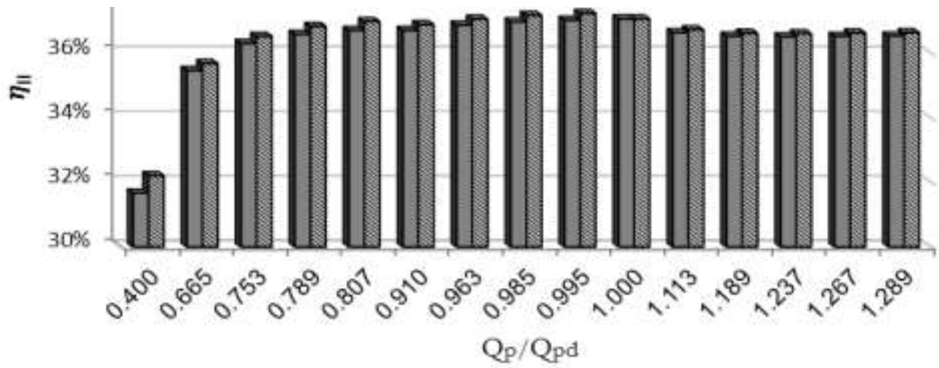


Figure (15) Second law efficiency with various relative heat processes Demands for third case study.

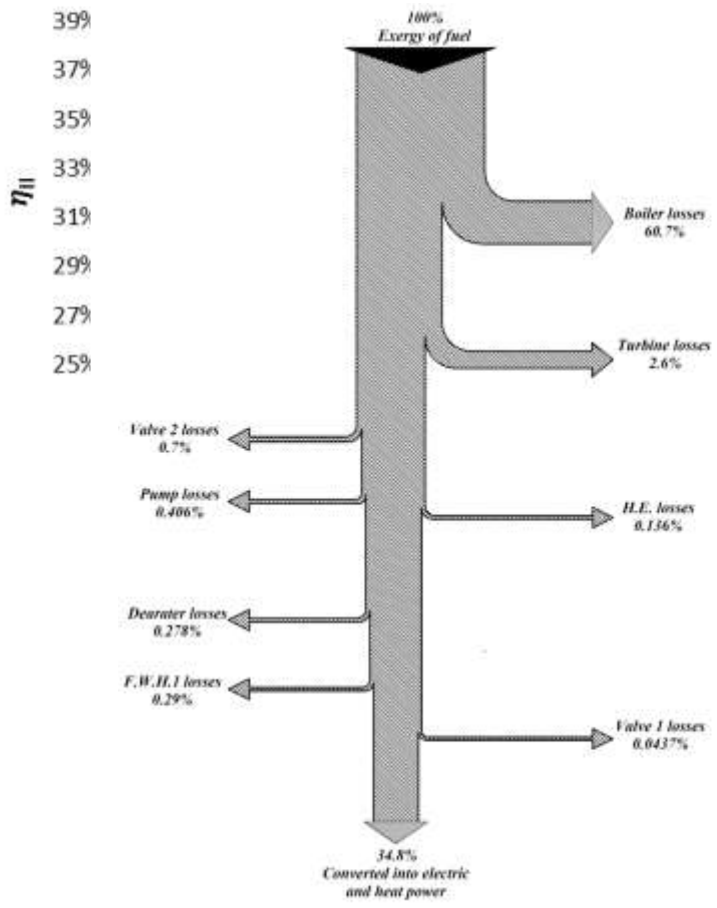


Figure (16) Second law efficiency with various relative heat processes Demands for fourth case study.

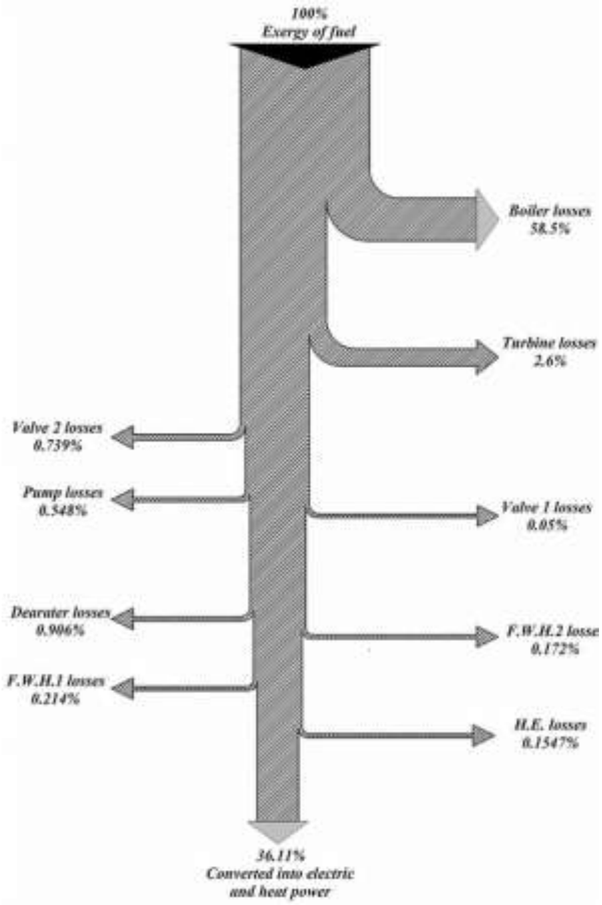


Figure (17) Exergy (Entropy) simplified flow diagram of CHP with R-12/90-18 turbine (Conventional method).

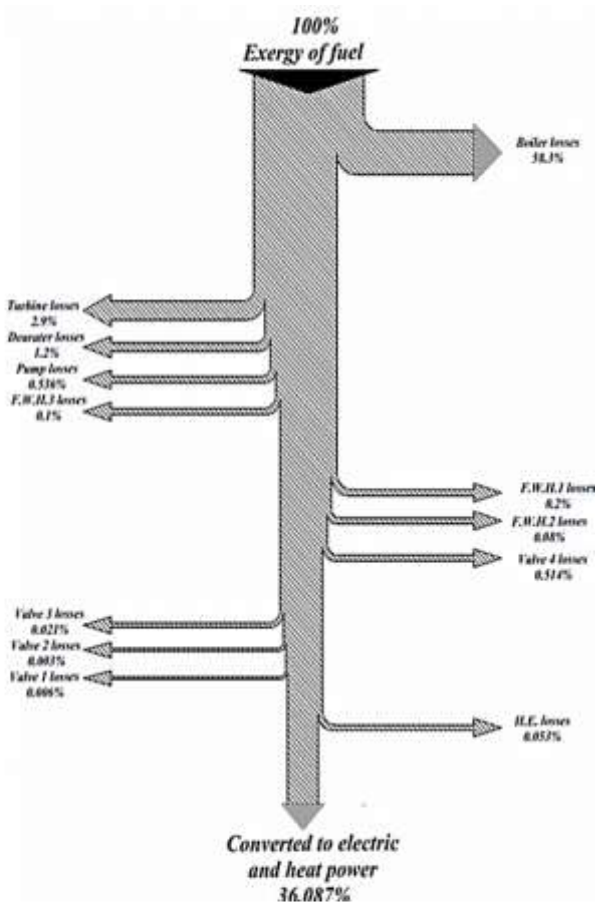


Figure (18) Exergy (Entropy) simplified flow diagram of CHP with R-40/127-31 turbine (Conventional method).

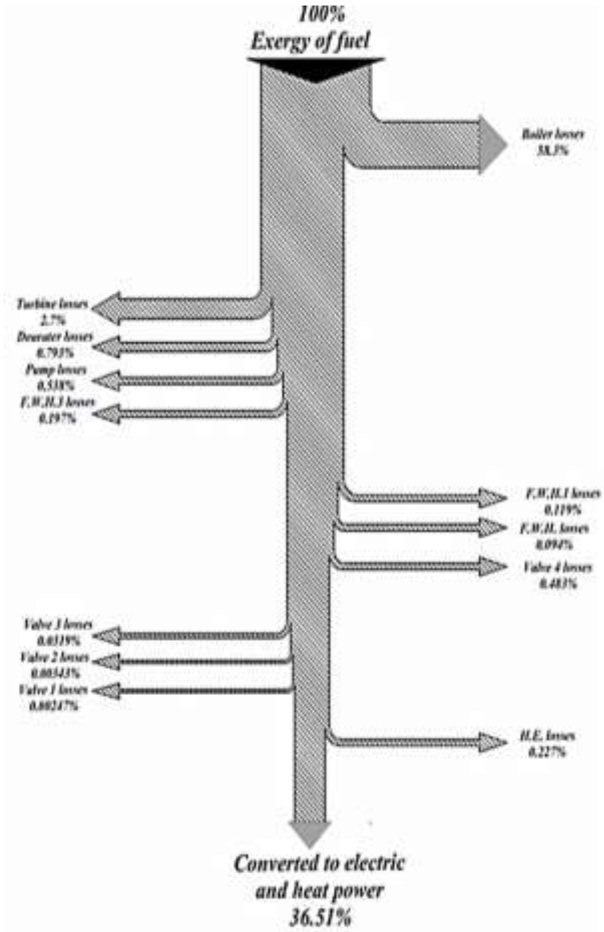


Figure (19) Exergy (Entropy) simplified flow diagram of CHP with R-50/130-13 turbine (Conventional method).

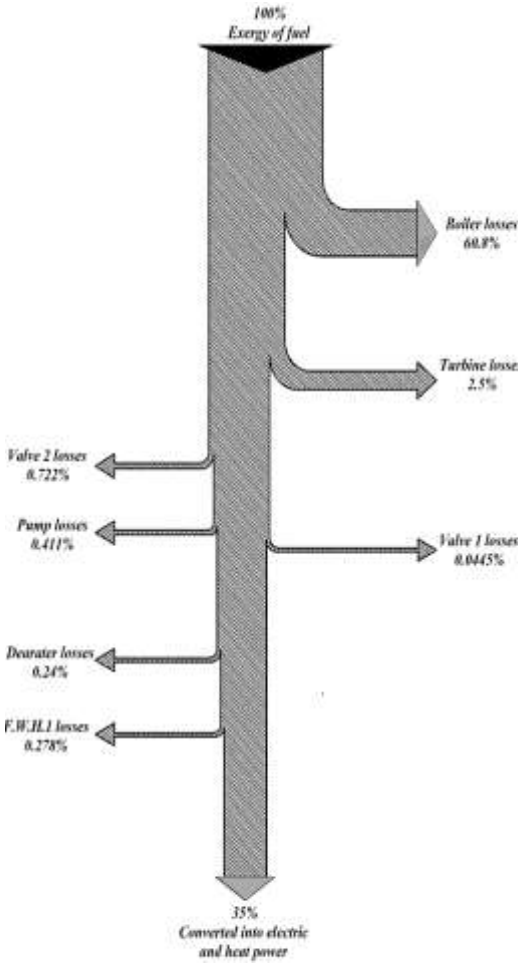


Figure (20) Exergy (Entropy) simplified flow diagram of CHP with R-100/130-15 turbine (Conventional method).

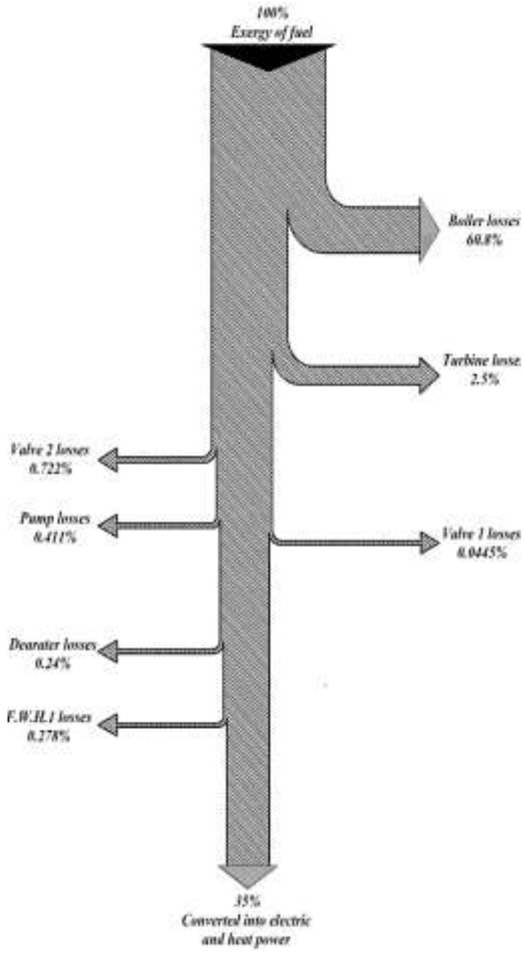


Figure (21) Exergy (Entropy) simplified flow diagram of CHP with R-12/90-18 turbine (suggested method).

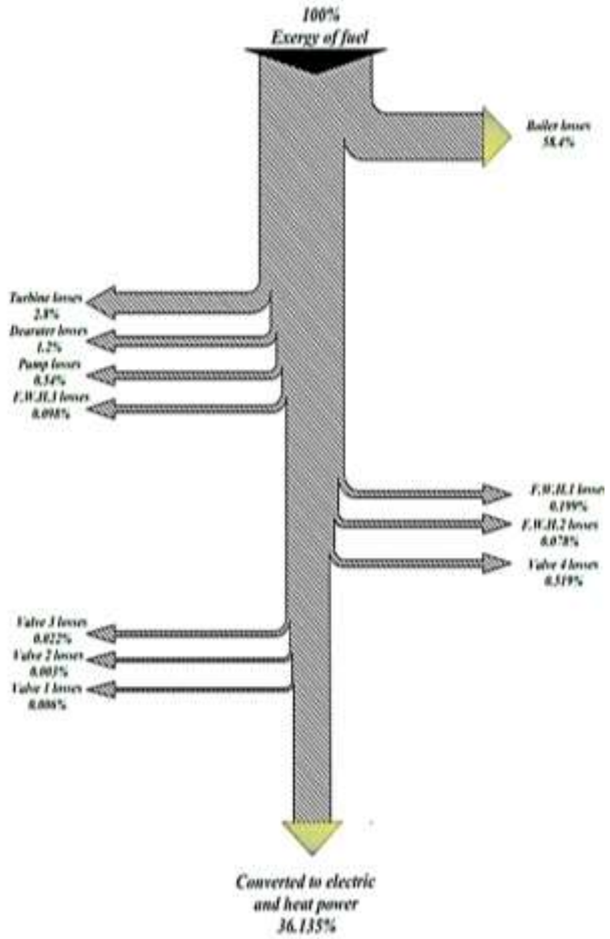


Figure (22) Exergy (Entropy) simplified flow diagram of CHP with R-40/127-31 turbine (suggested method).

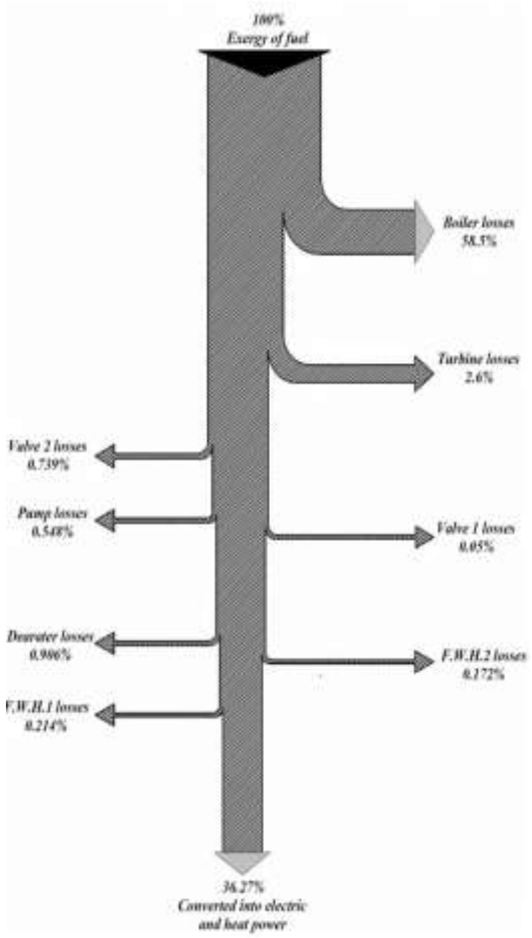


Figure (23) Exergy (Entropy) simplified flow diagram of CHP with R-50/130-13 turbine (suggested method).

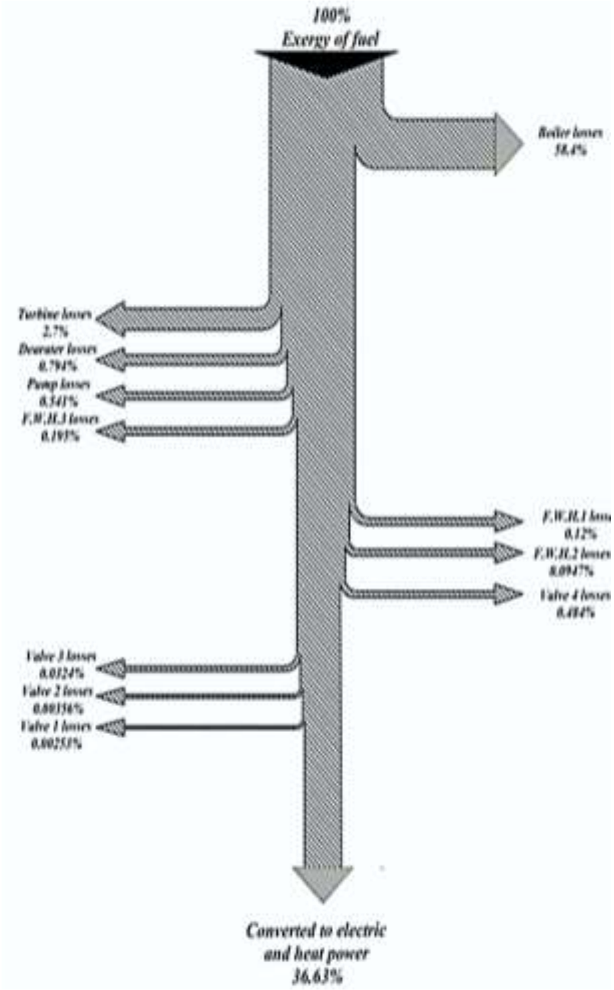


Figure (24) Exergy (Entropy) simplified flow diagram of CHP with R-100/130-15 turbine (suggested method).

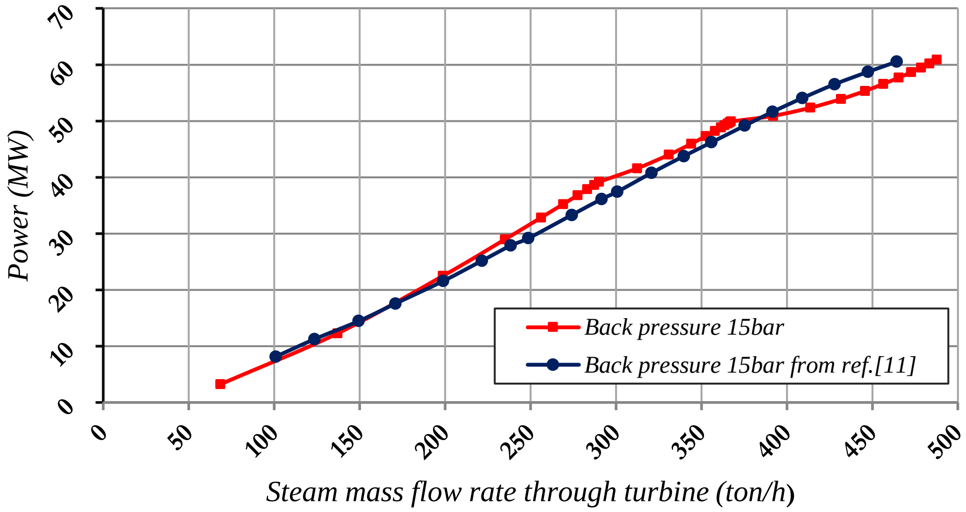


Figure (25) Comparison the variation of power developed with Steam mass flow rate between the result of simulation and reference [11] for turbine (R-50/130-13).

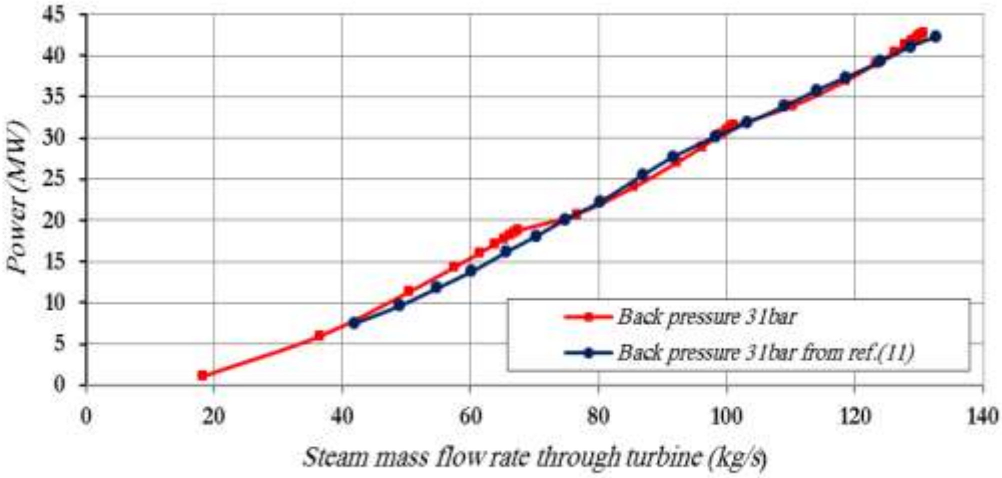


Figure (26) Comparison the variation of power developed with Steam mass flow rat between the results of simulation and reference [11] for turbine (R-40/127-31).