AN ANALYTICAL APPROACH FOR STEAM ENERGY CONSERVATION IN COMBINED HEAT AND POWER PLANTS

T.I. Sabry, N.H. Mahmoud and A.M. El-Ghonimy
Department of Mechanical Power Engineering Faculty of Engineering,
Menoufia University, Shebin El-Kom, EGYPT.

ABSTRACT

An analytical approach is given here for estimating thermodynamic performance of such a power plant of combined production of electric energy and heat. This approach is based upon the formulation of both energy and mass conservation equations for each constituent in a plant that combines all the components of any type steam power plant. Calculations have been carried out in terms of various thermodynamic performance parameters which are defined and discussed here. It is shown for such a combined heat and power (CHP) plant that the plant thermal efficiency is mainly affected with both process steam flow rate and condenser heat loss. Other parameters such as; boiler losses, auxiliary losses and unaccounted losses are found to slightly affect the plant efficiency. The predicted performance is compared to the actual one for three local industrial power plants in Egypt. A considerable agreement is observed through the comparison.

Keywords: ***Thermal power plants, Energy conservation, Combined heat and power plant, Industrial power plant, Thermodynamic performance, Efficient operation.***

Notation

Symbols

A_w	flow area of water	m^2
C.V.	calorific value	KJ/Kg/^{o}K
h	enthalpy	KJ/Kg
m	fraction of steam flow bled from turbine	Kg/Kg
M	mass flow rate	Kg/s
Q	heat rate	KJ/s
T	temperature	^{o}K
W	work	MW
Z	efficiency	----

Subscripts

1,2,3	referring to steam rates ahead turbine, to process and to condenser
a,b,c,d,e,f	referring to steam states across turbine stages
a	adiabatic
aux	auxiliary
b	boiler
cw	cooling water
e	out
E	electrically
EG	electrical generator
f	fuel
Fe	furnace exit
g	flue gases, stage
i	in, turbine stages number
is	isentropic
Int	internal
k	condensate
m	mechanical
n	net
p	process
S	steam
th	thermal
w	saturation corresponds to extracted steam pressure.

Abbreviations

AL	auxiliary loss
BL	boiler loss
CHP	combined heat and power
CL	condenser loss
DCSP	Delta company for sugar products
EEL	exhaust energy loss
MCWT	Miser company for weaving & textile industries.
NCMP	national company for maize products.
NP	plant net power
PE	process energy
UL	unaccounted loss

1. INTRODUCTION

The main purpose of power plants of combined production of heat and electricity is to generate electric energy and a sufficient rate of steam for heat exchange in an industrial process. These plants have been widely used abroad, particularly in the last decades, in several industries such as; paper making, textile, chemicals, sugar making, foods preserving, carpet making, etc. The efficient operation of such a plant of this type requires that planners and engineers have access to facilities that enable them to check different strategies of operation in order to obtain the best economical performance [1]. The first step in estimating this efficient operation is to develop a detailed thermodynamic analysis for the performance of all the plant elements (i.e., boiler, turbine, condenser and feedwater heaters, etc.). This analysis aims to obtain the following:

i- A prediction for the flue gases temperature distribution along the boiler passage besides the boiler efficiency.
ii- The predicted power and thermal efficiency of turbine stages.
iii- Mass flow rates of turbine extractions to feedwater heaters.
iv- Plant energy distribution and consequently plant efficiency.

The motivation for this work is the need for a methodology that incorporates the problems of energy conservation into a realistic simulation of the performance of an industrial (CHP) plant. The present study concerns with formulating and solving energy and mass balance equations for each element in such a (CHP) plant in order to deduce the performance parameters. For solving this problem analytically, it is necessary to construct suitably simplified model for the cycle. The approach illustrated here is suggested for a general power plant that contains all the components of any industrial power plant. This suggestion has been taken to facilitate the approach comparisons with measurements of field cases.

2. THERMODYNAMIC ANALYSIS:

Thermodynamic analysis presented below has dealt with illustrating energy balances over all the plant elements. Before starting the analysis; it should be noted that there are two main streams flowing through the plant elements namely combustion gases and water and/ or steam flows. The thermodynamic properties of these main two flows are computed here due to references [2,3].

2.1 *Boiler Analysis:*

Boiler efficiency and furnace exit temperature are utilized here to characterize boiler performance. Boiler efficiency is formulated here using the so-called "boiler gross efficiency" as:

$$\eta_b = 100 - \Sigma \text{ Heat losses} \qquad (1)$$

where the term (S Heat losses) involves the percent heat loss due to waste gases, incomplete combustion, unburned carbon, moisture in flue gases, moisture in air, blow- down losses and heat transfer to surroundings. The items of (S Heat losses) denoted above have been defined and computed analytically in a pervious work presented by the authors [4]. Whilst furnace exit temperature of flue gases is obtained using a semiempirical formula derived by Lin [5] as follows.

$$T_{gF_e} = \frac{T_a}{Z\left[\frac{5.67 x 10^{-11} \psi . A_w . e_f . T_a^3}{\phi . \dot{m}_f C}\right]^{0.6} + 1} \qquad (2)$$

where

T_a: is the adiabatic temperature of combustion, °K
A_w: is the flow area of water, m^2
M_f: is the rated fuel consumption of the boiler, Kg/s

The terms Z, Y, e_f, f and C are defined and estimated besides the unknown above terms (e.g., T_a) due to the method explained in [5].

2.2 *Feedwater Heaters Analysis:*

The mass and heat balance relations of a general feedwater element is expressed due to [3,6] as :

$$\sum \dot{M}_i = \sum \dot{M}_e \tag{3}$$

$$\sum \dot{M}_i h_i = \sum \dot{M}_e h_e \tag{4}$$

Applying the above relations upon the main types of feedwater heaters (i.e., surface and direct contact ones) releases the required governing equations of these heaters.

2.3 *Steam Turbine Analysis:*

Owing to the industry requirements of both superheated steam at high or moderate pressure and saturated or wet steam at a pressure giving the desired heat for process work, pass-out steam turbines are generally used in (CHP) plants. In the following, pass-out steam turbine will be analyzed thermodynamically.

2.3.1 *Pass-Out Turbine Analysis:*

Figure (1a) is a diagrammatic sketch for a pass-out steam turbine consists mainly of two stages, one of high pressure and the second of low pressure. The shape of turbine stateline arising under controlled pass-out conditions is illustrated in Figure (1b). In practice, this turbine operates under full pass-out ($M_3 = 0$), partial / or controlled pass-out ($M_3 = M_1 - M_2$) and zero pass-out ($M_3 = M_1$) conditions.

The present analysis aims to combine output power, rate of pass-out steam and total steam consumption of such a turbine. This analysis has been carried out under full, partial and no pass-out/or extraction conditions. This is because these conditions were found in equivalence to back pressure, bleeding and condensing turbines. The analysis presented below requires two correlations before starting, the first relates turbine load to steam consumption (i.e.; the so-called Willan's line) and the second estimates the pressure drop through the governing valves. These correlations were taken from [7,8].

Now, turbine analysis can begun using Figure (1a) in order to obtain the following combinations for turbine operation conditions which were discussed above.

i- Back-pressure operation, i.e. full pass-out condition:

This condition is characterized from energy conservation point of view by:

$$\dot{M}_3 = 0 \tag{5.a}$$

$$,W = \dot{M}_1 . (h_a - h_c) \tag{5.b}$$

$$,Q_p = \dot{M}_1 . (h_c - h_w) \tag{5.c}$$

Rearranging Equs. (5.b) and (5.c) gives

$$\dot{M}_1 \frac{W}{h_a - h_c} = \frac{Q_p}{h_c - h_w} \tag{5.d}$$

Rate of steam consumption that obtained from Equ. (5.d) is the largest one through the cases of turbine operation.

ii- Condensing turbine operation; i.e. zero pass- out condition:

This case is defined thermodynamically by:

$$\dot{M}_3 = \dot{M}_1 \tag{6.a}$$

$$,W = \dot{M}_1 . (h_a - h_c) \tag{6.b}$$

$$,Q_p = 0 \tag{6.c}$$

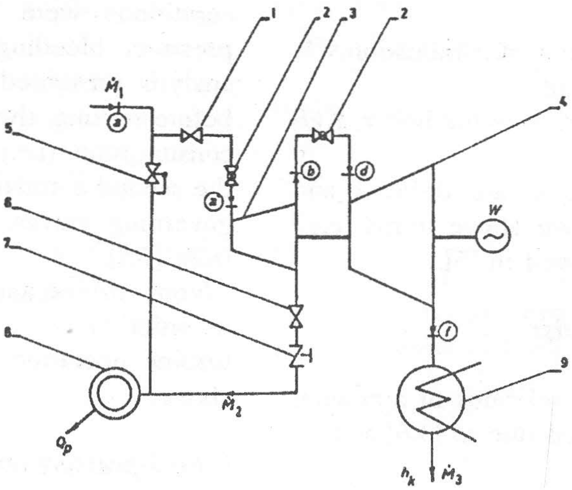

Figure 1a. Arrangement of single automatic extraction pass-out turbine.1- stop valve, 2- throttle valve, 3- high pressure stages, 4- low pressure stages, 5- regulating valve, 6- shutt-of valve, 7- Non-return valve, 8- heating process, 9- condenser.

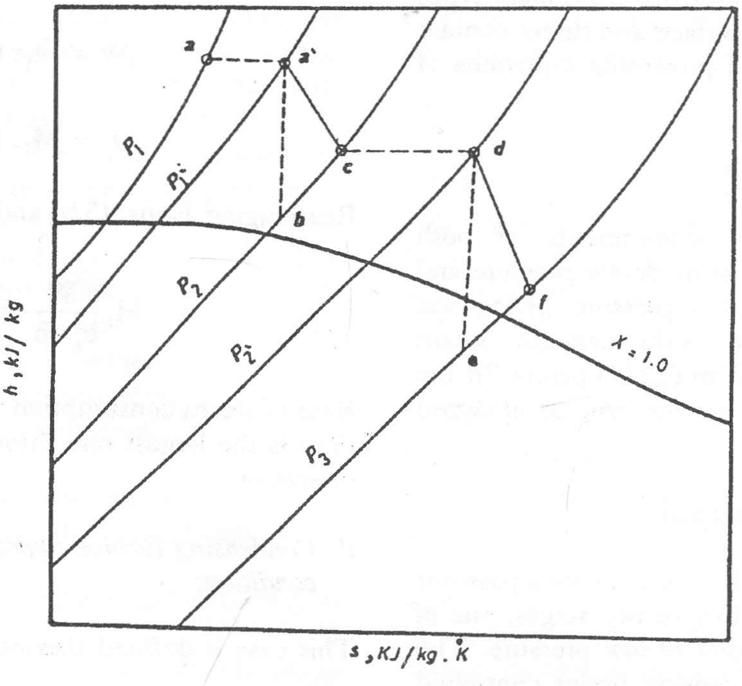

Figure 1b. Mollier diagram for pass-out turbine with throttle governing.

Now, turbine consumption of steam is obtained from Equ. (6.b) as :

$$\dot{M}_1 \frac{W}{h_a - h_f} \tag{6.d}$$

Equation (6.d) gives the minimum consumption of steam for any turbine load.

iii- Bleeding turbine operation, i.e. partial pass-out condition:

This mode of operation occurs when the steam required for process $\dot{M}_2 = \dot{M}_1 - \dot{M}_3$ is taken from the turbine as bled steam and the condensate from the process being returned to the cycle. This mode is expressed by :

$$\dot{M}_1 = \dot{M}_2 + \dot{M}_3$$

$$W = W_2 + W_3 \tag{7.b}$$

$$W_2 = \dot{M}_2.(h_a - h_c)$$

$$W_3 = \dot{M}_2.(h_a - h_f)$$

$$Q_p = \dot{M}_2.(h_c - h_w) \tag{7.c}$$

Substituting into Equ. (7.a) about M_2 2 and M_3 from Equ (7.b) to obtain.

$$\dot{M}_1 = \frac{1}{h_a - h_f}.\left[W + \dot{M}_2(h_c - h_f)\right] \tag{7.d}$$

2.3.2 *Turbine Energy Balance :*

Turbine energy balance aims to get the power produced from the engine and the plant thermal efficiency with the aid of total steam flow rate, rate of bled (or pass-out) steam and steam enthalpies at turbine main sections. Referring to Figure (2a) where a general schematic flow diagram of a turbine is shown; and Figure (2b) where the expansion line on the (h-s) diagram is given, the energy and flow rate balance relations is derived due to El- Wakil [6] and El-Mahallawy and others [9] as detailed below in th following subsections.

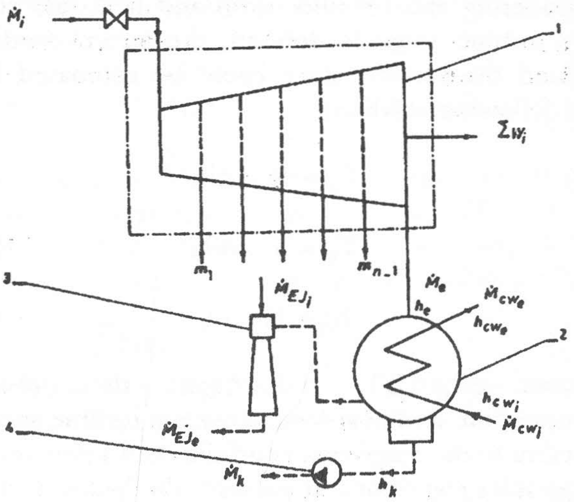

Figure 2a. Turbine-condenser arrangement.
1- Turbine 2- Condenser,
3- Ejector, 4- Condensate pump.

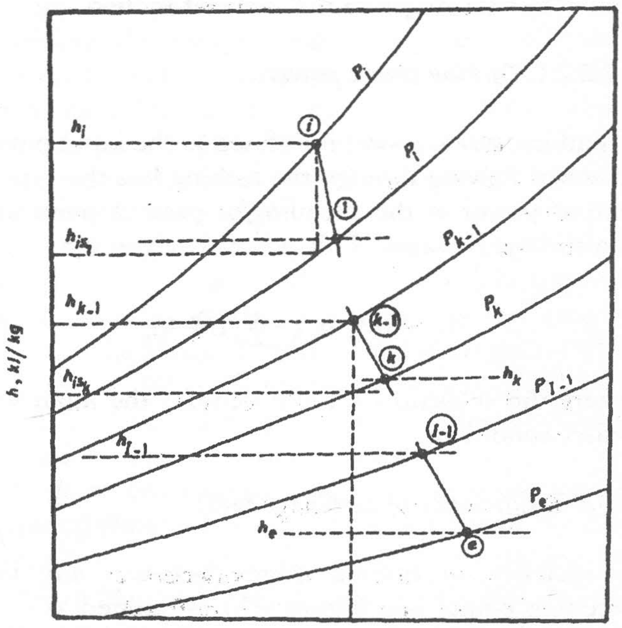

Figure 2b. Expansion line of the turbine shown in Figure 2a.

Before starting this balance the determination of turbine expansion or state line on (h-s) diagram may become appreciably more involved. The first step in determining this line is to evaluate the steam conditions at the exit of any turbine stage. Now, considering the pressure drop and heat loss across any turbine stage is defined, the steam condition behind the turbine stage could be estimated from the following relations:

$$P_e = P_i - \Delta P_l \tag{8.a}$$

$$T_e = Ti - \Delta T_l \tag{8.b}$$

$$h_e = h_i - \Delta q_l \tag{8.c}$$

where: ΔP_l; ΔT_l and $\Delta q_l$are the pressure; temperature and heat loss across any turbine section, e refers to the stage exit condition and i denotes the stage inlet condition. At present, the losses in steam pressure; temperature and enthalpy are evaluated depending on past experience and experimental findings. Finally, the turbine state line is characterized as the locus of the steam conditions gained using relations (8.a, b&c) from turbine inlet and across its stages to turbine exit section.

2.3.2.1 *Turbine steam power:*

Turbine steam power is defined as the input power in steam flowing through the turbine less the sum of output power at the bleeding/or passout point and the turbine exhaust. This power is given by:

$$W_s = (\dot{M}_s . h_s)_i - \sum (\dot{M} . h)_e \tag{9}$$

where the subscripts i and e denotes the input and outlet conditions.

2.3.2.2 *Efficiency of turbine stages:*

Efficiency of turbine stages between any two bleeding points, see Figure (2b), is defined as :

$$\eta_g = \frac{\Delta h}{\Delta h_{is}} \tag{10}$$

Where Δh and Δh_{is} are the actual and isentropic enthalpy drops of steam flowing through these groups. It is important to make clear that the kinetic energy of steam leaving turbine last stage is actually a power loss and known as the exhaust energy loss (EEL). This loss (EEL) is given usually by the manufacturers [10] as a function on the volumetric steam flow rate through turbine last stage. Therefore, efficiency of turbine last stages is expressed as:

$$\eta_g = \frac{\Delta h}{\Delta h_{is} + EEL} \tag{11}$$

2.3.2.3 *Turbine electric power output:*

The total internal power of the turbine is the sum of turbine cylinders output as :

$$W_{Int} = \sum_{i=1}^{i=n} W_i \tag{12}$$

In addition, turbine total electrical power output is defined as the turbine total internal power less mechanical and electrical losses in the electrical generator and can write as :

$$W_e = W_{Int} \cdot \eta_m \cdot \eta_{EG} \tag{13}$$

2.3.2.4. *Turbine gross thermal efficiency:*

This efficiency is the ratio between turbine total electrical power and fuel energy input rate to the plant (based on the heating value and fuel flow rate).

$$\eta_{th.g} = \frac{W_e}{Q_i} = \frac{W_e}{\dot{M}_f . (C.V)} \tag{14}$$

Commonly, the above system of equations (5-14) is sufficient for determining turbine output and its thermal efficiency knowing steam flow rate and steam enthalpies at all inlet and exit points of turbine sections.

2.4 *Steam Condenser Analysis:*

It is well known that the energy balance relations for surface condensers; that are widely used in thermal power plants, are similar to those of any closed type heat exchanger. Neglecting energy loss and the effect of air ejection from the condenser, the heat balance relation due to the schematic diagram in Figure (2a) is simply illustrated by.

$$\dot{M}_e . h_e + M_{cw_i} . h_{cw_i} = M_k . h_k + \dot{M}_{cw_o} . h_{cw_o} \quad (15)$$

Substituting $\dot{M}_e$ by $\dot{M}_k$ for steady state conditions can modify Equ. (15) to become:

$$\dot{M}_e . (h_e - h_k) = \dot{M}_{cw} . (h_{cw_o} - h_{cw_i})$$

$$= Q_{condenser} \quad (16)$$

2.5 *Plant Overall Performance:*

In order to characterize the overall performance of a thermal power plant; it is necessary to estimate the net power, net thermal efficiency, and heat losses of that plant under different mods of operation. Now, the net power and other performance parameters of the plant are calculated using definitions presented by El-Wakil [6] and El-Mahallawy and others [9] as: The plant net power is the generated power less the power consumed by the equipments that electrically driven and that using steam in driving.

$$W_n = W_e - W_{aux,E} - W_{aux,s} \quad (17\text{-a})$$

where

$$W_{aux,s} = \frac{W_e . \dot{M}_{aux,s}}{\dot{M}_i} \quad (17.b)$$

where $\dot{M}_i$ and $\dot{M}_{aux,s}$ are the steam flow rate at turbine inlet and rate of steam consumed in driving some of plant equipments respectively.

The plant net thermal efficiency is therefore defined as the ratio between plant net power and fuel energy input rate.

$$\eta_p = \frac{W_n}{M_f . C.V} \quad (18)$$

There are many types of the thermal losses encountered when analyzing the thermal power plants performance. The attention is directed in the present work to define the major ones of these losses such as condenser, boiler, auxiliary and unaccounted losses. Relations concerning with computation of these losses were presented in many text books [3,6]

3. COMPUTATIONAL TECHNIQUE:

In order to build a computer program used to give the performance characteristics of such a CHP plant, the system of equations (1-18) are solved. To perform the required simulation, the program utilizes three sets of data contains:

i- Initial conditions and all the plant data includes specifications and operation characteristics of all the plant elements.
ii- Digitized sets for manufacturer's performance characteristics of the plant constituents.
iii- Suitable relations for the thermal properties of the plant main flows.

First of all, the accuracy and behaviour of program output are checked. This can be achieved by assessing the computation procedure results using reported measurements for an actual CHP plant as input data. After all, the predictions of the computer program have been compared to creditable values for these results such that presented as design values by the manufacturers [10].

Considering now, first, the power plant of National company for Maize products (NCMP) in Ramadan Tenth city as the guide to check the program results. The elements of (NCMP) plant are shown in Figure (3). All the required information about this plant such as elements, specifications, boundary conditions, etc. are listed in Table (1).

Program calculations can start by obtaining the required thermodynamic properties for all the plant fluids using program subroutines. This starting step is followed with using the input data and digitized performance curves in the next steps of program computation procedure to obtain the required performance parameters for all the plant elements and for the plant as a whole.

4. FIELD MEASUREMENTS

Field measurements should be carried out in the present research parallel to the computational program in order to justify the computational obtained results. The suggested field measurements are directed to obtain the performance parameters of such existing local industrial CHP plants. Therefore, three local industrial CHP plants installed and operated at National company for Maize products (NCMP) in Ramadan Tenth city, Misr company for Weaving & Textile industries (MCWT) in El-Mehalla El- Kobra and Delta company for sugar products (DCSP) in Kafr El- Shiekh were chosen to carry out the field measurements. Plant characteristic and performance parameters which obtained from measurements in these three field cases are listed in Table(1).The flow diagrams, consequently the plant elements, of (MCWT) and (DCSP) plants are presented in reference [11].

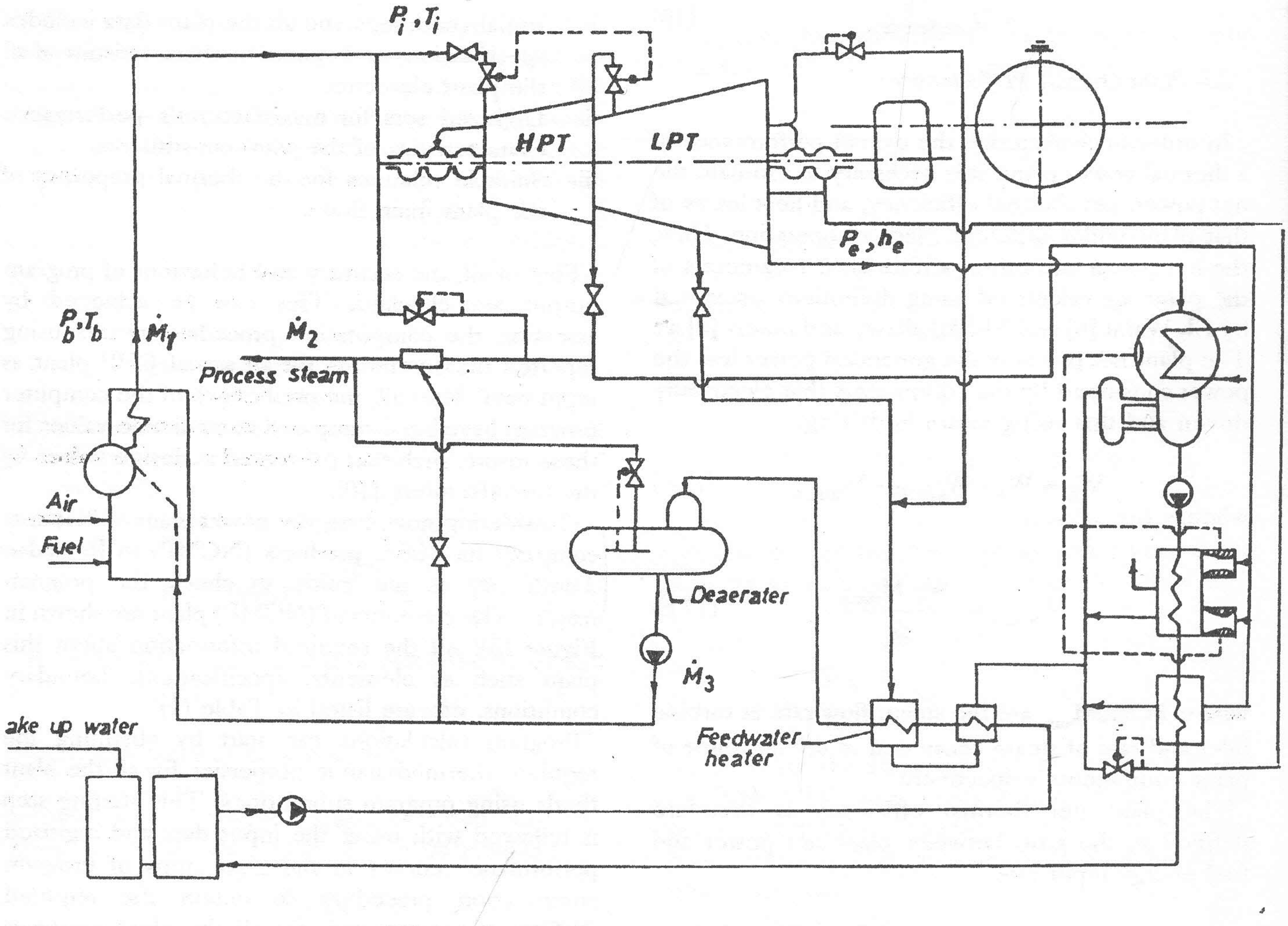

Figure 3. Flow diagram of NCMP plant.

Table 1. Plant characteristics and performance parameters derived from measurements of field cases.

Parameters	**Plant**		
	NCMP	**MCWI**	**DCSP**
Number of boilers	2	2	2
Boilers type	water-tube	water-tube	water-tube
Boiler capacity, (Ton/hr)	35	150	70
Blow down rate, (Ton/hr)	0.5	-	1.5
Boiler pressure (at drum), (bar)	44	90	-
Number of turbine sets	1	2	2
Turbine type	pass-out	bleeding	pass-out
Turbine electric energy rate at full load, (MW)	6.5	22	5.5
Steam pressure at turbine inlet, (bar)	40	85	32
Stem temperature at turbine inlet, (^{O}K)	673	788	633
Pressure of turbine exhaust, (bar)	0.11	0.1	2.4
Number of feedwater heaters	1+deaerator	2+deaerator	deaerator only
Steam pressure to heaters, (bar) : Heater No.1	12	9.4	-
: Heater No.2	-	6.6	-
: Deaerator	0.56	2.7	2.4
Feedwater pressure at boiler inlet, (bar)	47	125	50
Feedwater temperature at boiler inlet, (^{O}K)	378	445	378

Attention is devoted here to describe simplified field measurements that yield the required data for the suggested computational procedure. These measurements fall into two categories; those needed to compute the plant performance parameters, and those necessary to compare computed parameters from measurements to design ones.

Most of the measurements which carried out in any power plant are considered mainly as pressure, temperature and flow rate measurements. Pressures were measured in the present work by means of Bourdon-tube gauges (accuracy = ± 1% of full-scale reading) and pressure transducer systems (accuracy = ± 0.25 % of set span). Temperature were measured here using filled-system thermometers (accuracy = ± 0.2% of full-scale reading) and thermocouple systems (accuracy = ± 3 °C). Whilst flow rates were measured generally with the aid of calibrated orifice plates (accuracy = ± 0.3% of full-scale reading).

5. DISCUSSION OF RESULTS:

The results to be discussed below use the characteristics and performance parameters of NCMP plant at full load of 6.5 MW and partial loads of 3.86 MW and 4.7 MW as input data for the computer program. In order to evaluate the performance of such CHP plant, a comparison between measurements and predictions of the computer program must be carried out. Here, these predictions are obtained using the design values of operation conditions [10] as input data to the computer program.

Firstly, boiler overall efficiency (η_B) and boiler furnace exit temperature (T_{gfe}) are found dependent upon the boiler load. This is depicted in Figure (4) where predictions upon design data are compared to measured values. It is apparent from the figure that both h_B and T_{gfe} increase with increasing the boiler load. The increasing of T_{gfe} is due to increasing fuel firing rate while the boiler load raising until its rated capacity is the cause of efficiency improvement [12]. Furthermore, it is obviously that both the measured values of h_B and T_{gfe} are less than the computed ones upon design values. Thus, it is clear that the measured values of h_B at different loads are less than the computed ones by 4% in the average. The reasons for this difference are mainly due to the high dry and wet gases loss, incomplete combustion and some leakage of steam from steam traps and safety valves. The difference between the measured values of η_B and the computed ones in Figure (4) is mainly due to incomplete mixing of fuel and air as a result of burners malfunctioning.

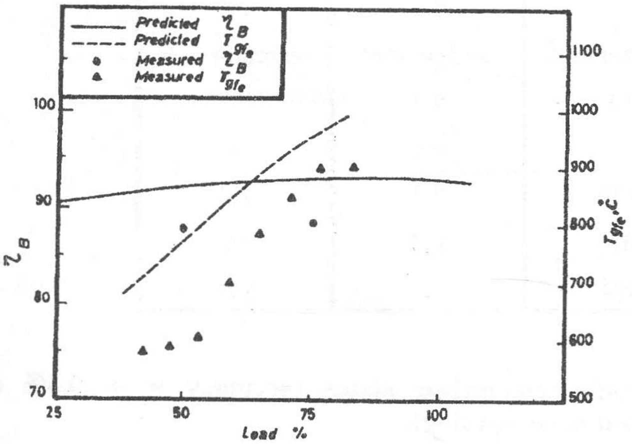

Figure 4. Variation of η_B and T_{gf_e} with boiler load in NCMP plant.

Figures (5) through (7) present comparisons of predicted; upon design values; rates of live steam and turbine extractions, plant and turbine efficiencies and elements of plants energy distribution with measured ones for NCMP plant. These diagrams (5:7) give variations of measured or predicted parameters in the range of plant load variation.

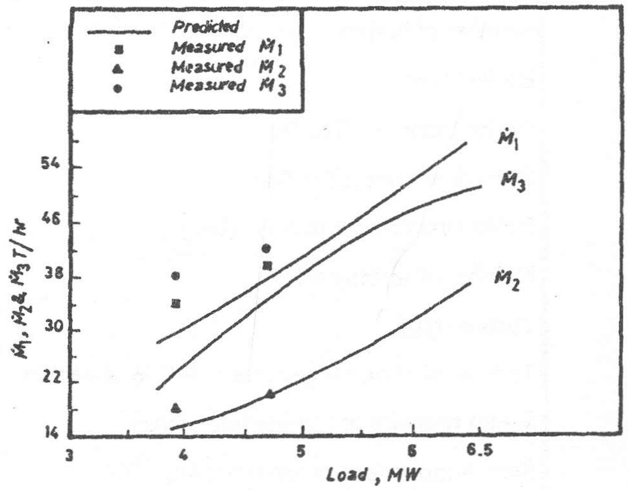

Figure 5. Variation of live steam rates and turbine extraction rates with load for NCMP plant.

Figure (5) shows the live steam and extracted steam flow rates $\dot{M}_1, \dot{M}_2 \& \dot{M}_3$ at the investigated plant loads. It is known from Figure (3) that the turbine first bleeding ($\dot{M}_2$) is extracted for process work while the second one($\dot{M}_2$) is ejected for feedwater heating. Therefore, increasing plant load tends to increase both $\dot{M}_1, \dot{M}_2$ and $\dot{M}_3$. It is also noticed in this figure that the predicted values of live steam and pass-out (or process) steam flow rates are close to the measured ones. This is because both the live steam consumption and the pass-out steam flow rates are related to the factory actual demand. On the other hand, the measured second bleeding rates ($\dot{M}_3$) are generally higher than the predicted ones. This is attributed to discharging the steam trap into the heater steam line {see Figure (3)}.

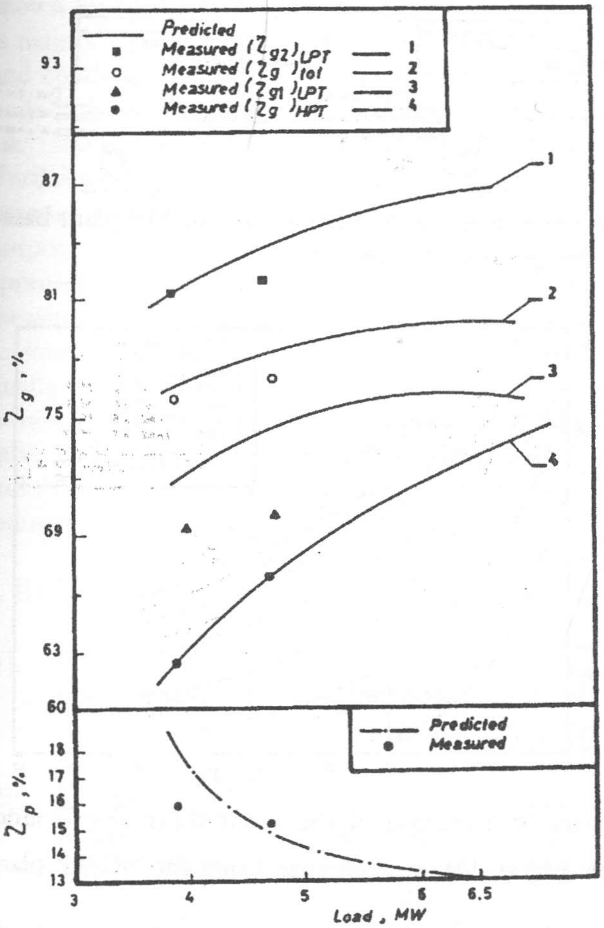

Figure 6. Variation of the plant and turbine efficiencies with the load for NCMP plant.

We now proceed to compare the computations of both turbine efficiencies and plant efficiency for NCMP plant with measurements. This comparison is given in Figure (6). The first look on this figure reveals that the isentropic efficiencies of turbine stages increase with the load increasing. This is because the percentage of mechanical and generator losses are decreased when turbine load is increased. The reduced values of the high pressure stages efficiency comparing to the values of low pressure stages is also noticed in this plot. This is mainly attributed to the increased labyrinth glands clearance and non-correct measured pressure values at the exit of high pressure stage. Further, the comparison in Figure (6) indicates a reduced agreement between the measured isentropic efficiencies of the first group of low pressure stage $(h_{g1})_{LPT}$ comparing to the computed ones. This deviation may be due to the uncertainty in both the measured pressures and evaluating the last stage exit condition. Finally, the present figure gives also the predicted and measured values of both turbine total isentropic efficiency h_{gtot} and plant net thermal efficiency h_p . It is apparent here that the predicted values are less than the design ones. This may be due to all the above reasons. Thus, a non-accurate predicted value of h_p at 3.86 MW comparing to the design one is evident in Figure (6). This is mainly due to increasing both the pass-out steam flow rate and condenser losses.

The plot of the measured and predicted elements of plant energy distribution at different loads of NCMP plant is shown in Figure (7). In discussing the results of Figure (7), it can be observed firstly that increasing plant load tends generally to increase the rate of pass-out steam and consequently increasing process energy (about 34% of cycle energy). This tendency reflects in condenser losses due to the expected reduction in rates of turbine exhaust. Thus, increasing process energy (PE) with plant load increasing causes also the plant net power (NP) to be decreased as shown clearly in Figure (7). Furthermore, increasing the factory demand from heat and power is accompanied with increasing boiler productivity and also its fuel consumption. The preceding reasons cause the boiler and unaccounted losses are increased proportionally with the plant load as observed in Figure (7). This tendency is mainly due to increasing the flue gases exit temperature with increasing boiler fuel consumption. As can be seen also in Figure (7), the comparison between the measured and predicted values of process energy and condenser loss shows a considerable disagreement. On the other hand, the agreement between the measured and predicted values of the remainder energy distribution elements is not as good but is considered reasonable. The reason for the discrepancies between the measured and predicted values of process energy and condenser loss is mostly due to increasing the actual condenser loss in NCMP plant. This increment in condenser loss of NCMP plant has been explained as the plant load increases, the temperature of condenser cooling water increases also due to the insufficient number of the cooling tower cells.

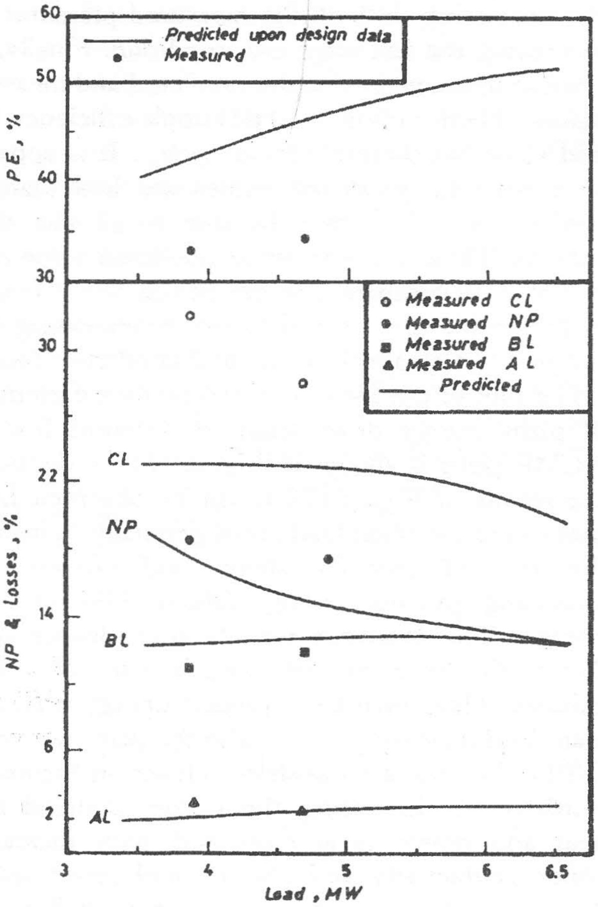

Figure 7. Variation of energy distribution elements with load for NCMP plant.

The energy distributions as well as the overall thermal efficiency of the plant at part loads of 3.86 MW and 4.7 MW are presented in Figure (8). In discussing results of Figure (8), it can be observed firstly that the energy converted into electrical power is low. This is because the factory demand of process steam represents about 50% of live steam flow. Quantities of both process (or pass- out) steam and condenser loss are the biggest terms of the plant energy distribution. The comparison illustrated in Figure (8) between the different heat balance elements indicates that there is a small increase in both auxiliary (AL) and boiler (BL) losses. Besides, increasing the percentage of process steam heat (PE) with the load is the reason for decreasing plant net power (NP) and condenser loss (CL). Moreover, energy distributions in the other two plants, i.e. MCWT and DCSP plants, were presented and discussed with details in Ref.[11].

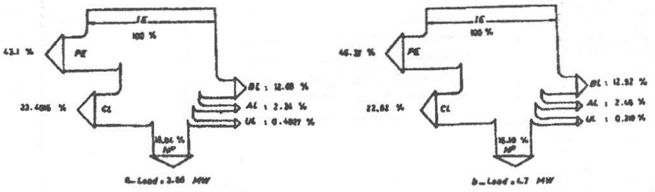

Figure 8. Energy distribution in NCMP plant based on design values data.

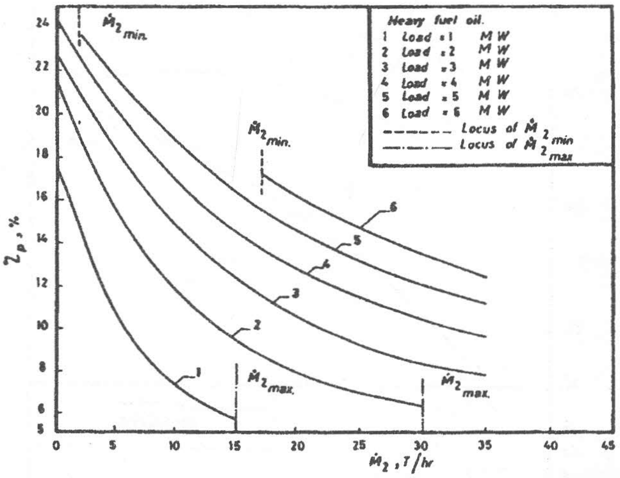

Figure 9. Variation of the plant thermal efficiency with $\dot{M}_2$ at different electric loads for NCMP plant.

In order to obtain useful data for the limits of pass-out steam flow rates, effect of these rates upon the cycle net thermal efficiency is given in Figure (9). Generally, this plot indicates that the plant net thermal efficiency decreases with increasing the pass-out steam flow rates for a certain electric load. The figure shows also the minimum and maximum limitations of pass-out steam flow rates within the range of plant load variation (from 1 MW to 6 MW). Results of Figure (10) were obtained when the plant operates using heavy fuel oil.

6. CONCLUSIONS

An analytical approach of relevance to analyze the performance of thermal power plants for power generation or combined production of heat and power could be constructed here. The developed approach has been applied upon an actual CHP plant in order to evaluate the effect of such operating parameters on the plant performance. It is shown for such a CHP plant that the plant efficiency is mainly affected with both process steam flow rate and condenser heat loss. Other parameters such as; boiler losses, pumping losses and unaccounted losses are found to slightly affect the efficiency. Furthermore, the results show the limitations for process steam flow rates corresponding the turbine output power. The comparison between the approach predictions based on design data and measurements in three local CHP plants indicates a reasonable agreement. It should be mentioned finally that utilizing the present approach, consequently the suggested computer program, can serve the power plant users and engineers to define and evaluate the energy dissipation generally in the industry and specially in CHP plants.

7. REFERENCES

[1] D. Foster, "Economic performance optimization of a combined heat and power station", *Proc. Inst. Mech. Engrs*, vol. 201, No. A3, pp. 201-205, 1987.

[2] J.B. Young, "An equation of state for steam for turbomachinery and other flow calculations", *Trans. ASME, J. Engng for Gas Turbines and Power*, vol. 110, pp. 1-7, 1988.

[3] F.T. Morse, *Power Plant Engineering*, East-West Press, New Delhi, India, 1974.

[4] S.I. Sabry, N.H. Mahmoud and A.M. El-Ghonimy, "A theoretical approach for energy saving in industrial steam boilers" , *Proc. of 1st Int. Symp. on Heat Exchangers & Boilers* (Fabrication and Safe Operation), Egyptian Atomic Energy Authority, Alexandria, Egypt, pp. 175-195, 23-24 June, 1993.

[5] Z.H. Lin, :_"Thermohydraulic design of fossil fuel fired boiler components", Chapter (8) in: *Boilers, Evaporators and Condensers*, Edited by: Sadik Ka'Kac, John Wiliy & Sons, New York, 1991.

[6] M.M. El-Wakil, *Power Plant Technology* , Mc-Graw Hill, New York, 1984.

[7] W.J. Kearton, "Back pressure and pass-out turbines", Chapter XVII in: *Steam Turbines-Theory and Practice*, Pitman Publ., Liverpool Univ., 1973.

[8] A. Kostyuk and V. Frolov, "Steam turbines for combined production of electric energy and heat", Chapter (7) in: *Steam and Gas Turbines*, Mir Publishers, Moscow, 1988.

[9] F.M. El-Mahallawy, (Edt.), "Performance and improvement of local thermal power stations", ERC/FRCU Project No. C/I/1, Progress Report No. 1, Cairo Univ., Egypt, 1987.

[10] "Standards of plant performance characteristics", Manufacturers Association.

[11] A.M. El-Ghonimy, "Applied study on steam energy conservation in the industry", *M.Sc. Thesis*, Faculty of Engng, Menoufia Univ., Egypt, 1995.

[12] B.G. Srotzki and W.A. Vopat, *Power Station-Engineering and Economy*, McGraw-Hill, New York, 1960.