Modeling and Simulation for Condensing and Cogeneration Steam Power Plants Operate at Constant and Sliding Live Steam Pressure

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Received on: 7/9/2014 & Accepted on:21/4/2016

ABSTRACT

The present paper deals with a theoretical study of condensing and cogeneration plants operate at partial load with constant and sliding (variable) live steam pressure controls. In this work, two Iraqi condensing units as well as two cogeneration with steam back-pressure turbine were chosen. These units are K-66-87-0.07, K-55-58.8-0.083, R-100-130-15 and R-40-130-29 respectively. A computer program had been written to work under MathCad 15 software to simulate these units under design and off design regimes with both types of control at nozzle and throttling steam distribution. The performance of the different schemes is analyzed in view of the first and second laws of thermodynamics. The results show that the selection of control type mainly depends on type of steam distribution. So, the heat rate (k) increases in condensing units with sliding live steam control and nozzle distribution according to first law of thermodynamics. The value of increasing (k) is about (0-6%) depending on the operation regime. While, this type of control with throttling steam distribution causes decreasing (k) in about (0-1%). Cogeneration units with back-pressure steam turbine operate only with nozzle distribution. So, the results show that using sliding live steam pressure control is associated with increasing heat rate (k), especially when ratio of flow rate is ≤ 0.9. This type of control for cogeneration units also causes increasing heat process directed to heat consumer and decreasing power to heat process ratio (α). According to the second law of thermodynamics the irreversibility losses were redistributed depending on control type.

Keywords: Cogeneration plants, condensing turbine, sliding and constant pressure controls.

Nomenclature

Symbol	Definition	Units
m_s	Design mass flow rate through the valve	kg/s
m	The mass flow rate through the valve at any other operation regime	kg/s
m_1	Mass flow rate of first stream	kg/s
m_2	Mass flow rate of second stream	kg/s
h_0	Steam enthalpy at boiler outlet	kJ/kg
h_1,h_2	The enthalpy at valve outlet	kJ/kg
p_1	Steam pressure at valve outlet	MPa
p_D	Pressure at deaerator	MPa
p_r	Pressure behind control stage	MPa
Q_{add}	The rate of heat added	MW
$Q_{process}$	The rate of process heat	MW
q_c	Specific heat consumption during constant pressure control	kJ/kW .h
q_s	Specific heat consumption during sliding pressure control	kJ/kW .h

https://doi.org/10.30684/etj.34.13A.10
2412-0758/University of Technology-Iraq, Baghdad, Iraq

P	Work	kJ/kg

Greek

Symbol	Definition	Units
Δ	Difference	
η	Efficiency	
ε	Utilization factor	
α	Power to heat ratio	
η_m	Mechanical efficiency	
η_g	Generator efficiency	

Subscript

symbol	Definition
act	Actual
P	Pump
Is	Isentropic
T	Turbine

Abbreviations

Symbol	Definition
CHP	Combined heat and power
TTD	Terminal temperature difference

INTRODUCTION

Nowadays, electricity plays a vital role in improving the standard of live. The new tendency of condensing power plants is to operate these units, not in base load (close to nominal load), but in semi base load (essentially with load variation). Also cogeneration plants operate according to the heat demand and as a result they spend a significant time on partial load. The performance of a turbine when operates at load different from designed or economic load depends on the particular method employed for controlling the supply of steam to the turbine, so the rotation speed will remain sensibly constant, irrespective of the load. These methods of governing are [1]:

- Throttle governing
- **Nozzle governing**
- By pass governing

In throttle governing the primary aim in the off-design operation is to reduce the mass flow of steam. Throttle valve would open to its maximum travel only if the turbine is operating at full load. During partial load (off design condition), the throttle valve opens only a fraction of its travel. Thus, at partial loads, all the quantity of fresh steam fed to the turbine undergoes throttling accompanied by heat losses leading to increase the entropy and a corresponding decrease in available energy, which causing reduction of turbine efficiency.

In the system of nozzle governing, fresh steam enters the first stage nozzles through regulating valves. Each control valve regulates the supply of steam to its own group. Under conditions of operation at full load, all the regulating valves are fully open. When the load on the turbine varies, the nozzle valves open or close in a definite consecutive order, and hence the degree of partial admission varies with the load carried by the turbine. If these regulating valves are partially open, throttling dose take place as in the case of throttle governing. But, since through each of the regulating valves only a certain fraction of the total steam flows, the losses due to throttling are smaller than in the case of throttle governing where all the quantify of steam has to be undergone throttling to the same extent.

In by pass governing as a rule for turbines with throttle governing, it is usual to use the system of external bypass. With this type of bypass, usually the turbine would be developing the most economic capacity (design condition) when the main throttle valve is fully open. Increasing supply of steam with loads greater than the economical is brought by feeding fresh steam directly to one or more intermediate stages of the turbine. Turbines with nozzle governing are also provided with internal bypass governing. In this method, further loading of the turbine is performed by opening simultaneously the bypass valve and the governing (control) valve which controls the steam flow rate through the additional nozzle segment of the governing stage. Since the pressure in the overload chamber increases with opening valve of internal bypass and the pressure in the governing stage remain unchanged, the flow rate of steam through the first group of turbine stages will decrease.

To change the output power of steam turbine there are two methods of control [2]:

- Constant live steam pressure.
- Sliding live steam pressure.

Constant live steam pressure in this type, the power is controlled by the steam-admitting elements of the turbine (valve control), while the boiler and pipelines are continuously under the rated steam pressure. Since the steam flow rate through turbine changes as a result of valve position (cross-sectional area of valve), the type of steam distribution plays a great role in defending the thermodynamic states through turbine.

Sliding live steam pressure in this method, the power is controlled by the boiler pressure when the steam-admitting elements of the turbine are fully opened and the load varies roughly proportional to the pressure of live steam. Long operation at reduced pressure increases the reliability and durability of heating surfaces of the boiler and of the steam pipelines connecting it with the turbine. In addition, since the steam pressure before the turbine varies smoothly (slides) and the steam temperature is maintained constant (rated temperature), the temperature of the majority of critical elements of the turbine at fully opened governing valves remains constant. Owing to this, load changes do not cause the appearance of non-uniform temperature fields in cross sections of turbine. Further load changes do not involve thermal expansion of rotor; bending stresses in blades of first stage are reduced. Furthermore, decreasing live steam pressure leads to decrease feed water pump work [3].

George D. et al. [4] investigated the main plant parameters for constant and sliding live steam pressure modes in the case of an existing Romanian condensing power plant. The results showed that the live steam sliding pressure operation mode introduces two opposite effects on a conventional steam unit thermal efficiency. On one hand, the thermal efficiency is decreasing due to the reduction of live steam pressure and on the other hand, the thermal efficiency is increasing due to the better behavior of the steam turbine control stage. Moreover, sliding pressure control causes increasing operation flexibility at partial loads. Jonshagen K. et al. [5] in their work studied a hybrid control strategy for (CHP) plant which is a combination of constant and sliding pressure control. The results depicted that the hybrid control increases the power to heat ratio (α) in a significant part of load range and will reduce the cycle stresses from the admitted and non-admitted sectors. The present work attempts to theoretically study the operation of two units of condensing type-K from Iraqi power plants under constant and sliding live steam pressure controls with different type of steam distribution. Furthermore, two units of cogeneration plant equipped with a back-pressure turbine type-R were chosen for this study. Based on the mathematical model simulation of condensing and cogeneration plants was developed. Researchers [6],[7] demonstrated that the exergy analysis is a useful tool in performance assessments of cogeneration and condensing types and it is a useful method, to complement not to replace energy analysis. So, all the above cases study were analyzed according to 1^{st} and 2^{nd} thermodynamics laws under different operation regimes, control methods and steam distribution modes.

Description of Cases Studied

Figures (1, 2) show the flow diagram for condensing unit type (K-66-87-0.07) and (K-55-58.8-0.083)of South Baghdad respectively. Tables 1&2 show the design operation conditions of these units. Ref. [8] shows more details of these units. The flow diagram of cogeneration plants with back-pressure turbine type R-40-130-29 and R-100-130-15 are shown in Figures (3, 4) respectively. Tables (3&4) demonstrate design operation regimes of these units. For more details about these units, ref. [9] is recommended.

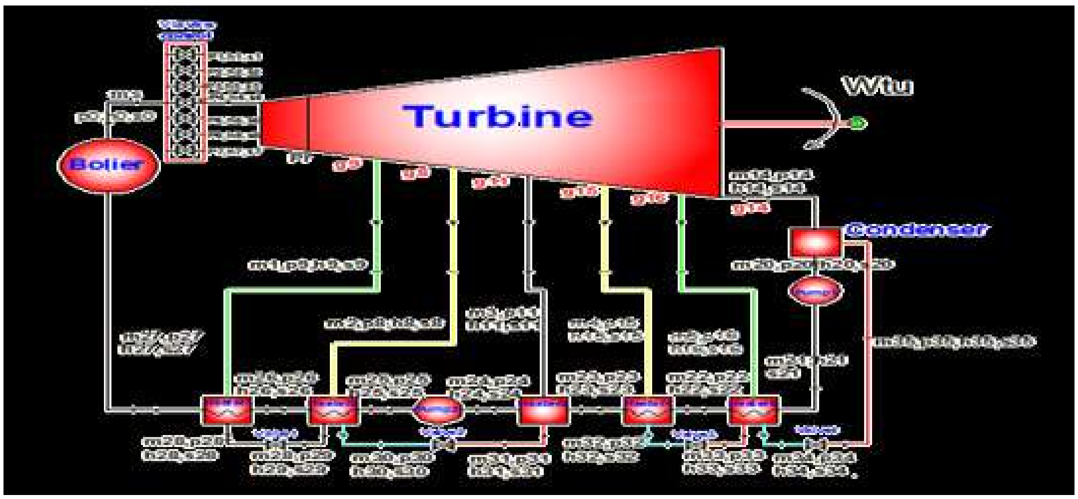


Figure (1): Flow diagram of South Baghdad unit with turbine type (K-66-87-0.07)

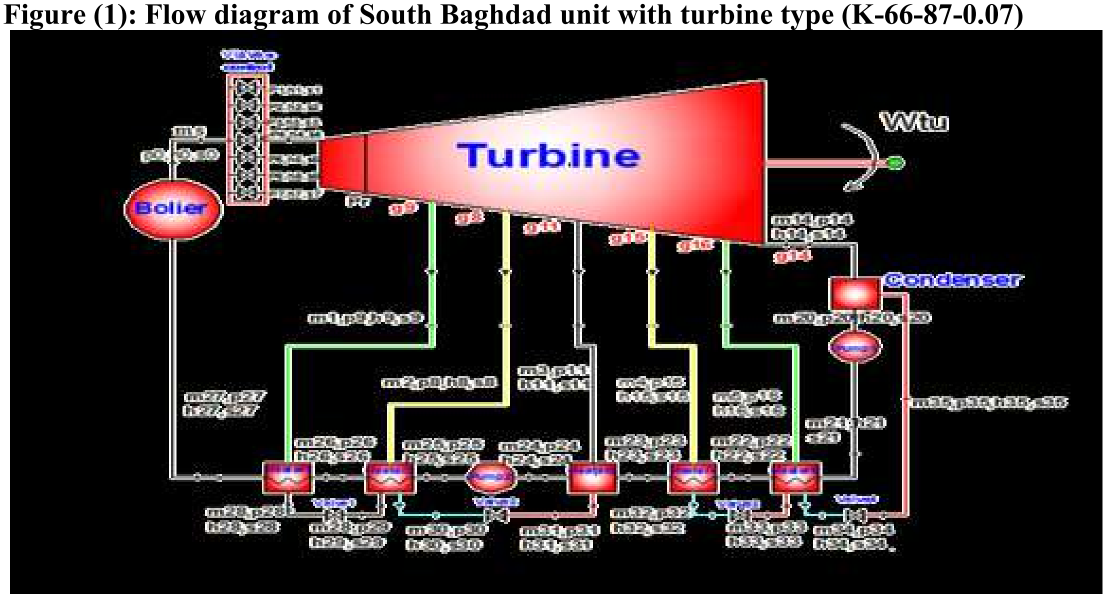


Figure (2): Flow diagram of South Baghdad unit with turbine type (K-55-58.8-0.083)

Table (1): Operation design condition for south Baghdad unit with condensing turbine type (K-66-87-0.07)

Position	pressure (bar)	specific enthalpy kJ/kg	specific entropy kJ/kg.K	mass flow rate kg/s
Inlet turbine (0)	87	3385	6.671	73.22
Control stage (r)	60	3319	6.743	10.46
The first bleed (9)	32.9	3181	6.806	5.42
The second bleed (8)	18.6	3056	6.861	3.99
The third bleed (11)	7.2	2874	6.946	3.38
The fourth bleed (15)	2.7	2716	7.029	4
The fifth bleed (16)	0.79	2547	7.133	4.85
Exit the turbine (14)	0.07	2289	7.369	51.58

Table (2): Operation design condition for south Baghdad unit with condensing turbine type (K-55-58.8-0.083).

Position	pressure (bar)	Specific enthalpy kJ/kg	specific entropy kJ/kg.K	mass flow rate (kg/s)
Inlet turbine (0)	58.8	3388	6.846	62.9125
Control stage (r)	38.47	3308	6.925	8.9875
The first bled (9)	27.054	3224	6.959	3.794
The second bled (8)	14.46	3084	7.017	3.25
The third bled (11)	6.66	2928	7.085	2.344
The fourth bled (15)	3.108	2793	7.146	3.75
The fifth bled (16)	0.866	2605	7.246	4.15
turbine exit (14)	0.083	2341	7.464	45.6245

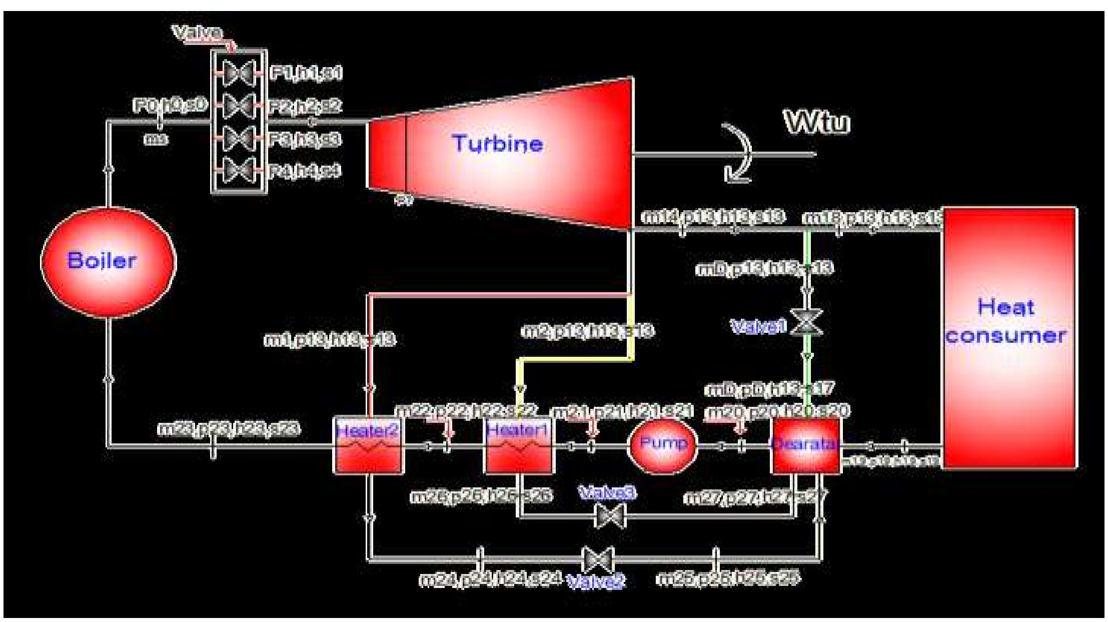


Figure (3): flow diagram of cogeneration plant with back pressure turbine type (R-40-130-29)

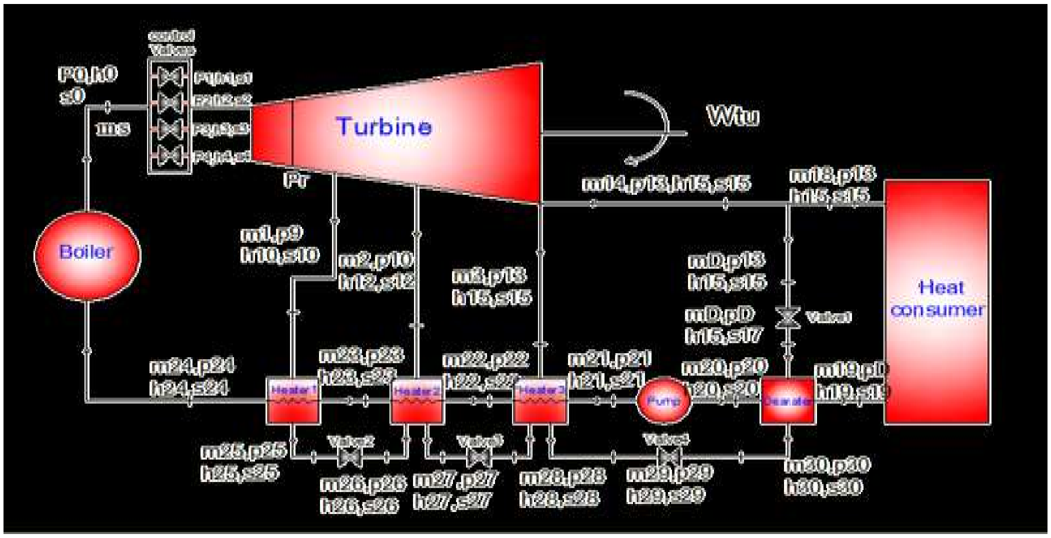


Figure (4): flow diagram of cogeneration plant with back pressure turbine type (R-100-130-15)

Table (3): Operation design condition for back-pressure turbine type (R-40-130-29)

Position	pressure (bar)	Specific enthalpy kJ/kg	Specific entropy kJ/kg.K	mass flow rate kg/s
Inlet turbine (0)	130	3509	6.654	130.556
Control stage (r)	72.2	3394	6.764	32.639
The first bled (13)	29	3163	6.833	8.407
The second bled (13)	29	3163	6.833	9.621
Turbine exit (15)	29	3163	6.833	112.528

Table (4): Operation design conditions for back-pressure turbine type (R-100-130-15)

Position	pressure (bar)	specific enthalpy kJ/kg	specific entropy kJ/kg.K	mass flow rate kg/s
Inlet turbine (0)	130	3509	6.654	211
Control stage (r)	94.4	3457	6.728	52.75
The first bled (9)	34.5	3193	6.805	9.444
The second bled (10)	23.1	3100	6.834	8.611
third bled (13)	15	3008	6.866	13.56
Turbine exit (15)	15	3008	6.866	179.385

Mathematical model and thermodynamics parameters

Mathematical model

Simulation model of condensing and cogeneration power plants 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 [9]. This model able to predict transient response, even for large process variations. The sliding and constant live steam pressure parameter operation is analyzed with a mathematical model which allows steam thermal cycle computation. The main components of the model are turbine process modeling, pumping and heating system of feed water system [4]. Steam turbine control stage feed is assured through control valves, unaffected by the operating mode (with or without sliding pressure); at design load these valves are completely opened. In the case of constant live steam pressure operating mode, at partial load, the control valves are sequentially closed. While, for sliding pressure mode all control valves remain fully opened through partial load [4]. The operation mechanism of control valves depends on type of steam distribution for constant live steam pressure mode. For accurate modeling it is usual to spit a turbine into number of cascaded sections. A section being in turn composed of number of cascaded stages [9]. The turbine stages belong to two categories.

- Control stage
- Pressure stage

Actual enthalpy drop in the control stage for any mass flow rate (regime), method of steam distribution and modes of control can be found by using Eq. (1) for turbines type K and R [9].

$$\Delta h = -316.98 \cdot \left(\frac{p_r}{p}\right)^{1.25} + 273 \quad \text{..........(1)}$$

where p_r , is the pressure of governing stage and p is the steam pressure behind any valve.
For pressure sWage the isentropic efficiency for condensing turbine is defined by Eq. (2) [9].

$$\eta_s = \left[0.915 - \frac{0.3}{m_1 \cdot v}\right] \cdot \left[1 + \frac{\Delta h_{is} + 1200}{25000}\right] - 0.03 \quad \text{........(2)}$$

And for back-pressure this efficiency is defined by Eq. (3) [9].

$$\eta_s = -0.185 \cdot \left(\frac{m_1}{m_s}\right)^{1.15} + 3.659 \cdot \left(\frac{m_1}{m_s}\right)^{0.05} - 2.634 \quad \text{.........(3)}$$

Where, m_1 is mass flow rate of steam through the group of stages, ν is specific volume at inlet of group of stages and Δh_{is} is isentropic enthalpy drop through the group.
So, actual enthalpy drop through pressure stage is defined by Eq. (4) [9].

$$\Delta h_{act} = \eta_s \cdot \Delta h_{is} \quad \text{..........(4)}$$

As a basis for steam turbines pressure calculus for design and off design flow distribution with the assumption that medium temperature variation is ignorable, pressure through turbine stages can be estimated from Stodla equation [9].

$$\frac{m}{m_s} = \sqrt{\frac{p_1^2 - p_2^2}{p_{10}^2 - p_{20}^2}} \quad \text{.........(5)}$$

Where, m, m_s represent steam flow rate at any regiem and at design condition respectively, p_1, p_{10} are steam pressures before any stages at specified regime and at design condition respectively, $p_2, p_{20}$are steam pressures after any stages at specified regime and at design condition respectively.

For feed water heaters, energy and mass balances are used to find extracted steam flow rate at each heater. Steam detention process in feed water line and steam boiler are similar in the two cases of control.

Based on the mathematical model, specialized computer program has been written to work under MathCad software. The program allows analyzing condensing and cogeneration plants under design and off design conditions with constant and sliding live steam pressure and different modes of steam distribution. So, according to 1st and 2nd laws of thermodynamics the following parameters are studied:

Thermodynamics parameters:

Irreversibility coefficient

According to 2nd law of thermodynamics, irreversibility (Ω) for each component of the plant is given by Eq. (6) [9, 10],

$$\Omega_i = \frac{\Phi_i}{m_f \cdot C.V} \quad \text{.........(6)}$$

Where, [10]

$\Phi = T_0 \cdot [\sum_{i=1}^{n}(m_i \cdot s_i)_{out} - \sum_{i=1}^{m}(m_i \cdot s_i)_{in}] + Q_0$ ………(7)

For adiabatic process $Q_0 = 0$

So, the overall irreversibility coefficient of plant is,

$\Omega_{total} = \frac{\sum_{i=1}^{n}\Phi_i}{m_f \cdot C.V}$ ………..(8)

Then the thermal efficiency of plant according to 2nd law of thermodynamics is,

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

Power developed

The power developed of steam turbine is give by Eq. (10),[9]

$P = H_o - \sum_0^j \alpha_j \cdot (H_o - H_{oj})$ ……….(10)

Where, H_o is the total heat drop through turbine, α_j is the relative extraction steam in the regenerative feed water heater and H_{oj} is the total heat drop from extraction (bleeding) point.

Power of pump

$P_p = \frac{m_p \cdot (\Delta p) \cdot v \cdot 1000}{\eta_p}$ ……….(11)

Where,

m_p is mass flow rate through the pump, Δp is pressure difference cross the pump, v is specific volume and η_p is pump efficiency.

Net power

$P_N = P - P_p$ ………(12)

The rate of heat added

$Q_{add} = m_s \cdot \frac{(h_0 - h_i)}{\eta_b}$ ……….(13)

Where,

m_s Is steam flow rate through boiler,h_0 , h_i are enthalpy of steam at boiler exit and boiler inlet respectively, η_b is boiler efficiency.

Cycle efficiency according to 1st law of thermodynamics

$\eta_I = \frac{P_N}{Q_{add}}$ …………(14)

Plant efficiency

$\eta_{plant} = \eta_I \cdot \eta_m \cdot \eta_g \cdot \eta_b$ ……….(15)

Where, η_m, η_g and η_b are mechanical, generator and boiler efficiencies respectivily.

Heat rate

To compare heat rate (kJ/kW.h) during constant and sliding live steam pressure control the following parameter is defined,

$K = 1 + \frac{\Delta q}{q_c}$ ………..(16)

Where,

q_c is heat rate at constant live steam pressure control= $\frac{rate\ of\ heat\ added}{net\ power}$ …….….(17)

$\Delta q = q_s - q_c$ ………….(18)

Where, q_s is heat rate at sliding live steam pressure control.

If K >1 constant pressure control is best than sliding pressure control and vice versa [11].

Rate of process heat

For cogeneration plant the rate of process heat (industrial heat demand) can calculated from Eq. (19).

$$Q_p = m_p \cdot (h_{exit} - h_{cond}) \qquad \text{……..(19)}$$

Where,

m_p is the process mass flow rate, and h_{exit} and h_{cond} are enthalpy of steam at exit of back-pressure turbine (inlet to heat process) and enthalpy of condensate from heat process respectively.

Power to heat ratio (α)

For cogeneration plant, power to heat ratio is defined as the ratio of net power developed to rate of heat demand (process heat),

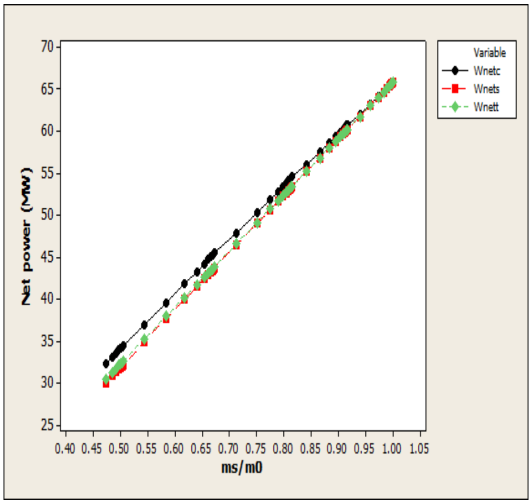
$$\alpha = \frac{P_N}{Q_p}. \qquad \text{................(20)}$$

Results and discussion

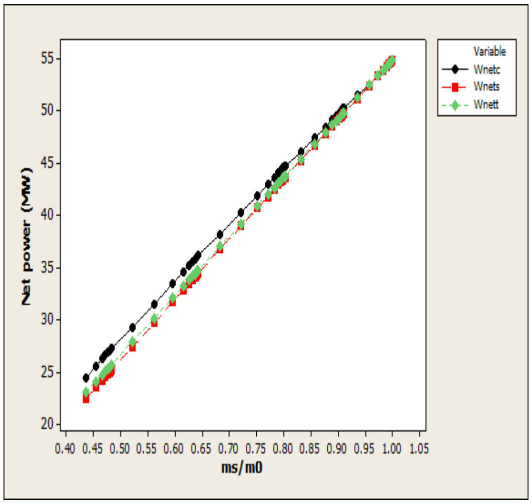
The characteristics of the condensing unit type (K-66-87-0.07) and (K-55-58.8-0.083) are as following:-

Net power developed

The net power is taking into account the effect of pump work. From figure (5), it is shown that the net power for constant pressure with nozzle distribution is greater than another type in spite of decreasing the pump work for sliding control. This makes this type of control lead to increase the turbine power developed in greater amount than the amount of decreasing pump work. The net power developed is the same for constant with throttling distribution control and sliding pressure.



a- type (K-66-87-0.07)

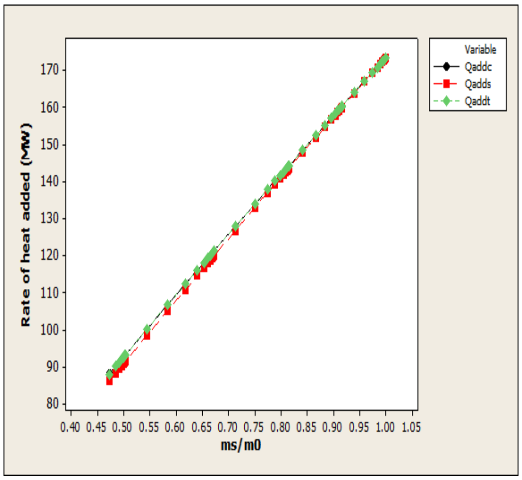


b- type (K-55-58.8-0.083)

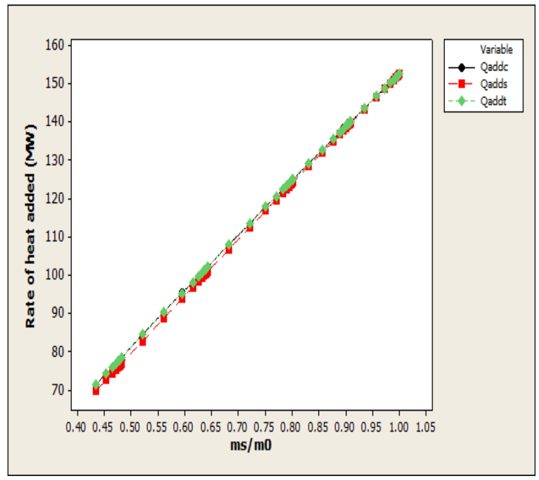
Figure (5): Variation of net power developed with flow rate ratio at various types of control for condensing power plant (c: constant pressure with nozzle, s: sliding pressure, t: constant pressure with throttling)

The rate of heat added

Figure (6) depict the change of heat rate during constant and sliding pressure controls. From this figures, it is shown that the rate of heat added for constant live steam pressure (nozzle and throttling distribution) is the same for specified flow rate due to the same enthalpy variation through steam boiler. For sliding pressure control, there is a small deviation at low flow rate from that for constant live steam control.



a- type (K-66-87-0.07)

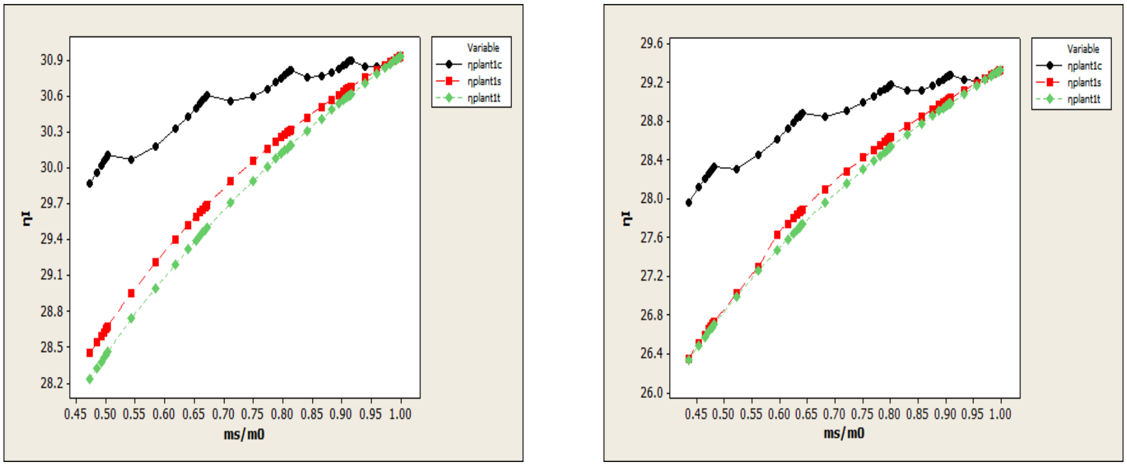


b- type (K-55-58.8-0.083)

Figure (6): Variation rate of heat added with flow rate ratio at various types of control for condensing power plant (c: constant pressure with nozzle, s: sliding pressure, t: constant pressure with throttling)

The plant efficiency according to first law of thermodynamics

From figure (7), it is shown that plant efficiency with nozzle distribution constant live steam control is higher than for throttling and sliding pressure. Although decreasing the initial pressure is leading to decrease cycle efficiency, but the enhancement turbine efficiency leads to the increase cycle plant efficiency above that for constant with throttling control. While, this enhancement in turbine efficiency cannot cover the decreasing of steam cycle compared with nozzle distribution, so the plant efficiency decreased [4]. Furthermore, the efficiency of type (K-66-87-0.07) is higher than that for (K-55-58.8-0.083) due to the enhancement of operation parameters.

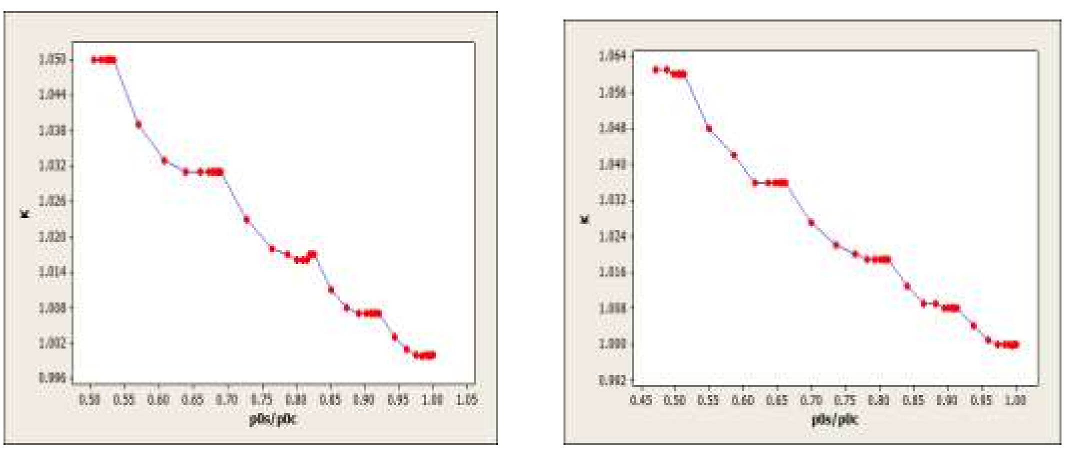


a- type (K-66-87-0.07) b- type (K-55-58.8-0.083)

Figure (7): Variation of plant efficiency in first law with relation to mass flow rate at various types of control for condensing power plant (c: constant pressure with nozzle, s: sliding pressure, t: constant pressure with throttling)

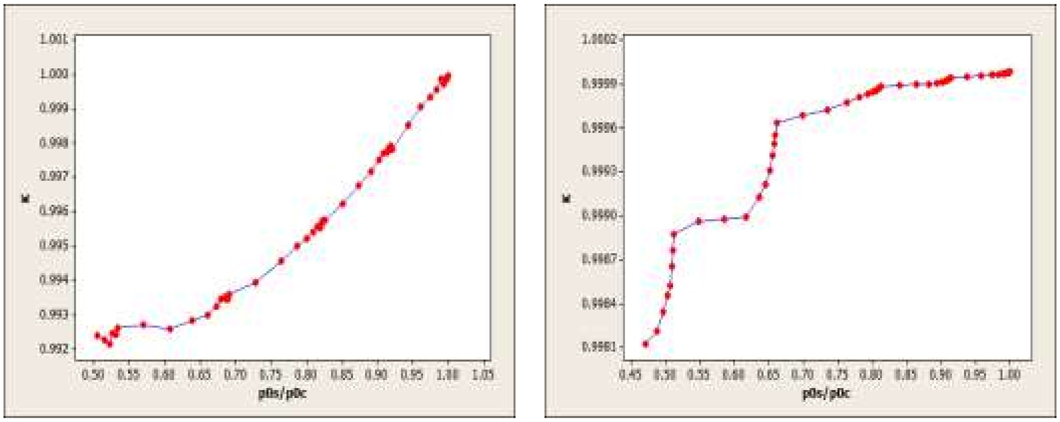
Heat rate

For all off design regimes of condensing unit, the parameter (K) is greater than unity for constant live steam pressure with nozzle distribution compared with sliding pressure. This means according to equation (16) that constant pressure control leads to decrease heat rate. From figure (8), it is shown that the increasing in heat rate is a bout (6-0%) depending on the operation regimes. While, using sliding type control leads to decrease heat rate compared with constant pressure with throttling, as shown in figure (9). This is about (1-0%) depending on operation regimes.



a- type (K-66-87-0.07) b- type (K-55-58.8-0.083)

Figure(8): Variation of heat rate with pressure ratio when constant pressure (nozzle distribution) is used for condensing power plant

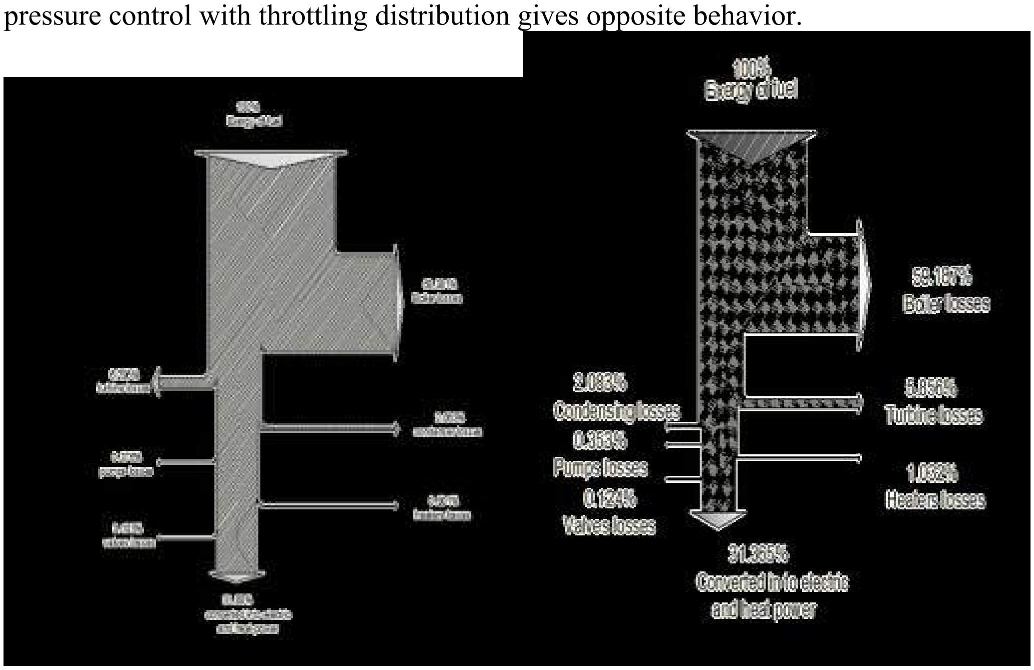


a- type (K-66-87-0.07) b- type (K-55-58.8-0.083)

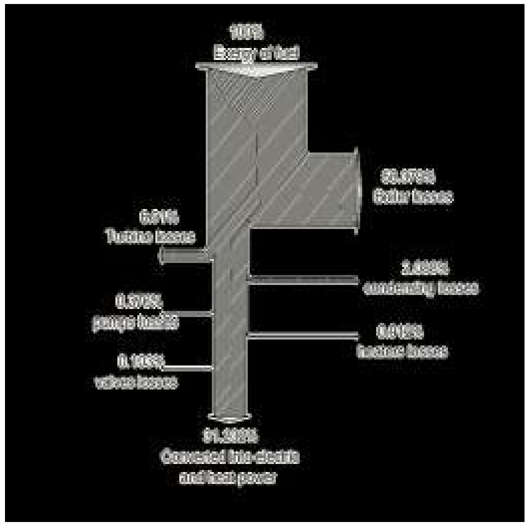
Figure(9): Variation of heat rate with pressure ratio when constant pressure (throttling distribution) is used for condensing power plant

Losses distribution in plant according to 2nd law of thermodynamics

Figure (10) illustrates the losses distribution according to 2nd law of thermodynamic for (K-66-87-0.07) for constant pressure with nozzle distribution, sliding pressure and constant with throttling distribution. It is shown that the major losses occurred in steam boiler and steam turbine. The selection of control type affects these losses, while the losses in other components are nearly the same regardless the type of control. From the figures, it is shown that decreasing of live steam pressure leads to increase boiler losses due to lower saturation temperature and then increases entropy generation. While, this type of control leads to decrease entropy generation through turbine, because throttling losses decreases. So, according to re-distribution the losses of these components, the constant pressure with nozzle have smaller losses and higher efficiency. While, constant pressure control with throttling distribution gives opposite behavior.



(a) (b)



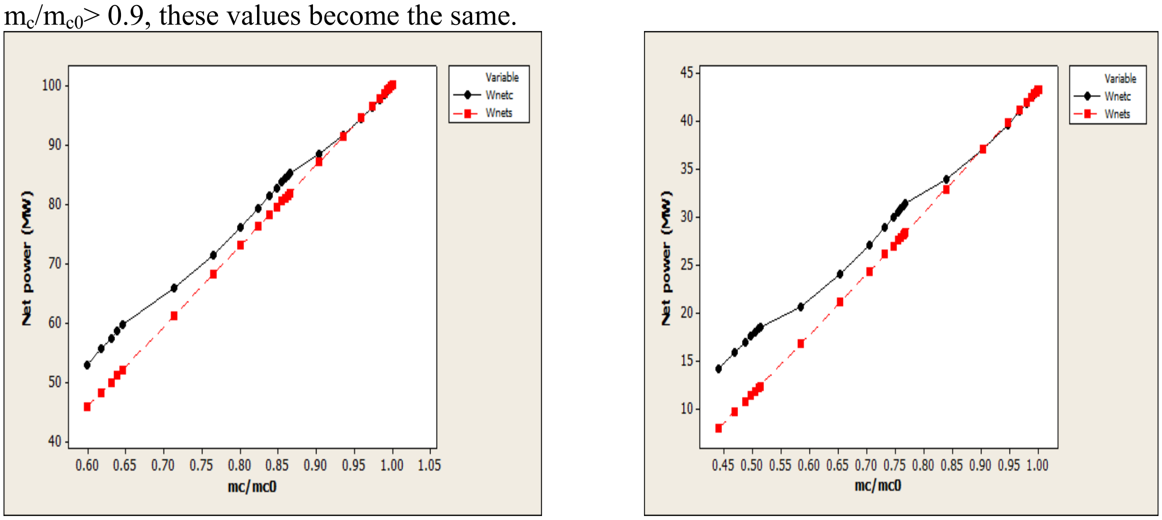
(c)

Figure (10): Losses distribution in condensing unit type (K-66-87-0.07) at various types of control, (a) constant live steam pressure with nozzle distribution (b) sliding live steam control (c) constant live steam control with throttling distribution.

In cogeneration Power Plants the throttling steam distribution is not advised for back- pressure turbine; because the throttling losses depend on the ratio of main steam pressure to back pressure. With a lower ratio, these losses are higher [1]. So, for cogeneration plant, only nozzle steam distribution was considered.

The net output power

Figure (11) reveal the variation of net output power developed for turbines (R-100-130-15) and (R-40-130-29), at constant and sliding pressure control. The net power developed at constant pressure control is still higher than sliding pressure control for regimes $m_c/m_{c0} \leq 0.9$ and for $m_c/m_{c0} > 0.9$, these values become the same.

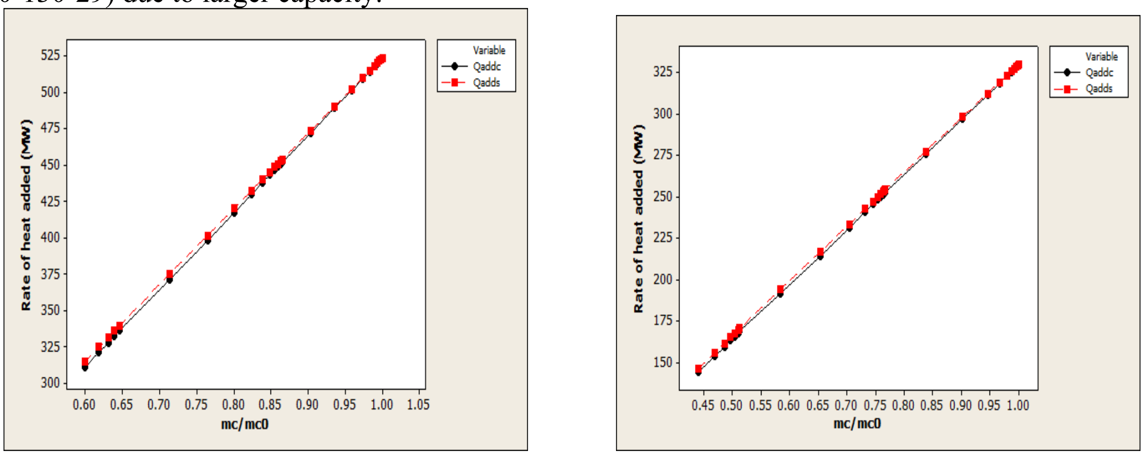


a- type (R-100-130-15) b- type (R-40-130-29)

Figure (11): Variation of net power developed with flow rate ratio at various types of control for cogeneration power plant (c: constant pressure with nozzle, s: sliding pressure)

The rate of heat added

The rate of heat added for both types of control is nearly the same as shown in figure (12) for turbine (R-100-130-15) and (R-40-130-29). From these figures, it is clear that the rate of heat added increases with increasing flow rate, and it is higher for turbine type (R-100-130-15) than turbine (R-40-130-29) due to larger capacity.

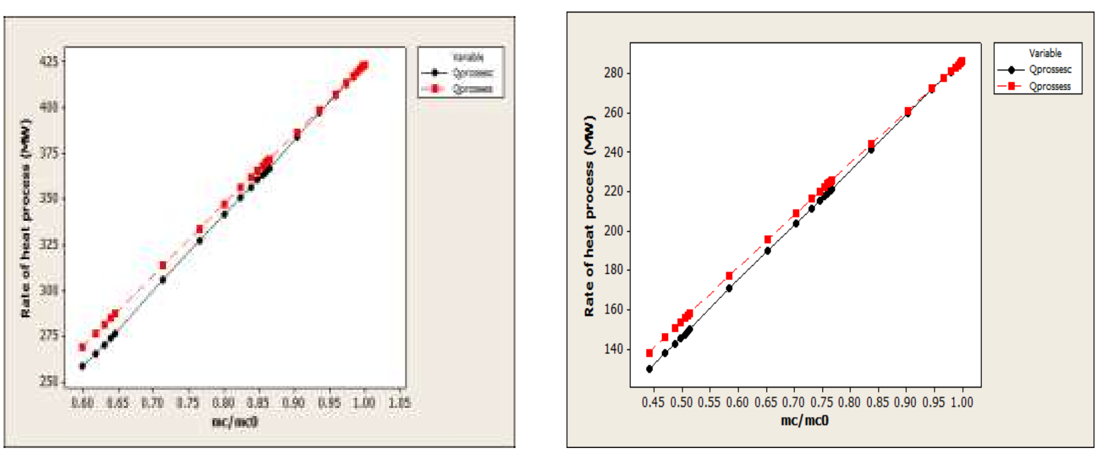


a- type (R-100-130-15) b- type (R-40-130-29)

Figure (12): Variation rate of heat added with flow rate ratio at various types of control for cogeneration power plant (c: constant pressure with nozzle, s: sliding pressure)

The rate of heat process

Variation of rate of heat process with steam flow rate directed to heat consumer is shown in figure (13) for turbines type (R-100-130-15) and (R-40-130-29). The heat process supplied with sliding pressure control is greater than constant control, especially at regimes $m_c/m_{c0} \leq 0.9$. This is attributed to increased steam enthalpy at turbine exit (inlet to heat consumer) at sliding pressure control at the same flow rate. So, to achieve the same heat process, the steam flow rate through turbine must be decreased at sliding pressure.

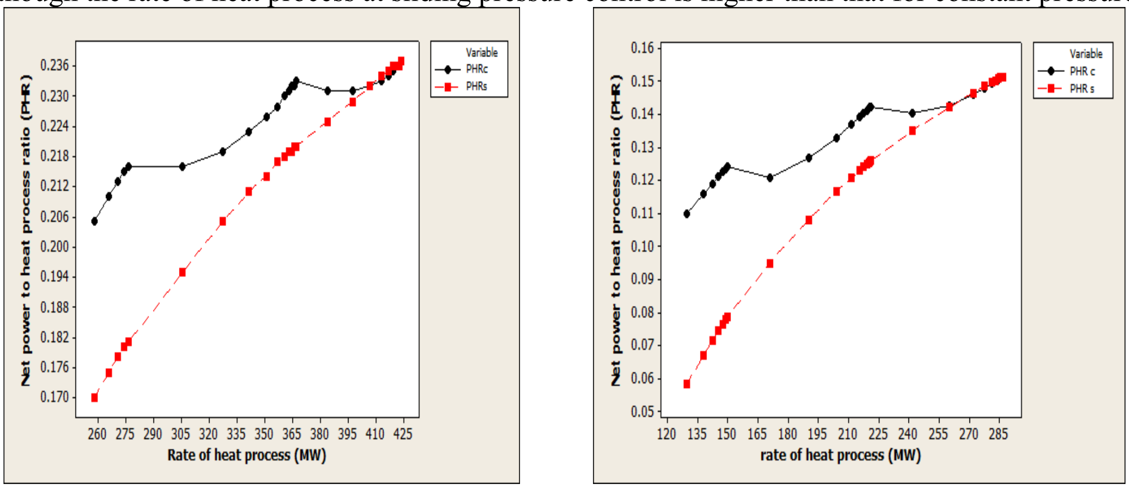


a- type (R-100-130-15) b- type (R-40-130-29)

Figure (13): Variation of rate of heat process with flow rate ratio directed to heat consumer at various types of control for cogeneration power plant (c: constant pressure with nozzle, s: sliding pressure)

Net power to rate of heat process ratio (PHR) (α)

Figure (14) show the variation of (PHR) with heat process in constant and sliding live steam pressure control for turbine type (R-100-130-15) and (R-40-130-29). PHR with constant live steam pressure control is greater than sliding pressure control. This is attributed to higher net power although the rate of heat process at sliding pressure control is higher than that for constant pressure.

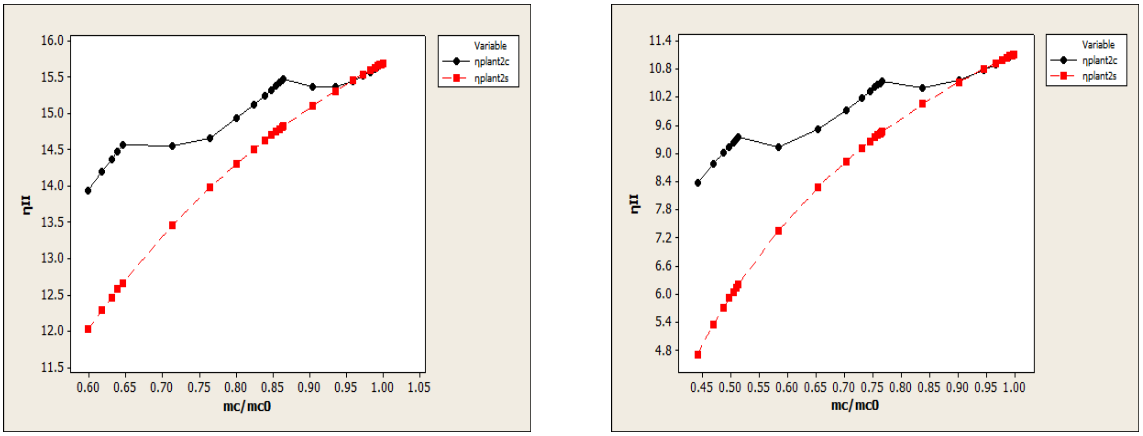


a- type (R-100-130-15) **b- type (R-40-130-29)**

Figure (14): Variation of net power to heat process ratio with rate of heat process at various types of control for cogeneration power plant (c: constant pressure with nozzle, s: sliding pressure)

The plant efficiency

Efficiency of cogeneration plant equipped with back-pressure turbine according to first law of thermodynamics is nearly 100% (when mechanical and generator losses are neglecting) regardless of control type and operation conditions. So, first law analysis cannot give any indication of performance of this plant. This is attributed to transform all heat added in steam boiler to work and heat process (absence steam condense), then efficiency becomes 100%. Hence, this type of plant must be analyzed according to 2nd law of thermodynamics which can sense the effect of variation control type and all operation conditions [1]. Figure (15) depict the variation of plant efficiency with steam flow rate for turbines (R-100-130-15) and (R-40-130-29). It is shown that sliding pressure control leads to decrease plant efficiency. The variation is about (0-2%) for (R-100-130-15) and (0-3.2%) for (R-40-130-29) depending on operation regimes.

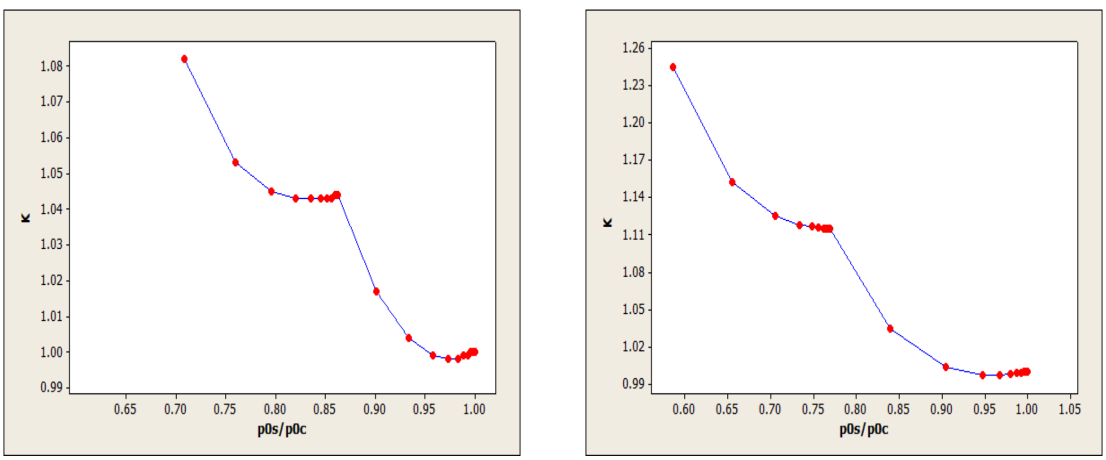


a- type (R-100-130-15) b- type (R-40-130-29)

Figure (15): Variation of plant efficiency with flow rate ratio directed to heat consumer at various types of control for cogeneration power plant (c: constant pressure with nozzle, s: sliding pressure)

Heat rate

The heat rate (kJ/kW· h) for sliding pressure control is larger than constant pressure control. So, the factor (K) is greater than unity for all operation regimes. Figure (16) reveal the variation of this fact with inlet pressure ratio for type (R-100-130-15) and (R-40-130-29). It is shown that the (K) value decreases with increasing inlet pressure ratio (increasing mass flow rate) and approaches to unity at pressure ratio ≥ 0.95.

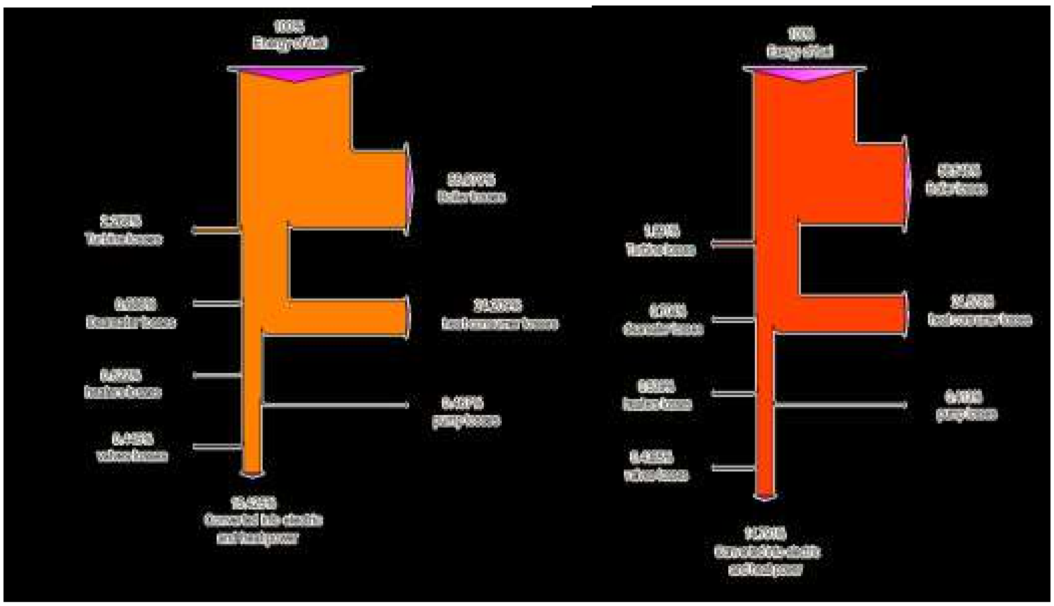


a- type (R-100-130-15) b- type (R-40-130-29)

Figure (16): Variation of specific heat consumption with pressure ratio for cogeneration power plant

Losses distribution in plant according to 2nd law of thermodynamics

Figure (17) shows losses distribution for (R-100-130-15) with sliding and constant live steam pressure control, respectively. Surely, changing the control type is associated with variation of properties (enthalpy and entropy) through the plant. So, using sliding pressure control leads to increase entropy generation in steam boiler according to increase irreversibility losses during heat transfer and combustion (increasing temperature difference between work substance and gases). At another position, such as steam turbine, these losses are lowering because absence of throttling losses (decreasing entropy generation). For heat consumer, these losses become larger than constant pressure control because steam at turbine exit has a larger enthalpy at sliding pressure. So, the temperature difference increases and leads to increase entropy generation. As a result of this redistribution of losses, the plant efficiency with sliding pressure control is less than constant control.



(a) (b)

Figure (17): Losses distribution in cogeneration unit type (R-100-130-15) at various types of control, (a) constant live steam pressure control (b) sliding live steam pressure control

Verification of mathematical model

In order to verify the mathematical model, the design data was used as the source of information for modeling. Validation versus design date is a basic step towards certification [9]. For the designing regime, the maximum power developed for condensing unit type (K-66-87-0.07), (K-55-58.8-0.083) and back-pressure unit type (R-100-130-15), (r-40-130-29) are 65.739 MW, 54.867 MW, 100.2 MW and 43.39 MW, respectively. These values are obtained from the simulation and have maximum deviation from design condition equal to 0.4%, 0.24%, 0.2% and 8.475%, respectively. So, the model gives a good agreement with the design conditions.

CONCLUSIONS

1. For condensing and cogeneration units, employing sliding live steam pressure control leads to enhance steam turbine efficiency and decrease pump work.
2. For condensing units, selection of control types depended on the type of flow distribution, where net power developed decreases upon sliding pressure control and nozzle steam distribution, while it is nearly the same as constant pressure control when steam distribution is throttling type.
3. With nozzle distribution, using sliding pressure control leads to decrease plant efficiency (increase specific heat consumption) according to first law of thermodynamics, while employing

sliding pressure control with throttling steam distribution leads to increase plant efficiency (decrease specific heat consumption) according to 1st law of thermodynamics.

4. Irreversibility losses distribution through the plant depends on the type of control. So, sliding control leads to increase these losses in steam boiler and decrease irreversibility losses in steam turbine.

5. For cogeneration plant with back-pressure turbine, employing sliding pressure control causes increasing process heat and decreasing power to heat process ratio.

6. Variation of initial conditions for condensing units and variation capacity of cogeneration units do not exhibit effect on the results behavior.

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