THERMODYNAMIC OPTIMIZATION FOR REPOWERING MARINE GAS TURBINES

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ABSTRACT

A wide scan for thermodynamical parameters affecting the thermal analysis of marine combined gas-steam cycles has been carried out. The influence of three major parameters of the gas turbine cycle in addition to four principal parameters related to the steam cycle has been investigated. By repowering existing gas turbines, additional power and energy savings are achieved represented by the amelioration of the combined cycle thermal efficiency and the reduction of the overall specific fuel consumption at the expense of adding a moderate heating surface for the heat recovery boiler.

INTRODUCTION

The rise in fuel costs is a stimulus for the tendency to improve the efficiency of devices which convert thermal energy into mechanical work.

The efficiency of existing gas turbine power plants can be greatly improved by converting them into combined cycle power plants. By combining the gas turbine and steam turbine cycles, a new thermodynamic process is created with a high upper and a low lower temperature, whereby the cycle approaches the ideal form proposed by Carnot .

For a relatively low extra outlay, it is possible to raise the total plant efficiency from about 30% for gas turbines to 40% to 45% [1,2]. Neither extra fuel, nor additional fuel treatment is required for the steam cycle of combined cycle plant, and there is no increase in environmental pollution [1].

Conversion of an existing simple-cycle unit entails the use of a waste heat recovery boiler, on the gas turbine exhaust path and using the steam production to drive a steam turbine. This enables a 50% power increase for the same fuel consumption compared with simple-cycle gas turbines [2,3]. The increased power output resulting from combined cycle conversion primarily depends on the flow rate and temperature of the gases at the gas turbine exhaust outlets [3].

The development of gas turbines is mainly tied to the solution of technological problems connected with the increasing of the maximal gas temperature which is in the range of 800 - 1250 °C in the biggest engines. On the other hand, the use of such high temperatures required high pressure ratios to obtain good efficiency and prevents the utilization of cheap fuels[4].

Pfenninger H.[5] analyzed a combined gas-steam cycle power plant in which the maximal gas temperature was 880 °C and concluded that for maximum combined cycle efficiency the gas turbine cycle must have the maximum exhaust gas enthalpy (i.e.,the maximum product of the mass flow rate of the gas and its temperature). The resulting pressure ratio was intermediate between those for which the specific work of the gas turbine and its efficiency are maximum.

Wunsch A.[6] claimed that the efficiency of combined gas-steam power plants was more influenced by the gas turbine parameters (Θ,β) than by those of the steam cycle, the parameters of which were fixed in an appropriate manner. The conclusion was that the maximal combined gas-steam efficiency was reached when the exhaust gas turbine temperature was higher than the one corresponding to the maximum gas turbine efficiency.

Fraise and kinney [7] analyzed three gas turbine combined cycles, one of which was a single pressure gas-steam cycle without regenerative preheating of the feedwater.They assigned high parameters of steam conditions (12.41 MPa,811 K) and of gas turbine high gas temperature (1700 K); moreover, other parameters of less importance for the analysis were assigned. The maximum acceptable pinch point value was 28 K (50 °F). three values of pressure ratio were analyzed: 12, 16,20; efficiency and specific work output versus steam air mass flow ratio were given.

Tomlinson and George [8] analyzed a gas-steam combined cycle without supplementary firing. The major assumptions underlying the analysis were that the efficiency of the steam cycle varies slightly with the characteristics of the gas turbine (Θ,ß) plants. The conclusion was that the gas-steam cycle overall efficiency was influenced only by the specific work of the gas turbine and consequently it reached the maximum value when the pressure ratio corresponded to the maximal specific gas turbine work.

In [9] a thermoeconomic assessment of the marine combined gas/ steam cycle was carried out provided that standard steam parameters with prerequired steam turbine's power are to be used.

One obvious incentive for repowering over new construction is the savings in capital investment. Other important advantages of repowering are lower operation and maintenance costs resulting from sharing existing support facilities and staff. There are also several disadvantages to repowering. One is the added complexity that can result when matching old equipments with modern, high efficiency equipments. A second is the potential for increased maintenance frequency of refurbished equipment. The weight of incentives over disincentives provided the basis to study repowering rather than construction [10]

THE MARINE CYCLE UNDER DISCUSSION

The marine combined gas-steam cycle (**COGAS** or **STAG**) under consideration is composed of a simple gas turbine power plant (compressor,combustion chamber and gas turbine) combined through a single pressure finned tubes crossflow heat recovery boiler **HRB** with a steam regenerative cycle. The steam cycle consists of a steam turbine,steam condenser, make up evaporator and the feedwater heaters. Either two or three feedwater heaters are incorporated into the steam cycle; the deaerator being always the second heater. Only a make up evaporator is used for steam compensation , whereas salt water and contaminated water evaporators are discarded. Despite the air ejector and the drain cooler are eliminated from the cycle for simplicity , their thermal influence does not attain 6 °c . Since the steam pressure is restricted to 2500 KPa with a superheated steam temperature (T_{sup}) less than that of the exhaust gas temperature at the outlet of gas turbine ($T_{4'}$) by **DTS**= 50-100 K, supplementary firing can be dispensed . With the relatively low conditions of steam , the steam cycle is intended to electric load generation rather than the main propulsion .

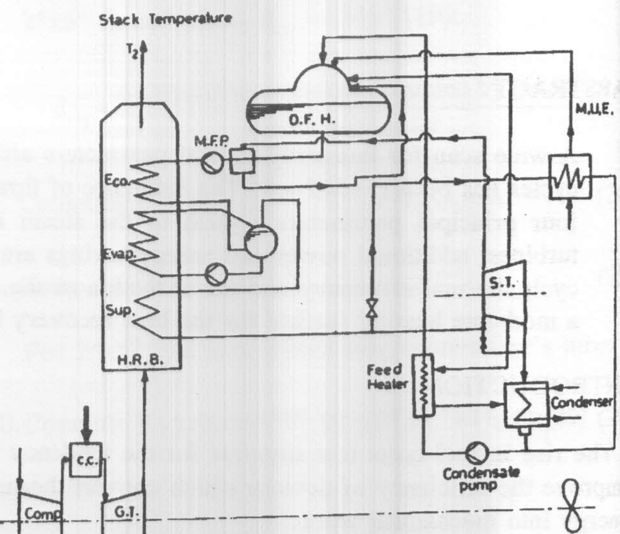

Figure 1. The marine combined gas-steam cycle COGAS.

NOMENCLATURE AND DATA FOR PARAMETRIC ANALYSIS

A,B,C	Temperature rise distribution ratios for low,medium and high pressure feedwater heaters respectively.
DT	The temperature difference between the exhaust gases and the superheated steam; K
FWT	Feedwater temperature; K
HSTOT	Total heating surface; m^2
HSWG	Heating surface per gas mass flow rate; $m^2.s/kg$
HSEC	Economizer heating surface; m^2
NBS	Non-bled steam
NH	Number of heaters
P	Boiler pressure
P_c	Condenser pressure; KPa
POWG	Gas turbine power; KW
POWS	Steam turbine power; KW
T_1	Ambient temperature; K

T_2	Stack gas temperature; K
T_3	Maximum gas temperature; K
T_4'	Exhaust gas temperature; K
T_c	saturation temperature of water at condenser pressure; K
T_{sat}	saturation temperature of steam at boiler pressure; K
T_{sup}	Superheated steam temperature $=(T_4'-DT)$; K
β	Pressure ratio
θ	Maximum temperature ratio = T3/T1
η_{cc}	Combined cycle efficiency
η_{gt}	Gas turbine adiabatic efficiency
η_{gc}	Gas cycle efficiency
η_{sa}	Steam turbine adiabatic efficiency
Ψ	The ratio of the total temperature drop of the gases to the superheated steam temperature $= (T_4'-T_2)/ T_{sup}$

DATA:

Air compressor:

$\beta = 6, 8, 10$
$\eta = 0.85$
$T_1 = 288.15$ K

Gas turbine :

POWG= 15000,20000,25000 Kw
$\theta = 4, 4.5, 5$
$\eta_{gt} = 0.9$

Fuel used:

Sulphur content	= 0.02
Chemically correct air/fuel ratio	= 14,332
Calorific value	= 44122 kj/kg

Heat recovery boiler :

P	= 1500 ,2000 ,2500	KPa
DT	= 50-100	K
Outer tube diameter	=3.8	Cm
T2	= FWT + 30	K

Steam turbine:

$\eta_{sa} = 0.9$
$P_c = 6.23$ KPa
$T_c = 310$ K
NH = 2 , 3

Initially assumed steam amount to condenser= 0.9 NBS
Initially assumed steam to deaerator = 1.15 NBS

Temperature distribution between heaters (Table 1) is based on the optimum temperature rise = $[\{40+8.\text{NH}\}(T_{sat}-Tc)/100]$

Table 1.

	2 Heaters			3 Heaters			
	1	2	3	I	1	2	3
1st heater A	1/2	1/3	2/3	1st heaters A	1/2	1/3	1/6
2nd heater B	1/2	2/3	1/3	2nd heater B	1/3	1/6	1/2
				3rd heater C	1/6	1/2	1/3

NUMERICAL TREATMENT

Thermal analysis of the gas turbine power plant was carried out. It was assumed that the gas cycle operates according to Brayton standard cycle. From the knowledge of ß,θ and power delivered by the gas turbine, all the thermodynamical properties at all points of the gas cycle were determined. The gas turbine power plant efficiency, excess air factor, rate of exhaust gases, and enthalpy difference in both the compressor and gas turbine were calculated. In addition, both the specific fuel and gas consumption were computed. Next to the gas turbine power plant, comes the steam power plant.

In order to obtain the superheated steam temperature, the exhaust gas temperature out of the gas turbine is decreased by DT = 50 - 100 °C. From the predetermined boiler pressure, the computed superheated steam temperature, adiabatic steam turbine efficiency and selected number of heaters the steam cycle could be thermally analyzed. The optimum temperature rise through the feedwater heaters was calculated .

From tables of saturated and superheated steam properties which were completely programmed and processed to the digital computer-the actual heat drop and steam rate were determined .

To proceed calculations, three initial assumptions should be made and by iterative procedures these assumptions were corrected till the exact values were established. The preassumed parameters are the feedwater temperature, the amount of steam to condenser and the amount of steam to deaerator.

The stack gas temperature, non bled steam, evaporation and the power delivered by the steam turbine could be calculated. From the knowledge of the required percentage temperature distribution through each heater both the condensate and the drain bled steam temperature were determined. Accordingly the extracted steam pressure , the replacement factor, the amount of bled steam to each heater and the throttle steam were calculated. A check must be made for the assumed value of the amount of steam to condenser. In case that the calculated amount of steam to condenser does not match with the assumed value, the assumption should be modified in an iterative procedure. Next, both high pressure live steam and low pressure compensating steam from the make up evaporator were computed. At this stage the preliminary heat balance was terminated and precise heat balance at each point of the steam cycle should be carried out. should the calculated amount of steam to deaerator does not correspond, within a specified accuracy, to the preassumed value, the assumption is modified and the iterative procedure should be repeated till matching of both values.

If the cycle incorporates three heaters, heat balance of the third heater should be executed to find all the properties of the condensate, from which the true feedwater temperature is obtained. Both the true and the assumed feedwater temperatures have to be compared, within a prespecified accuracy, which if not fulfilled the iterative procedure on the feedwater temperature should be repeated. At this stage,the power essential to drive the steam driven circulating and condensate pumps could be calculated. To proceed,the overall specific fuel consumption of the combined cycle in addition to the steam to air ratio could be determined. Besides, the power necessary to drive the steam driven feed pump could be evaluated. Care should be made concerning the condensate pressure behind the feed pump whether either two or three feedwater heaters are incorporated. At the termination of the computation of steam power plant , the steam cycle efficiency, the combined cycle thermal efficiency and the power ratio of the turbines ;namely the ratio of the power delivered by the steam turbine to the total power of the gas and steam turbines; are computed. Lastly, thermal design of the heat recovery boiler, the linking equipment of the gas and steam turbines power plants, should be carried out. The significance of such calculations is that if the thermal design of HRB fails, hypothetical non-realistic combined cycle can exist. Foster Wheeler incorporation design procedure for steam generating equipments was adopted. The superheater, evaporator and economizer portions of the heat recovery boiler were separately dealt with. Due to the relatively poor heat transfer coefficient at the gas side ,all the tubes of the HRB were considered of the extended surface type. In order to achieve the thermal design of HRB, the heat transfer from the gas to the steam side should be first calculated, from which the true value of specific heat of the gases could be adjusted. The gas mass flow, the kinematic viscosity of the gases, and Reynold's number have to be determined; from which the convective heat transfer coefficient could be deduced. Furthermore, the radiative heat transfer coefficient is next calculated. From the knowledge of the amount of heat transferred, the total heat transfer coefficient and the logarithmic mean temperature difference; the heating surface could be computed. Besides, the thermal efficiency of the heat recovery boiler **HRB** -if required- could be obtained.

To achieve this task lengthy integrated **FORTRAN** programs were designed and were executed on the **VAX/VMS** version **V 5.3** digital computer of the Faculty of Engineering, Alexandria University.

RESULTS AND DISCUSSION

The relationship between **POWG** and **POWS** for different values of **ß** and **Θ** which is indicated in Figure (2), is almost straight line. The higher the value of **Θ** and the lower the value of **ß** yield the higher value of **POWS** for a specified value of **POWG** .

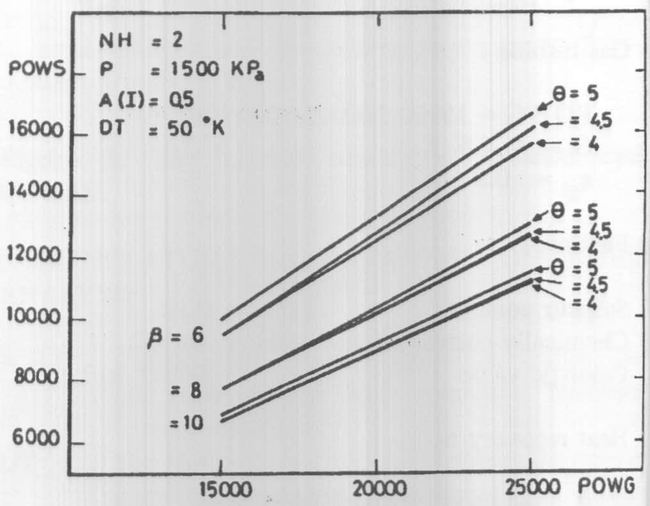

Figure 2. POWG and POWS for different values of ß and Θ.

An illustration of T_4' versus ß at various values of Θ, which represents linear relationship, is displayed in Figure (3). In order to select a standard superheated steam temperatures for the steam turbine say 727.5 K, 755 K and 783 K with a reasonable temperature difference tolerance between the steam and gases at any required value of Θ, the value of ß could be predicted from this plot as a preliminary design.

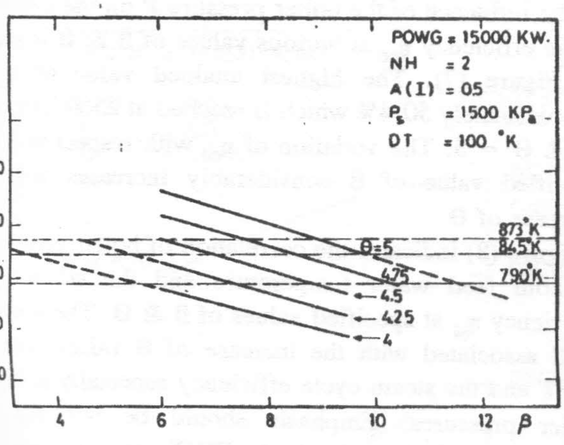

Figure 3. Exhaust gas temperature T_4' versus Pressure ratio β.

The ratio of the total temperature drop of the gases to the superheated steam temperature Ψ versus the boiler pressure P at varying values of β is represented in Figure (4). For higher boiler pressures the curves are seemingly flattened while the lower the value of β, the higher the value of Ψ which is emphatically evident at values of β lower than 8.

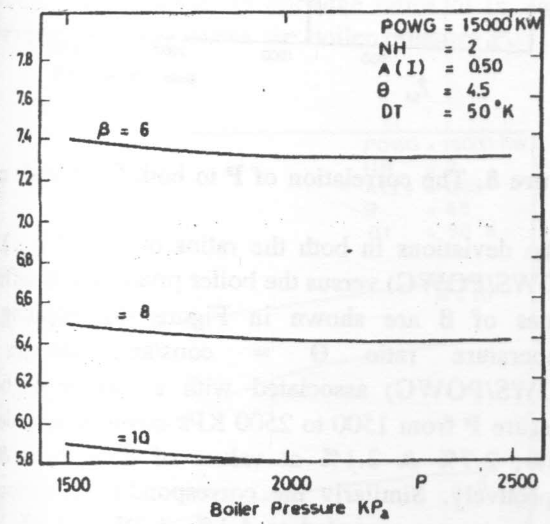

Figure 4. Ψ versus the boiler pressure P at varying values of β.

Cross-curves relating (η_{cc}/η_{gc}) to (POWS/POWG) at different values of Θ, β and boiler pressure P for either two or three feed heaters are plotted in Figure (5) & (6) respectively. These plots are independent on POWG and represent powerful charts relating the most significant thermodynamical relationships of the problem under analysis. In what concerns Figure (6), with temperature distribution ratios 1/2, 1/6, 1/3 for the low, medium and high pressure feed heaters respectively, it is noticed that the value of $(\eta_{cc}/\eta_{gc})$is relatively lower than the corresponding one in Figure (5) despite the number of heaters was increased from two to three. Since the gas turbine cycle's efficiency is invariable in both charts at corresponding points, it results that η_{cc} assumed a small drop. This may be attributed to a drop in the steam cycle efficiency with three heaters than that with two heaters due to the uneven temperature distribution ratio through the heaters.

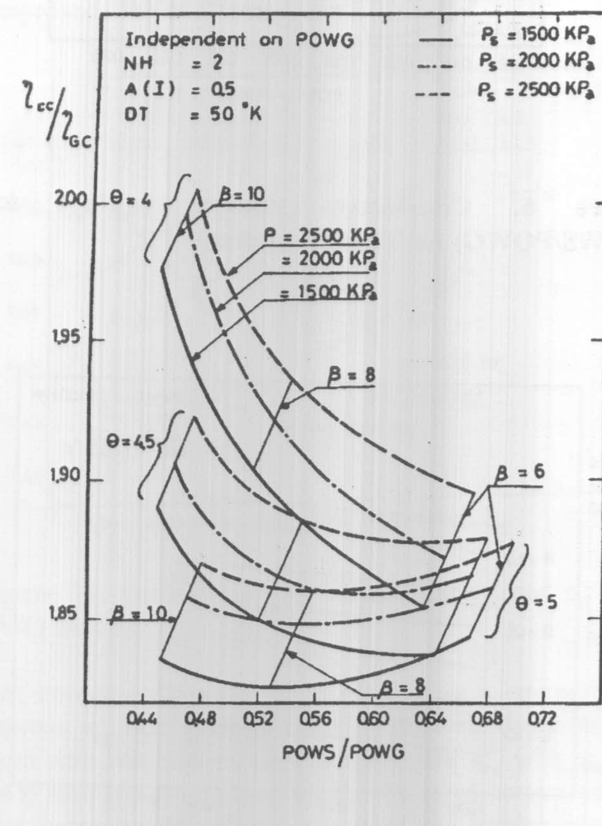

Figure 5. Cross-curves relating (η_{cc}/η_{gc}) to (POWS/POWG) for TWO heaters.

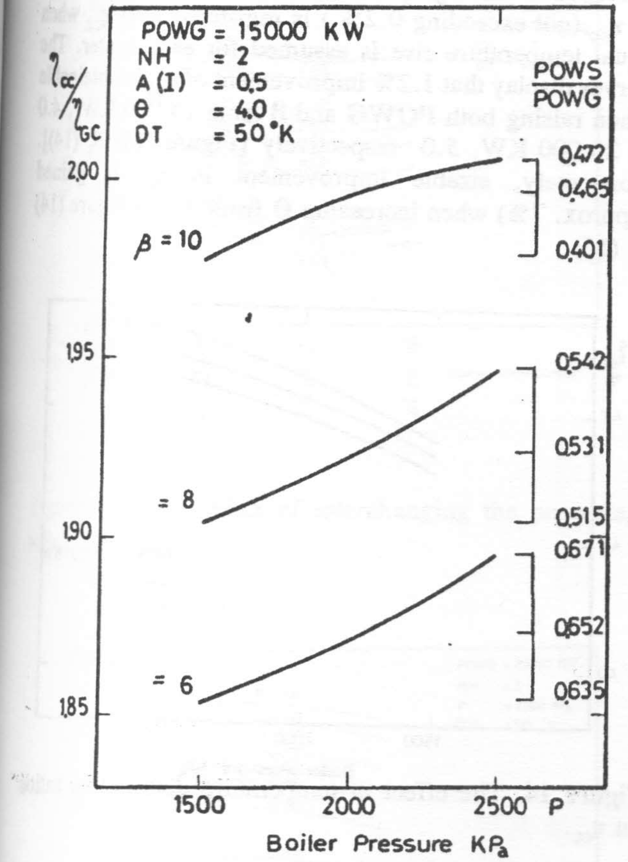

Figure 9. The deviations in both the ratios of (η_{cc}/η_{gc}) and (POWS/POWG) versus the boiler pressure **P**.

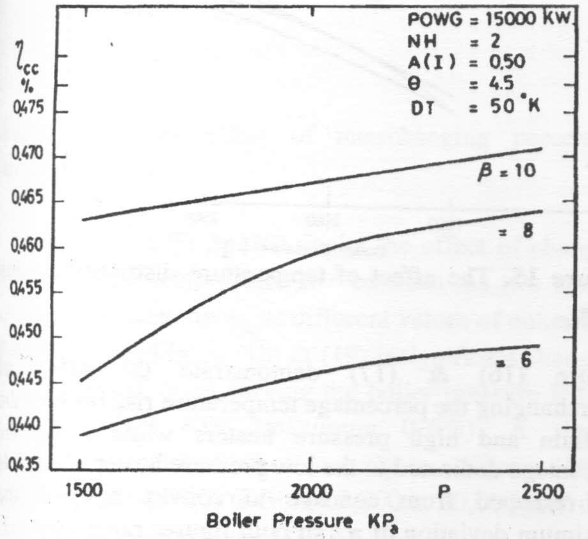

Figure 10. The effect of **ß** and **P** on η_{cc}

Among the significant parameters affecting the combined cycle efficiency η_{cc} are the pressure ratio **ß** and the boiler pressure **P** when the temperature ratio **Θ** being held constant, Figure (10). The increase of both **P** & **ß** tend to raise η_{cc}. However, the rate of increase in η_{cc} depends mainly on the value of **ß** and secondly on the value of **P**. At relatively low pressure region it is beneficial to use a pressure ratio **ß** = 10 instead of **ß** = 8 while the gain in efficiency from raising **ß** from 8 to 10 at high pressure region is inconsiderable.

Interesting two grids of curves relating the η_{cc} and the steam cycle efficiency η_{st} at constant boiler pressure lines **P** and constant pressure ratio lines **ß** (corresponding also to constant gas turbine cycle efficiency η_{gc}) are represented in Figure (11) , (12) The steam cycle represented by the data associated with Figure (11) contains two feed water heaters while the data attached to Figure (12) assumes regenerative steam cycle with three heaters. At constant pressure **P** raising **ß** leads to increase η_{gc}, decrease η_{st} while the resulting η_{cc} is improved.

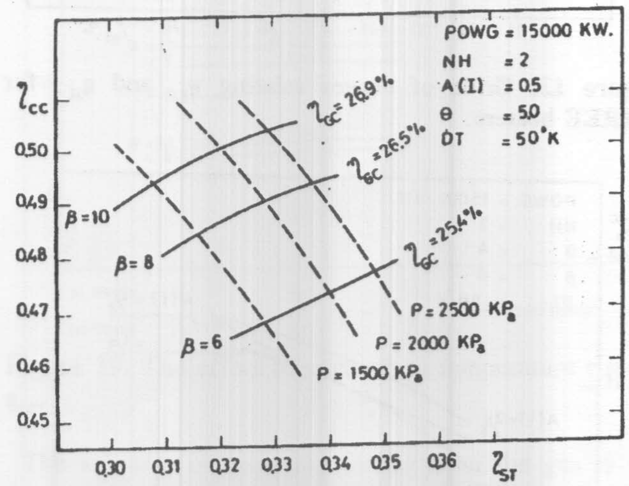

Figure 11. Grids of curves relating the η_{cc} and η_{st} for **TWO** heaters.

At corresponding values of parameters a comparison between η_{cc} in Figure (11) & (12) preveals that a steam cycle with two heaters provides a higher η_{cc} with about 0.4% than that for a steam cycle with three heaters when each being combined to a specified gas turbine power plant.

The uneven distribution of temperature ratios among the three heaters gives the reasoning of this drop in efficiency as mentioned in the discussion of Figure (6).

Concerning the doted lines shown in Figure (12) ,if the

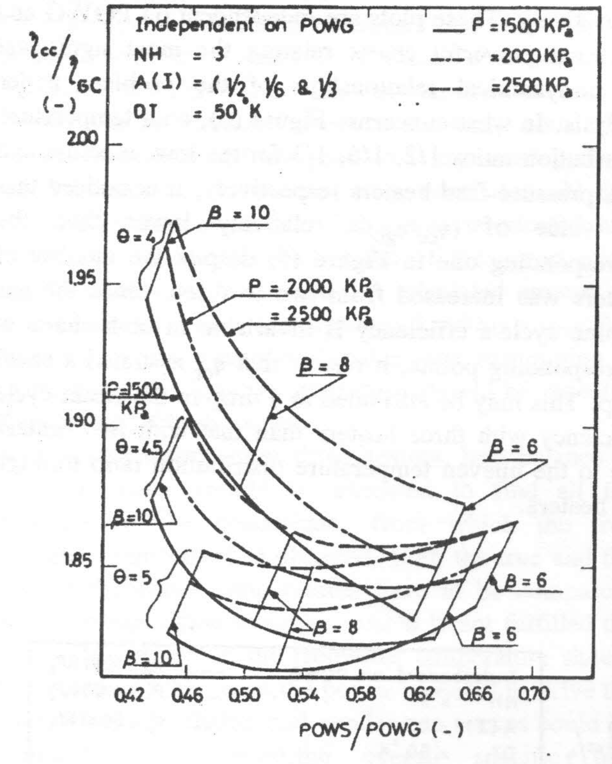

Figure 6. Cross-curves relating (η_{cc}/η_{gc}) to (POWS/POWG) for THREE heaters.

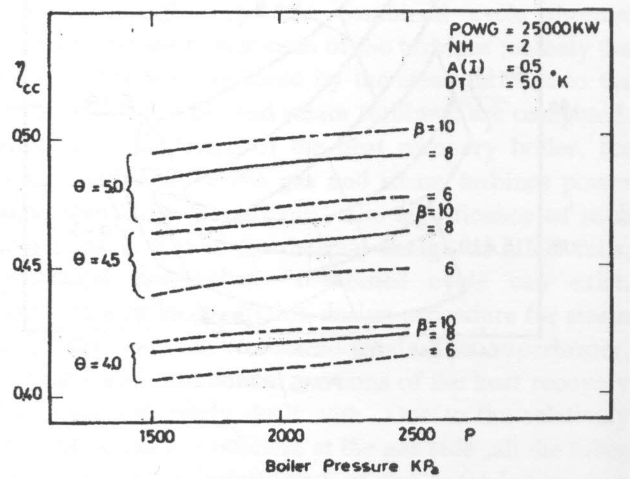

Figure 7. The influence of P on the combined cycle efficiency η_{cc}.

The influence of the boiler pressure P on the combined cycle efficiency η_{cc} at various values of β & θ is shown in Figure (7). The highest attained value of η_{cc} is approximately 50.4% which is reached at 2500 KPa, β = 10 & θ = 5. The variation of η_{cc} with respect to β at a specified value of β considerably increases with the increase of θ.

Figure (8) indicates the correlation of boiler pressure P to both feed water temperature and the steam cycle efficiency η_{st} at specified values of β & θ. The decrease of β associated with the increase of θ raises both the FWT and the steam cycle efficiency especially at higher boiler pressures. Emphasis should be put that the observed variations in both FWT and η_{st} are not considerable.

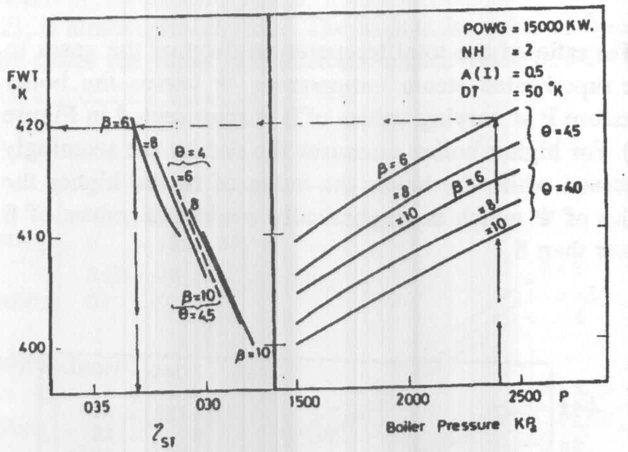

Figure 8. The correlation of P to both FWT and η_{st}

The deviations in both the ratios of (η_{cc} / η_{gc}) and (POWS/POWG) versus the boiler pressure P at different values of β are shown in Figure (9). Holding the temperature ratio θ = constant, changes of (POWS/POWG) associated with a change of boiler pressure P from 1500 to 2500 KPa could be recorded as 3.6%, 2.7% & 2.1% at values of β = 6, 8 & 10 respectively. Similarly the corresponding deviations in (η_{cc}/η_{gc}) were recorded as 4.1%, 3.2% & 2.8% . The maximum value of (POWS/POWG) is obtainable at the highest value of P with the lowest value of β, while the maximum value of(η_{cc} / η_{gc}) could be reached at the highest value of P associated with the highest value of β.

power of gas turbine is raised from 15000 KW to 20000 KW while fixing the value of ß, Θ and the temperature distribution ratios among the heaters (1/2, 1/6, 1/3) a drop in η_{cc} of about 0.4% is recorded.

On the contrary, the temperature ratio distributing factors if varied from (1/2, 1/6, 1/3) to (1/3, 1/2, 1/6) respectively cause an increase of about 0.5% η_{cc} at POWG = 20,000 KW and P=2500 KPa.

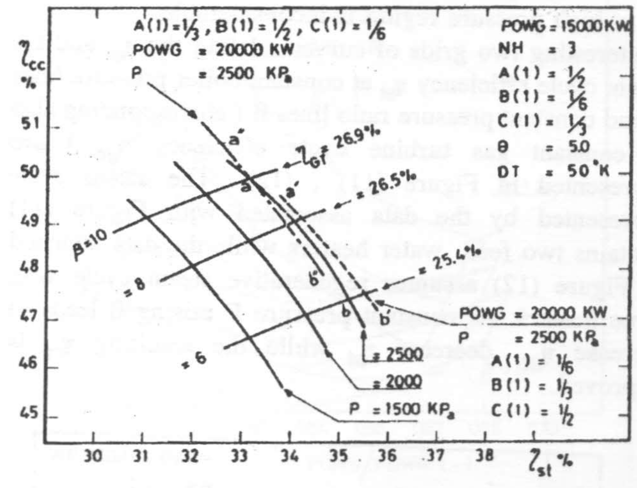

Figure 12. Grids of curves relating η_{cc} and η_{st} for THREE heaters.

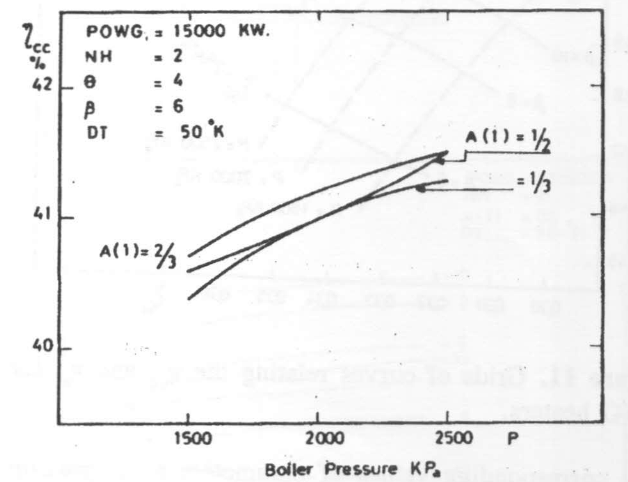

Figure 13. The effect of temperature distribution ratios on η_{cc}.

The effect of varying the temperature distribution ratios between the two feedwater heaters incorporated into the steam cycle on η_{cc} versus the boiler pressure **P** is illustrated in Figure (13, 14 & 15). If 2/3 of the temperature rise is specified to the low pressure feed heater while the other 1/3 of this temperature rise is specified to the high pressure one, an insensible increas in η_{cc} (not exceeding 0.2%) is registered over η_{cc} whe equal temperature rise is assumed for each heater. Th curves display that 1.2% improvement of η_{cc} is obtainabl when raising both POWG and ß from 15,000 KW , 4. to 20,000 KW, 5.0 respectively [Figure (13) & (14)] Conversely, sizable improvement in η_{cc} is gaine (approx. 7%) when increasing Θ from 4 to 5 Figure (14 & (15).

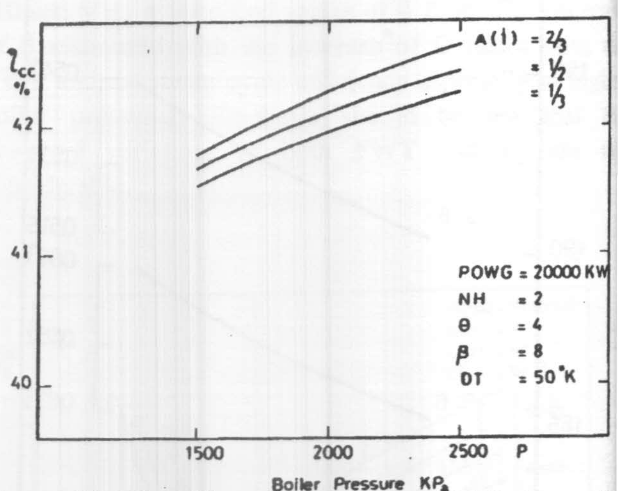

Figure 14. The effect of temperature distribution ratio on η_{cc}.

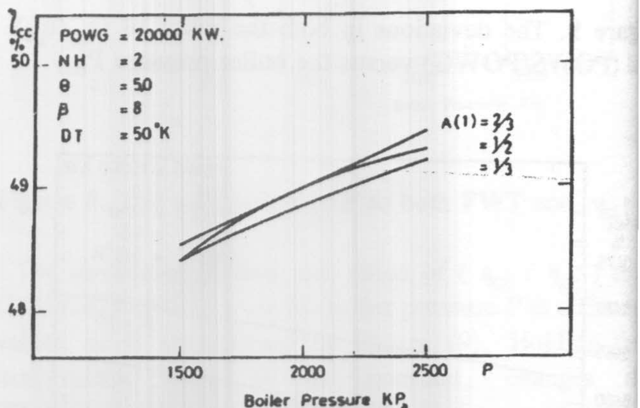

Figure 15. The effect of temperature distribution ratios on η_{cc}.

Figure (16) & (17) demonstrate the effect of interchanging the percentage temperature rise between the medium and high pressure heaters while fixing the percentage dedicated to the low pressure heater. The plots are reshaped from concave to convex however the maximum deviation of η_{cc} in both figures ranges approx. from 0.7% to 0.9%

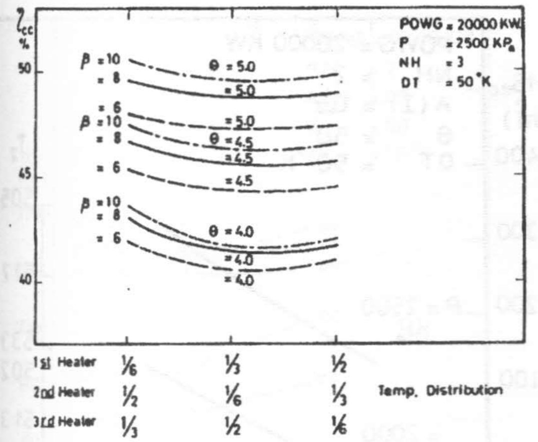

Figure 16. The effect of interchanging the percentage temperature rise.

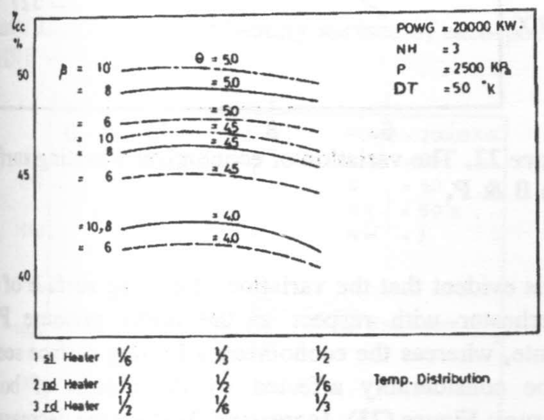

Figure 17. The effect of interchanging percentage temperature rise.

Figure (18), (19) & (20) depict the effect of changing the percentage temperature rise between the two heaters of the steam cycle on η_{cc} at different values of either **P** or POWG. From Figure (18) & (19) insignificant change in η_{cc} is observed when varying the boiler pressure **P** from 1500 to 2000 KPa (maximum 0.6%). A similar conclusion is reached when investigating Figure (18) & (20). Increasing **POWG** from 15,000 to 20,000 KW does not raise η_{cc} by more than 0.2%

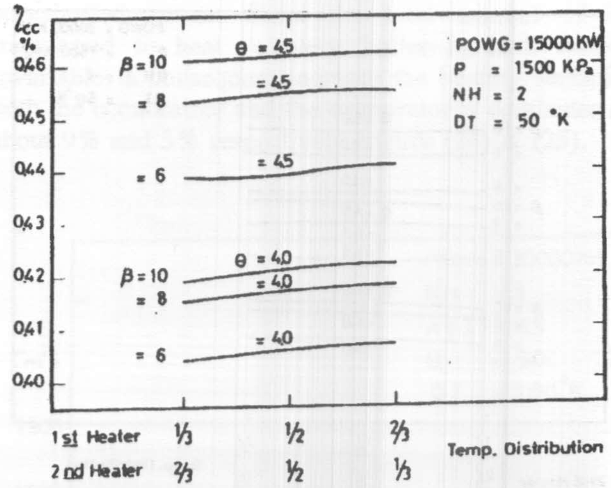

Figure 18. The effect of percentage temperature rise on η_{cc}.

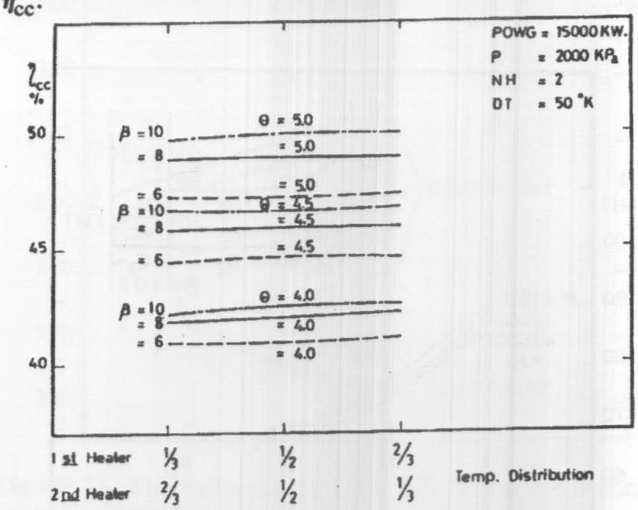

Figure 19. The effect of percentage temperature rise on η_{cc}.

The amounts of heat transferred from the gas to the steam side through each portion of **HRB** namely the superheater, the evaporator, and the economizer (Q_1, Q_2 & Q_3 respectively) against ß at various boiler pressures are indicated in Figure (21). It is clear that Q_1 is nearly insensitive to the variation in ß and represents the least portion of the total heat transferred which lies in the range of 10%. Conversely, the major percentage of heat transferred is absorbed by the evaporator and represents about 60% of the total heat absorbed in **HRB**. The increase of ß tends to decrease Q_1, Q_2 & Q_3 since the heat drop in the gas turbine increases (associated with reduced exhaust gas temperature) and the weight of gases decreases for a specified gas turbine power. On the other hand, the decrease of boiler pressure P tends to increase both Q_2 & Q_3 and reduce Q_1.

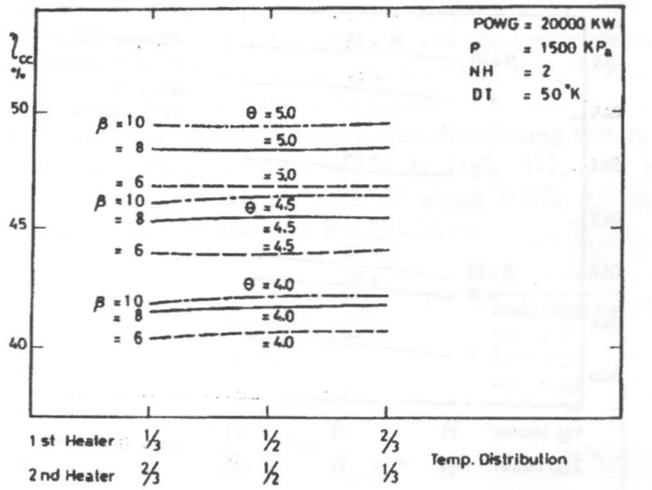

Figure 20. The effect of percentage temperature rise on η_{cc}.

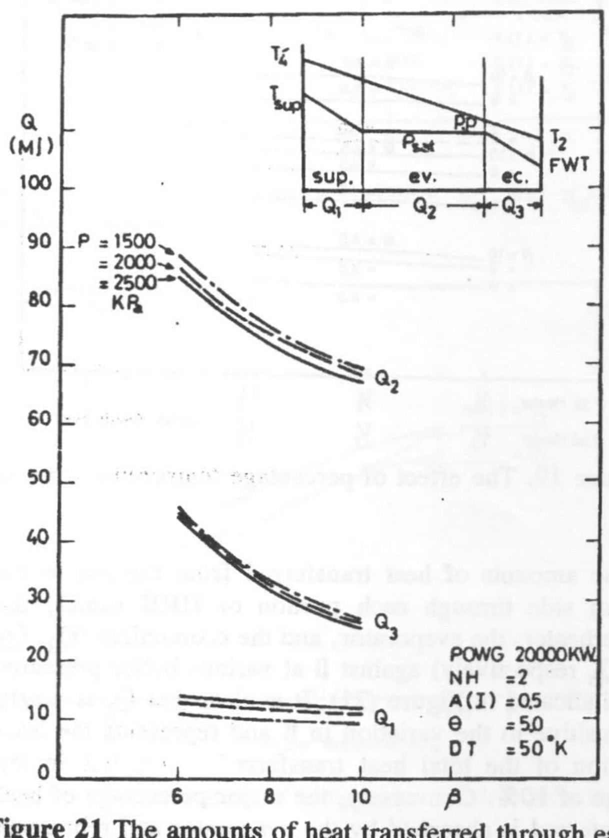

Figure 21 The amounts of heat transferred through each portion of HRB (Q_1, Q_2 & Q_3) against ß.

In what concerns the investigation of the heating surface of the individual portions of HRB, Figure (22) displays the heating surface of economizer whereas figures (23), (24) & (25) illustrate the heating surface of each portion of HRB versus ß at different P.

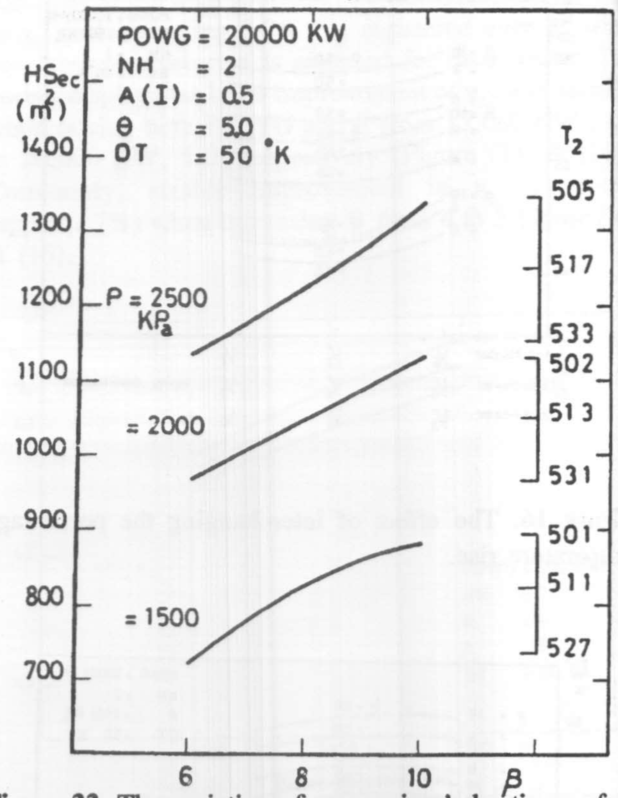

Figure 22. The variation of economizer's heating surface with ß & P.

It is evident that the variation of heating surface of the superheater with respect to the boiler pressure **P** is minute, whereas the economizer's heating surface seems to be considerably affected by the change of boiler pressure; Figure (23). Increasing ß causes the decrease of the superheater surface due to the reduced gas mass flow rate together with the decrease in exhaust gas turbine temperature. On the other hand, the increase of ß tends to increase the economizer surface which may be attributed to the slight deviation in the **FWT** consequently with the stack gas temperature T_2, in addition to the small change in the location of the pinch point **P.P.** obtained when performing thermal equilibrium of the evaporator. This in turn, enlarges the logarithmic mean temperature difference for the economizer and accordingly raises the economizer heating surface. Considering the influence of the number of heaters on the heating surface - Figure (23) & (24)-the increase of number of heaters NH from 2 to 3 reduces the economizer surface by about 200-300 m^2 in the range of ß = 6 - 10, while it does not affect the heating surface of the evaporator and the superheater.

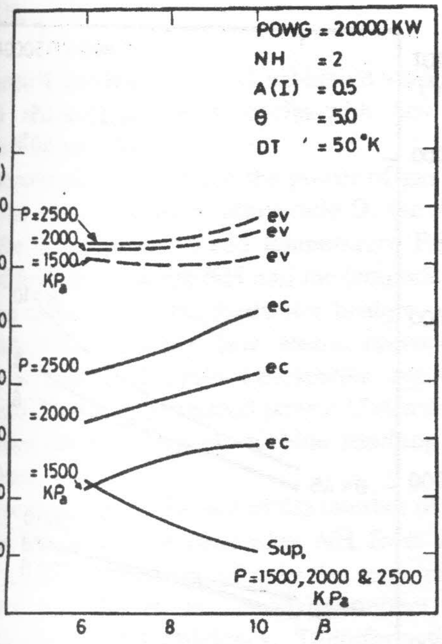

Figure 23. The effect of heating surface of each portion of HRB vs ß & P.

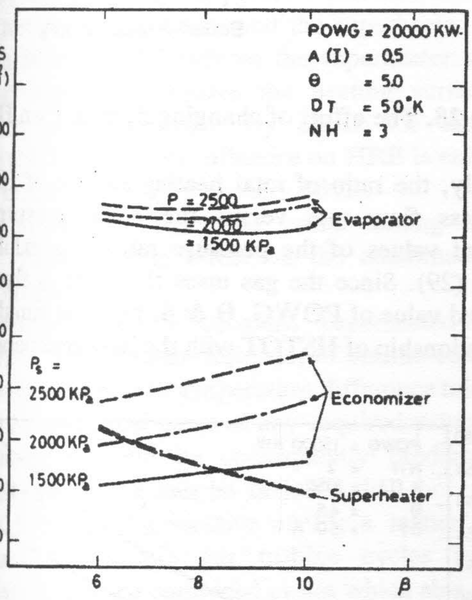

Figure 24. The influence of DT=50 °C on the heating surfaces of HRB.

Certain variations occur in the heating surface of the economizer, evaporator and superheater when increasing the temperature difference between the exhaust gases and the superheated steam from DT = 50 to 100 °C. The superheater surface will be reduce by about 30% due to the decrease in the required heat transfer to the superheated steam. Since the total amount of heat transferred to heat recovery boiler in both cases is invariable, a consequent increase the heating surface of both the economizer and the evaporator is originated by about 9% and 3% respectively Figure (24) & (25).

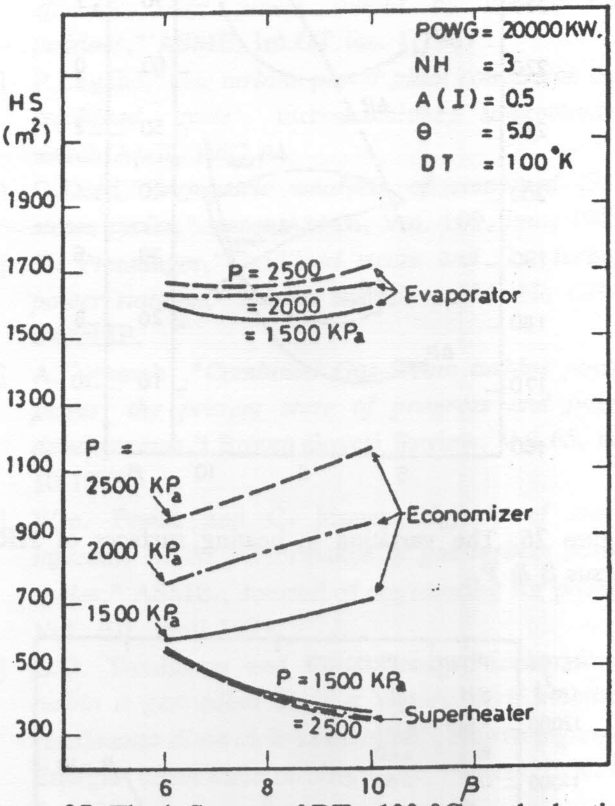

Figure 25. The influence of DT=100 °C on the heating surfaces of HRB.

Figure (26) visualizes the change in heating surface of the economizer, evaporator and superheater versus ß when raising the boiler pressure **P** from 1500 to 2000 KPa(ΔP_1), then when raising the boiler pressure **P** from 2000 to 2500 KPa(ΔP_2).

A pictorial representation of the total heating surface of heat recovery boiler **HSTOT** -which may be regarded as a measure of additional weight and cost- against the boiler pressure **P** and at various values of ß & θ is given in Figure (27) & (28). The total heating surface increases proportionally with boiler pressure **P** and the pressure ratio ß, but it is inversely proportional to θ. An augmentation in ß associated with a reduction in θ magnifies the increase in HSTOT particularly with the increase of **P**.

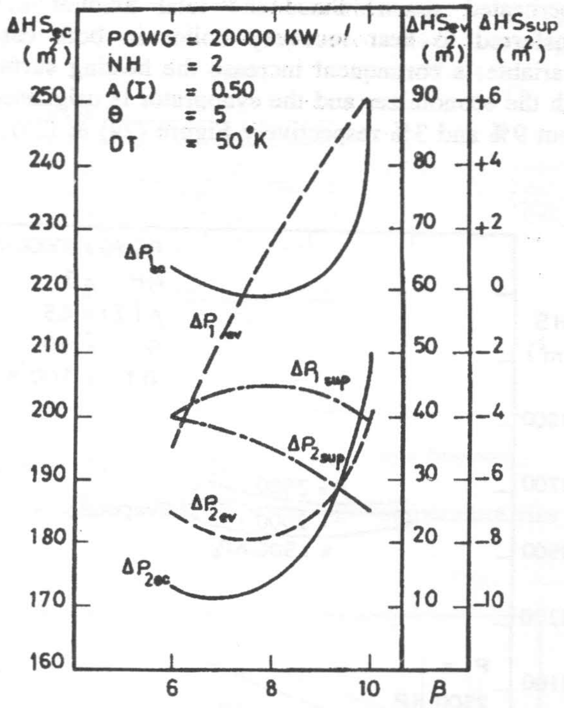

Figure 26. The variation in heating surfaces of HRB versus ß & P.

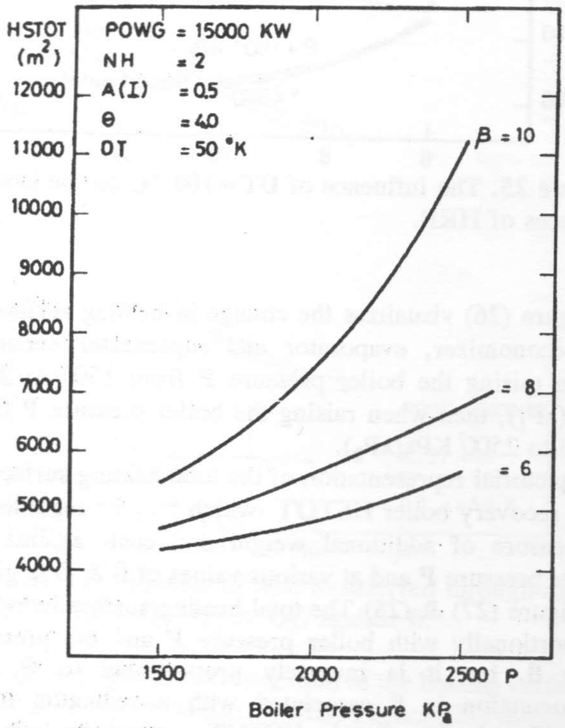

Figure 27. The total heating surface HSTOT vs P for different ß (θ=4).

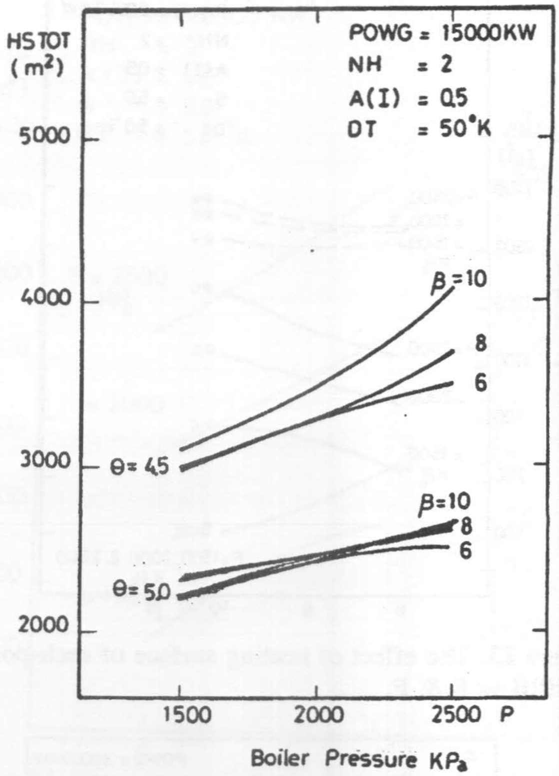

Figure 28. The effect of changing ß, θ & P on HSTOT

Finally, the ratio of total heating surface of HRB per gas mass flow rate versus the boiler pressure P at different values of the pressure ratio ß is exhibited in Figure (29). Since the gas mass flow rate is fixed for a specified value of POWG, θ & ß, the plots simulate also the relationship of HSTOT with the indicated parameters.

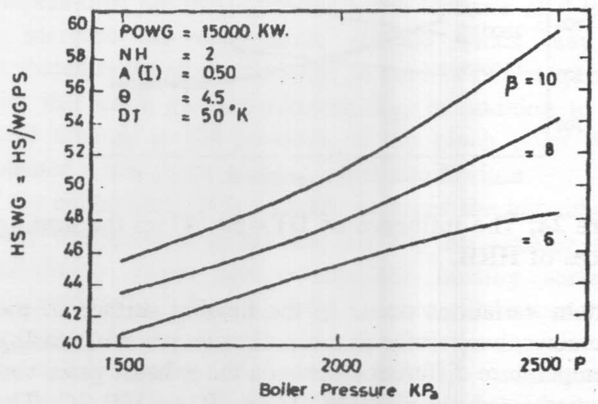

Figure 29. Total heating surface per gas mass flow rate versus P

CONCLUSION

A parametric thermodynamical extensive study of the combined marine gas-steam cycles-with low steam conditions was carried out.

The parameters dealt with were the power of gas turbine POWG, the maximum temperature ratio **Θ**, the pressure ratio ß, the steam pressure and temperature **P&T**, the number of feedwater heaters **NH** and the temperature rise distribution ratios among the feedwater heaters.

When adopting relatively low steam pressures and temperatures combined cycle efficiencies approaching 50.4% besides ratios of generated power of steam turbine to the power delivered by gas turbine reaching 67.0% were realized.

In what concerns the influence of the number of heaters NH it was concluded that increasing **NH** from 2 to 3 , decreases the heating surface of only the economizer by about 9.4% but almost has insensible effect on the combined cycle thermal efficiency. Therefore, the third heater could be eliminated from the steam cycle, sparing the troubles encountered from high pressure heaters.

Likewise, increasing the temperature difference between the gas turbine exhaust gases and the superheated steam DT from 50 to 100 °C reduces the superheater surface by about 30% but increases the heating surfaces of economizer and evaporator by about 9% and 3% respectively. However, its influence on **HRB** is enlarging its total heating surface by about 1.88% .

The temperature distribution ratios among heaters showed inconsiderable effect on the combined cycle thermal efficiency.

In order to select a standard superheated steam temperatures for the steam turbines, of marine combined cycles, with a reasonable temperature difference tolerance between the steam and gases at any required value of **Θ**, the value of ß could be consequently predicted for preliminary design. It can be deduced that the greatest value of ß lies in the vicinity of 13; a matter which emphasizes that simple gas turbine cycles will be appropriate for marine combined cycles which also fulfills the requirement of compactness.

Lastly, the thermodynamical charts demonstrated in this research throw the light on the preliminary selection of the significant combined cycle parameters together with the resulting performance a priori to the final design stage.

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