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"ENERGY STUDY OF A GAS-FIRED TURBO-GENERATOR"

By

M.R.S. Okelah,
Assistant Professor,
Faculty of Engineering,
Qatar University, Doha,
Qatar, Arabian Gulf.

ABSTRACT

As far as energy utilization is concerned, the performance of a gas turbine-generator combination, depends on some key factors among which are; engine features, methods of utilizing heat energy associated with exhaust gases, fuel characteristics, variations in ambient conditions and variation of electric load. The present work is intended to evaluate specific heat consumption and specific fuel consumption of a 44 MW turbo-generator unit, at Ras Abu-Fontas Power and Water Station, in view of likely changes in ambient conditions of Qatar. The study also covers the effect of load changes on specific heat consumption and component efficiencies.

The method used for performance evaluation is herein explained; necessary on-site measurements are identified and procedures used in the analysis are outlined. Validation of results is confirmed through comparison between calculated turbine inlet temperatures and measured values. Excellent agreement between these temperature values is shown to exist. Causes of energy loss are also discussed.

1. INTRODUCTION

Ras Abu-Fontas Power and Water Station is now the primary source of electrical power and potable water in Qatar. Started in 1972 and completed in early 1984, it is one of the largest installations of its type in the world (1). The final configuration of Ras Abu-Fontas comprises twelve gas turbine generator units and two black-start gas turbines with a total rating of 622 MW. which is equivalent to total ISO rating of 965 MW.

The performance of such a plant, as far as energy consumption is concerned, depends on a number of factors. One basic factor thereof is the design feature of the gas turbine-generator units, which in turn depends mainly on compressor pressure ratio, maximum cycle temperature (Turbine Inlet Temperature) and engine component efficiencies. A second factor is the method of utilizing heat energy associated with exhaust gases. Once the engine is installed on site these two factors are fixed. A third basic factor is the variation of ambient conditions, temperature, pressure, humidity ratio and dust content. Among these, temperature is the most influencial factor affecting the thermodynamic performance of the engine. Generator loading has also remarkable effects on engine performance. There are some other important factors that should be taken into consideration in an energy study such as: availability of fuel, its characteristics and price. The present study evaluates engine performance and energy consumption of a selected gas turbine unit in view of those factors which are likely to vary during operation of the turbo-generator unit.

2. DESCRIPTION OF PLANT

The twelve large gas turbine-generators are housed longitudinally one after the other in a turbine building of 722 m in length. Along the front of the building, on the eastern side, are the air intake structures, the waste heat boilers with the by-pass stacks and the desalination plants. Local control stations, the electric switchgear and the transformers are arranged in the western side. A view of the plant is shown in Fig. (1). Among the twelve gas turbine-generator units, six are manufactured by Kraftwerk Union AG (KWU), and are of the V.93 type. The other six units are manufactured by Mitsubishi Heavy Industries Ltd. (MHI), and are of the MW 701G type. They all operate in base-load mode, with filtered air intakes. The first two KWU gas turbines, supplied for the initial installation, operate at turbine inlet temperatures of approximately 820^{0}C at base load and at approximately 850^{0}C at peak load. These values were increased to approximately 850^{0}C and 870^{0}C for the four KWU gas turbines installed in the second extension of the plant.

The gas turbine-generator (turbo-generator) unit considered in the present study is one of the two KWU units in the initial installation: it is identified as unit GT.2 in the power station. Exhaust gases from each KWU turbine, at a temperature of 440^{0}C, are used to generate saturated-steam flow of 126.6 Mg/hr in the succeeding forcedcirculation drum type boiler. This rate is increased to 160 Mg/hr with the aid of additional gas burners. The steam pressure is 14 bar. The feed water is deaerated at a temperature of 140^{0}C and the temperature of the return flow from the desalination plant is 90^{0}C. In addition, the plant has four auxiliary boilers at hand, each capable of producing saturated steam of 80 Mg/hr; these can be connected to the desalination plant even if the gas turbines are not operating. Gas turbines thus provide mechanical power for generation of electricity, which is the primary product of the plant. Moreover, heat rejected from turbines is used to generate steam for production of potable water, this being considered a by-product of the plant. One consequence of such an arrangement is that the heat utilization efficiency of the whole plant improves substantially, while the thermal efficiency of the gas turbine becomes not as important as in the case of gas turbines entirely devoted to the production of electricity.

3. GENERAL DESIGN FEATURES OF GAS TURBINE UNIT GT.2

KWU gas turbines are single-shaft machines of standard design. They are suitable for driving generators in base-load and peak-load plants. They can be used in combined gas-steam cycles, also for provision of district heating. They can burn liquid fuels such as light or heavy fuel oils as well as gaseous fuels such as natural gas or blast-furnace gas.

The compressor and turbine, the principal components of a single-casing single-shaft gas turbine, have a common rotor. It is supported in only two journal bearings and located by a single thrust bearing, all of which are located outside the hot pressure area of the machine.

The turbine rotor is internally cooled. A small percentage of the compressed air is bled off from the main flow at the end of the diffuser and is admitted to the interior of the rotor through holes in the centre hollow shaft. Air is distributed between turbine discs and is directed to blade root fixings. It discharges thereafter into the hot gas flow thus providing a cooling film over the blade roots in the process.

Two identical combustion chambers are arranged vertically on either side of the turbine and are connected to side flanges on the turbine casing. This design provides short and concentric gas and air paths from compressor to combustion chambers and from chambers to turbine, thus involving minimum flow losses (3). A cross-sectional view of a typical gas turbine engine is reproduced in Figure (2).

Fig. 1: A View of Ras Abu Fontas Power and Water Station (Adapted from reference (2)).

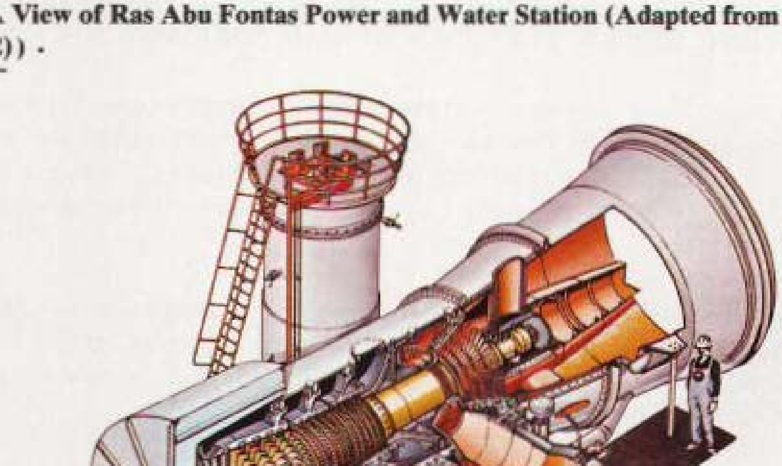


Fig. 2: The KWU, V.93 Type, Gas Turbine Engine (Adapted from reference (3)).

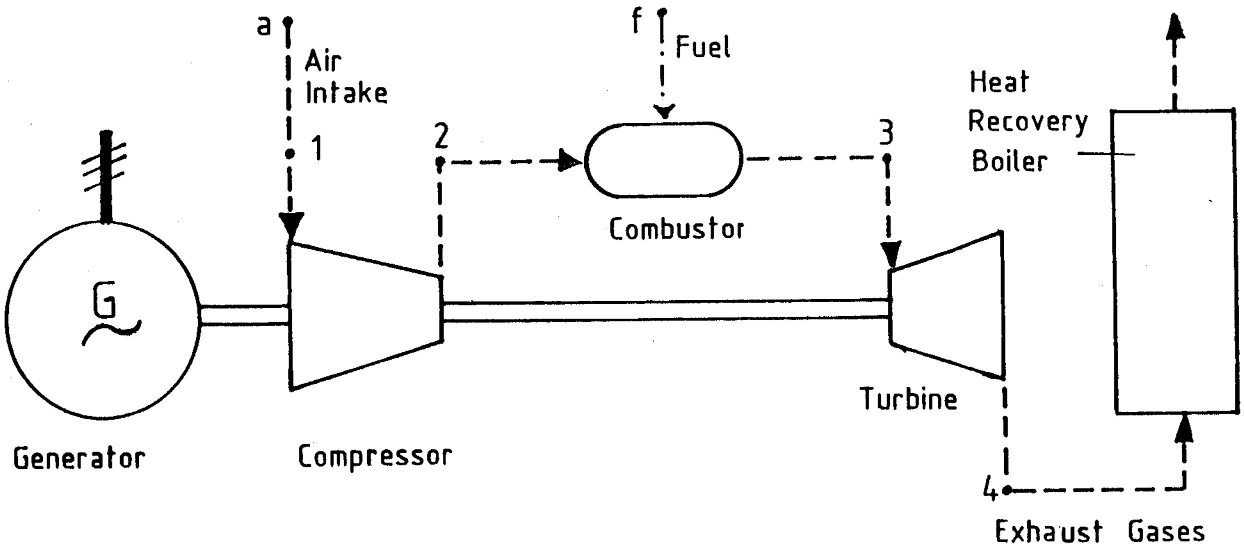


Fig. 3: A Schematic Representation of a Turbo-Generator Unit Showing Station Numbering Adopted for the Study.

4. EVALUATION OF PERFORMANCE

The method of performance analysis used in this study is based on the International Standard ISO 2314-1973 "Gas Turbines Acceptance Tests". It shows how mass flow and turbine inlet temperature of a single-shaft gas turbine can be calculated by means of relatively easily measureable parameters. Mass flow balance, power balance and combustion chamber balance are used for this purpose. Details of the method are presented in reference (4) and may be summarized as follows:

4.1 Measurements Required

The parameters required for performance analysis are listed in Table (1) with subscripts in accordance with station numbering of Figure (3).

In addition to these variables, the following parameters have to be estimated:

— Fuel lower calorific value: Hu, and specific gravity: w
— Generator efficiency: n_G
— Combustion chamber efficiency: n_b
— Mechanical efficiency: n_m, or mechanical power losses

Table (1): Required Measurements for Performance Analysis.

Quantity	Fluid	Location	
a. Temperature, k	Air	— Ambient	: T_a
		— Compressor inlet	: T_1
		— Compressor outlet	: T_2
	Gases	— Exhaust of turbine	: T_4
	Fuel	— Inlet to combustor	: T_f
b. Pressure, bars	Air	— Compressor inlet	: P_1
		— Compressor outlet	: P_2
	Gases	— Exhaust of turbine	: P_4
	Fuel	— Inlet to Combustor	: P_f
c. Flow rate, m^3/hr	Fuel	— Inlet to Combustor	: Q_f
d. Power, KW	—	— Generator Terminal	: P_{out}

4.2 Data Required

In order to carry out performance analysis the following data should be made available:

1. Enthalpy versus temperature tables for air,
2. Enthalpy versus temperature tables for combustion gases,
3. Compressor air flow versus ambient temperature chart,
4. Chart of oxygen content in the products of combustion versus specific fuel output (or fuel-to-air ratio).

This data is provided by KWU in reference (4).

4.3 Basic Equations

The following equations are used to calculate air mass flow rate m_a, enthalpy at turbine inlet h_3, overall thermal efficiency n_o, specific heat consumption SHC, specific fuel consumption SFC, compressor isentropic efficiency n_c and turbine isentropic efficiency n_t.

$$m_a = \frac{m_f\,(n_b\,(H_u + h_f - h_{fs}) - h_4\,) - P_{out}/\,n_m}{h_4 - h_1} \qquad (1)$$

$$h_3 = \frac{m_a}{m_3}\,.\,(h_2 - h_1) + \frac{P_{out}/n_m}{m_3} + h_4 \qquad (2)$$

$$n_o = \frac{P_{out}}{m_f\ \ n_b(H_u + h_f - h_{fs})} \qquad (3)$$

$$SHC = \frac{1}{n_b}\ \times\ 3600 \qquad \text{KJ/KW.hr} \qquad (4)$$

$$SFC = \frac{m_f}{P_{out}}\ \times\ 3600 \qquad \text{KJ/KW.hr} \qquad (5)$$

$$n_c = \frac{T_1}{(T_2 - T_1)}\,.\,\left(\frac{P_2}{P_1}\right)^{E_a} - 1\ , \qquad E_a = \frac{K_a - 1}{K_a} \qquad (6)$$

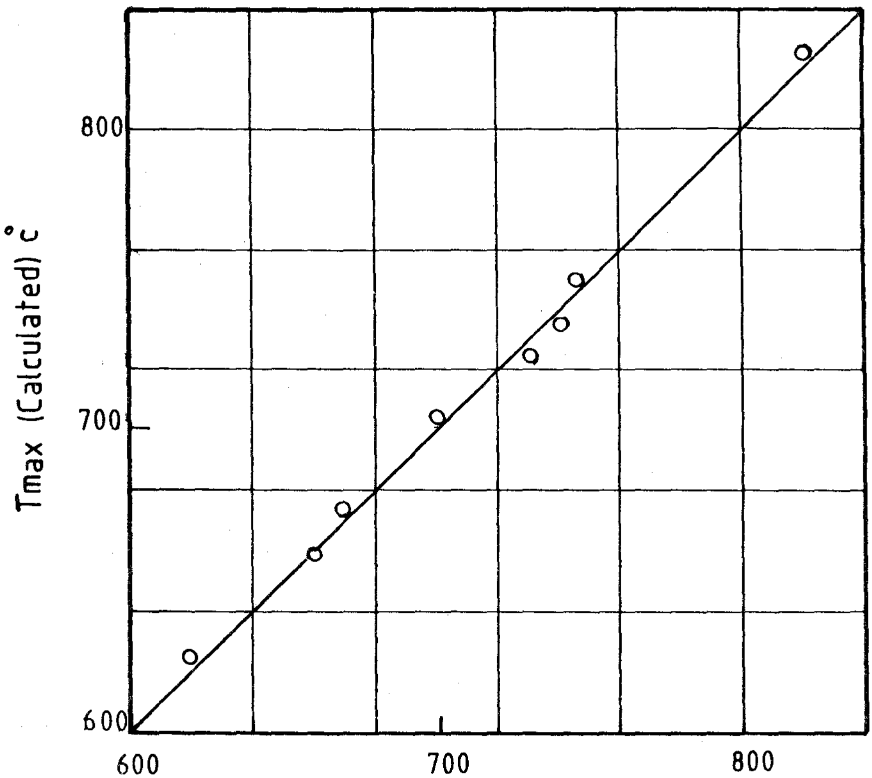


Fig. 5: Calculated Turbine Inlet Temperature Versus Measured Temperature.

Figure (6) presents a summary of variations in ambient air temperature, pressure and relative humidity in Qatar during the year 1984, (6). The total number of observations used in each histogram is 8784 (24 hours x 366 days).

The air temperature varies from 8°C to 47°C with mean temperature of 27°C. Pressure varies from 0.990 bar to 1.026 bar with mean pressure of 1.010 bar. Relative humidity varies from 5% to 100% with a mean value of 55%. Of these variables the temperature is the most influential factor and therefore it is given the prime consideration. With such variation in T_a, the engine mass flow varies by about 14% from its design value (4). Output power and overall efficiency at 48°C air temperature decrease to 65% and 90%, respectively of their values at 8°C air temperature. These figures may vary from one gas turbine engine to another but the general trend of variation of mass flow, output power and overall efficiency with ambient temperature remains the same.

Results of performance analysis of the turbo-generator unit at different air temperatures are shown in Figures (7), (8) and (9). Figure (7) shows overall efficiency n_o as a function of ambient temperature and output power. In this figure, it is extremely important to plot the overall efficiency curves, for fixed power setting, at same maximum cycle temperature since the latter has the highest impact on overall efficiency. With this factor taken into consideration in producing Figure (7), a reduction of efficiency, with temperature increase, of approximately 2% is noted at higher values of output power (44 MW), while slight change in efficiency is noted at low output power (20 MW). Figures (8) and (9) show variation of compressor isentropic efficiency n_c and turbine isentropic efficiency n_t as functions of both ambient temperature and power output. The increase of n_c with increase of T_a is explained in the next section.

7. EFFECT OF LOADING ON PERFORMANCE OF GAS-GENERATOR COMBINATION

Gas turbine engine performance is at its best when the engine is at its design values. In the present case the engine is designed to operate mainly in a base-load mode. Therefore, a degradation of performance is expected at part load. Figure (7) illustrates this fact by showing the remarkable effect of reducing load on overall efficiency.

It is worthwhile discussing the effect of both load variation and ambient

Fig. 6: Frequency Histogram of Annual Population of Hourly Data for Selected Meteorological Elements (6).

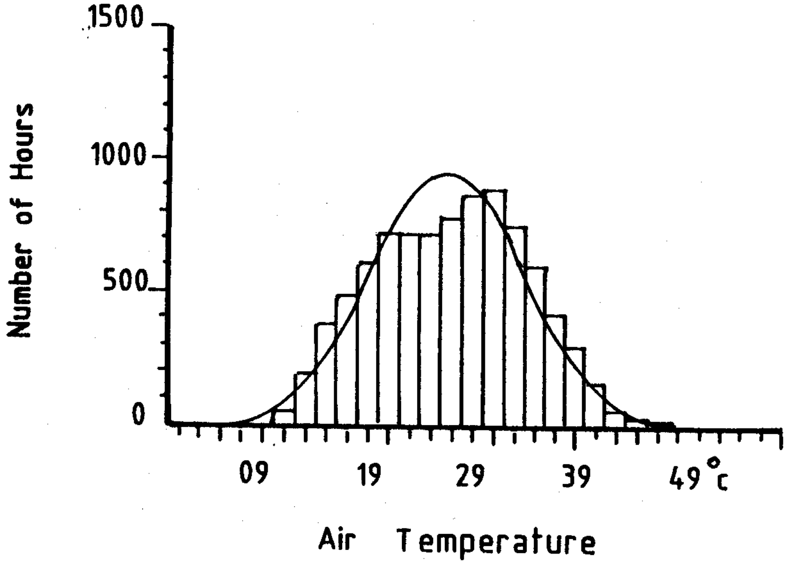


Figure (6) a

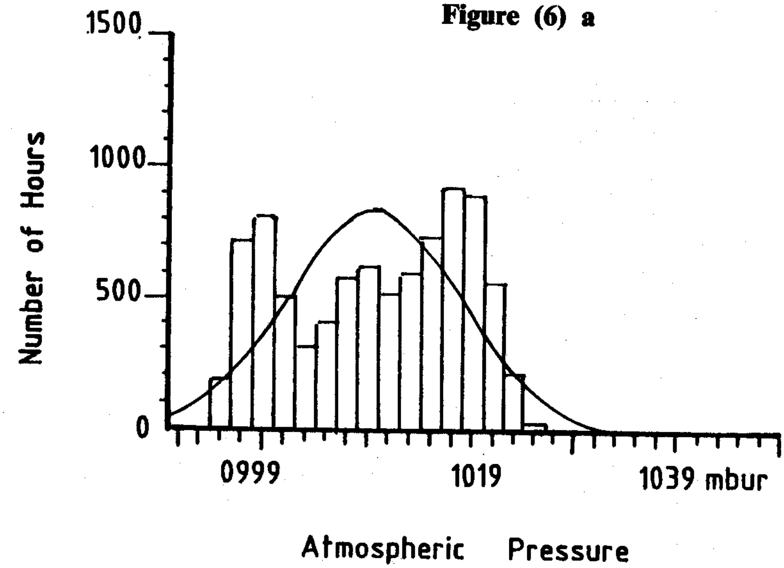


Figure (6) b

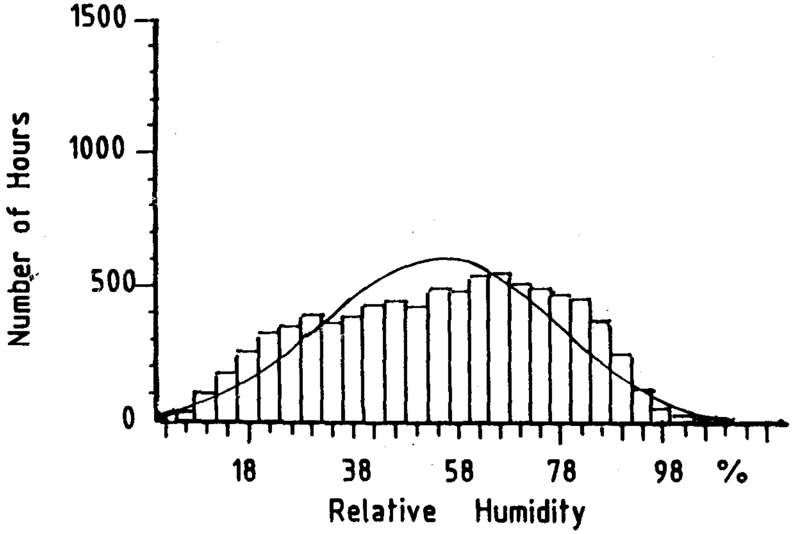


Figure (6) c

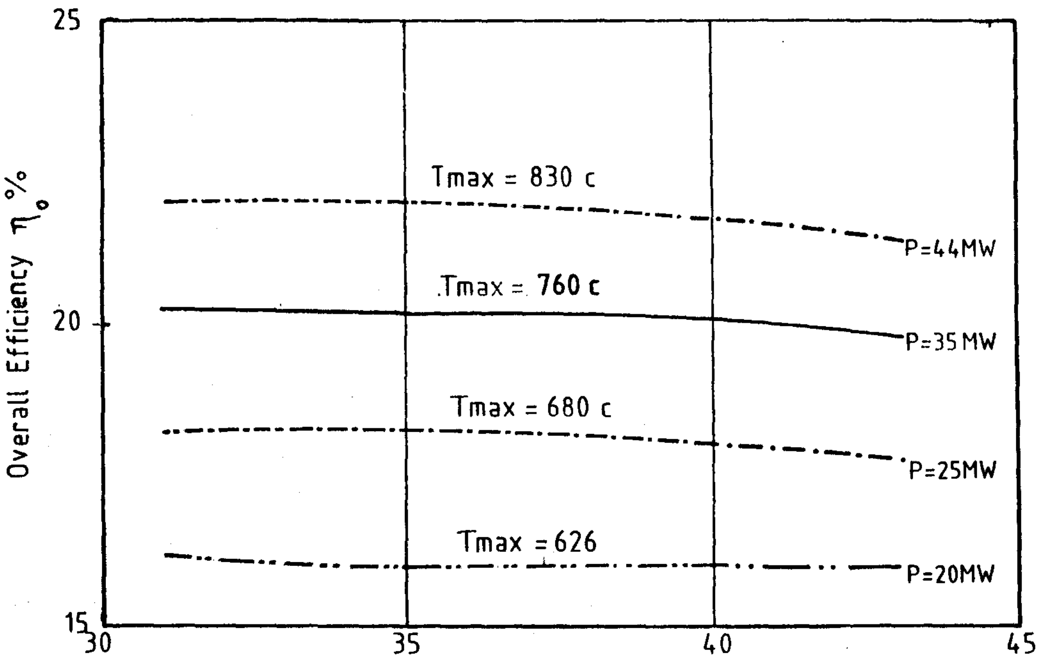


Fig. 7: Effect of Ambient Temperature on Overall Efficiency of Turbo Generator Unit.

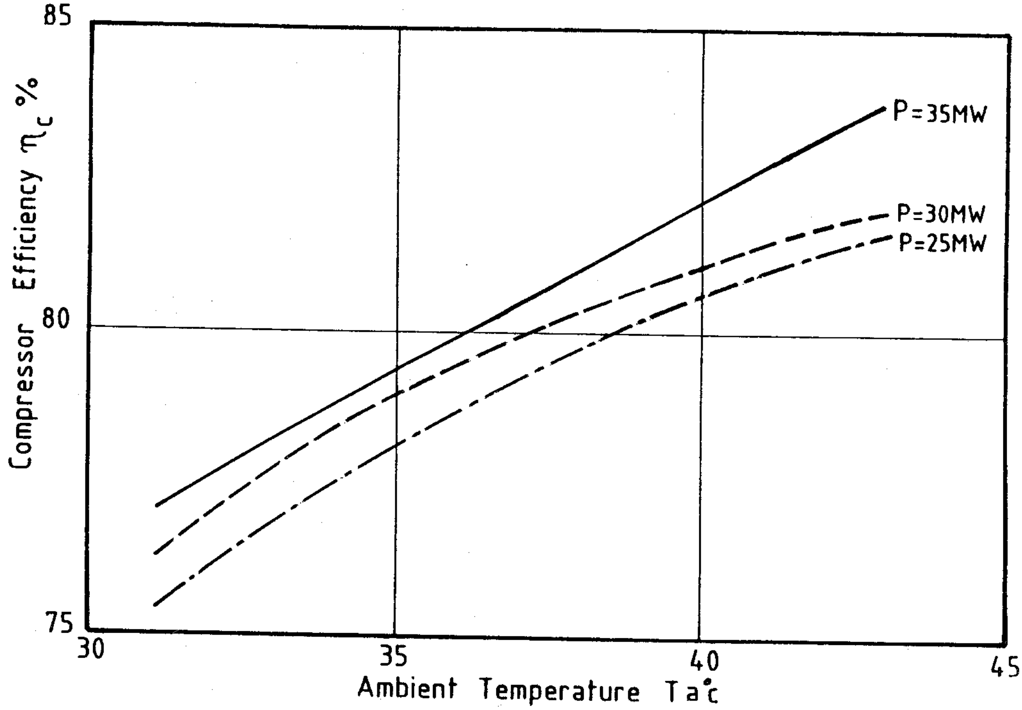


Fig. 8: Variation of Compressor Isentropic Efficiency with Output Power and Ambient Temperature.

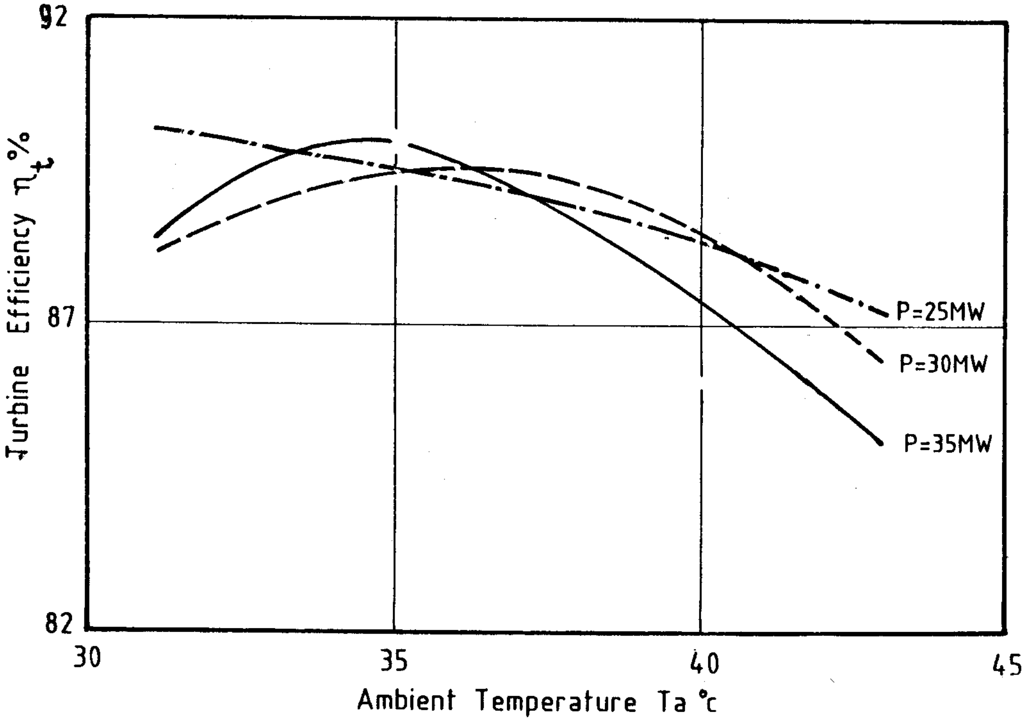


Fig. 9: Variation of Turbine Isentropic Efficiency with Output Power and Ambient Temperature.

temperature variation on the behaviour of a single-shaft gas turbine engine driving a generator with the aid of the hypothetical compressor characteristics shown in Figure (10). For fixed generator rotational speed, N and for fixed compressor inlet temperature, T_1, the non-dimensional compressor speed $N/\sqrt{\Theta}$ maintains its value irrespective of load variation (Θ being equal to T_1/reference temperature). Thus the operatingne coincides with a corresponding $N/\sqrt{\Theta}$ line c-a-b-, where points c, a and b represent no load, base load and peak load conditions. Figure (10) also shows the contours of constant compressor efficiency n_c with n_c increasing towards design values. On this chart are superimposed lines of constant turbine inlet temperature (or maximum cycle temperature) $T_{max}{}^1$, this is shown to increase towards the surge line and also to increase with increasing compressor pressure ratio (P_2/P_1).

The effect of part-loading is illustrated in Figure (11)-A. As load decreases, the operating point moves from point a to point a^1 thus resulting in reduction of compressor efficiency $(n^1{}_c < n_c)$ and possibly in small increase in mass flow. More important is the evolved reduction in pressure ratio, (P_2/P_1) and maximum cycle temperature, $(T^1{}_{max} < T_{max})$. The reduction of these two parameters results in substantial drop in overall efficiency.

A rise of ambient temperature increases 0, this causing in effect a shift in the whole operating line to the left as indicated by the dotted line in Figure (11)-B. A large reduction of mass flow and pressure ratio is expected with consequential substantial drop of output power. Maintaining same maximum cycle temperature at increased ambient temperatures minimizes, however, the drop of overall efficiency, Figure (7). Increase of compressor efficiency with increase of T_a as noted in Figure (8) can be justified by observing the compressor characteristics displayed in Figure (11). A rise of T_a moves point (a^1), Figure (11)-A, to the left where higher values of compressor efficiency may be obtained.

A combination of increased ambient temperature along with reduced load is the worst that can happen as far as thermodynamic performance is concerned. Point (a) in Figure (10) moves in such case to point (d), thus resulting in reduction of compressor efficiency, overall efficiency and mass flow rate. The effect of load decrease, (with the consequential decrease in T_{max}), on overall efficiency exceeds however, the effect of ambient temperature increase. This can be observed clearly in Figure (12).

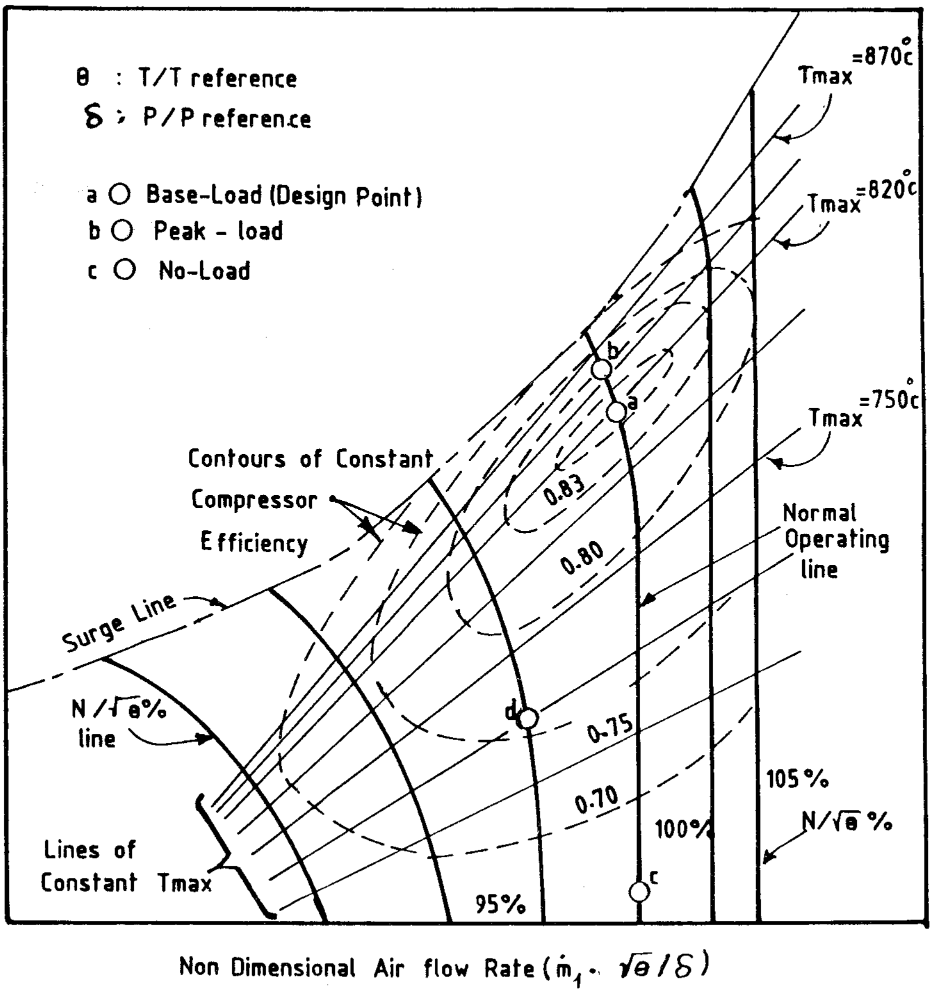
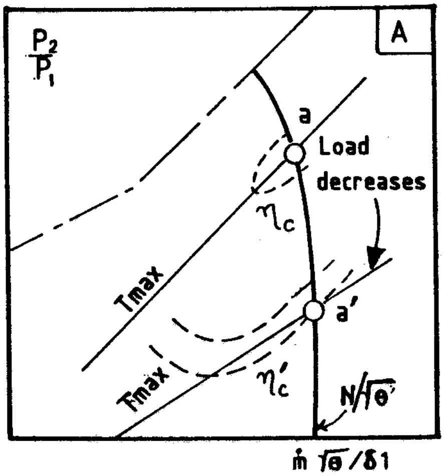


Fig. 10: Compressor Characteristics (Schematic).



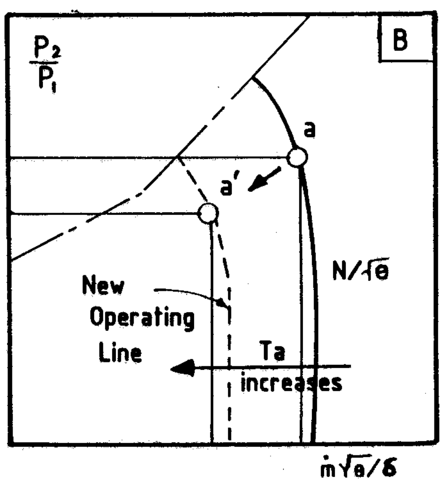


Fig. 11: Effect of Both Part Loading and Ambient Temperature on Turbo-Generator Performance.

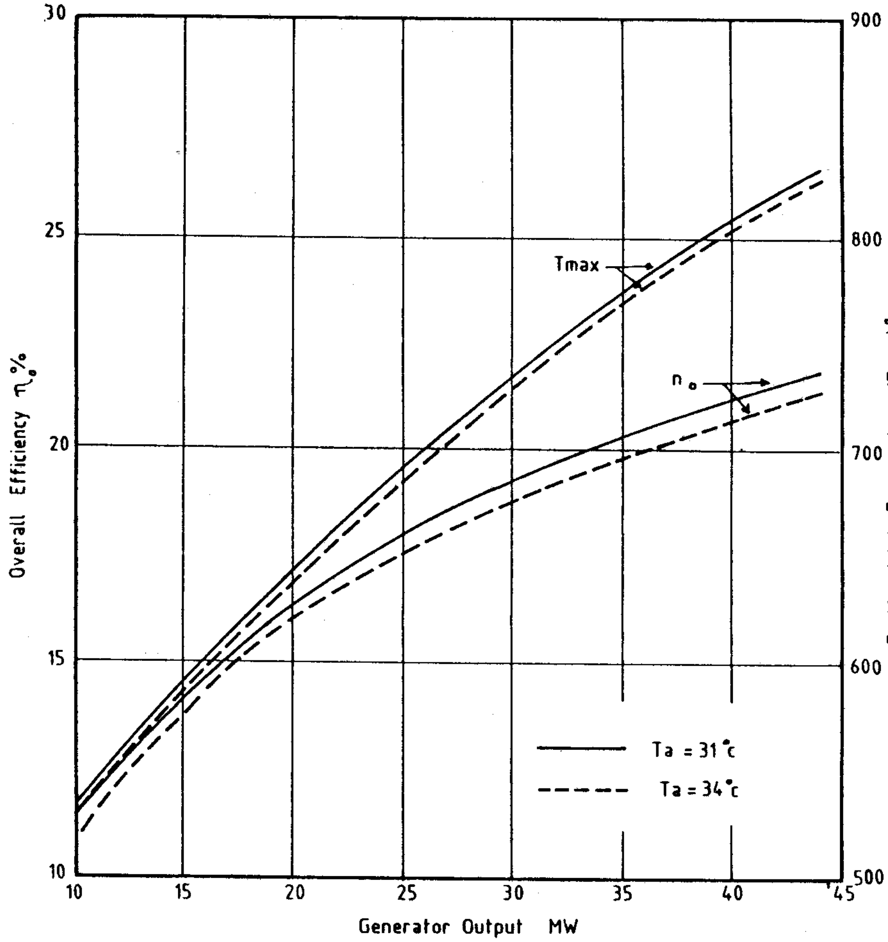


Fig. 12: Variation of Both Maximum Cycle Temperature and Overall Efficiency of the Turbo-Generator Unit with Power Output and Ambient Temperature.

8. SPECIFIC HEAT CONSUMPTION AND SPECIFIC FUEL CONSUMPTION

Specific heat consumption SHC, as defined by equation (4), is a measure of efficient utilization of heat energy for producing electrical energy, since SHC is inversely proportional to overall efficiency. SHC variation with variations of output power and ambient temperature is shown in Figure (13).

Manufactures of gas turbines, such as KWU, normally provide an estimate of SHC under different operating conditions, either through their technical publications or through some on-site performance tests. The shaded area in Figure (13) represents the manufacturer's guaranteed specific heat consumption, (7) corrected for the considered range of ambient temperatures. Comparison between actual SHC and manufacturer's guaranteed SHC shows how much additional energy need to be consumed, in order to produce same power output due to performance degradation. This additional energy may be considered as waste if we are considering the gas turbine-generator unit alone. On considering the gas turbine-generator-waste heat boiler-distiller as a whole, the additional heat energy is not, however, wasted since it is utilized in producing useful steam.

Specific fuel consumption SFC can be calculated from equation (5) in kilograms of fuel per (KW.hr). Since the primary fuel at Ras Abu-Fontas Power and Water Station is natural gas, mostly Khuff Gas, then it is more convenient to express SFC in normal m^3 per (MW.hr). The variation of SFC with generator output and ambient temperature is shown in Figure (13), from which it can be seen that operating the turbo-generator unit at lower than 50% of its capacity requires some 50 to 100 percent extra amount of fuel per unit power produced.

9. DISCUSSION OF ENERGY CONSUMPTION IN LIGHT OF TURBO-GENERATOR PERFORMANCE

According to the 1984 Electricity Statistical Report (8), Ras Abu Fontas Power and Water Station contribution in state generated electrical energy was 2,121,588 MW.hr. This is approximately 60% of the total electrical energy generated in Qatar in 1984. To generate this amount of electrical energy, most of the 60,710 million standard cubic feet (1,719 million Nm^3) of natural gas, consumed by the power station, was burned to drive the gas turbine-generator units. When dealing with such huge amounts of fuel, every effort should be made to ensure efficient utilization thereof. Sources of heat loss should be studied carefully in the light of actual performance

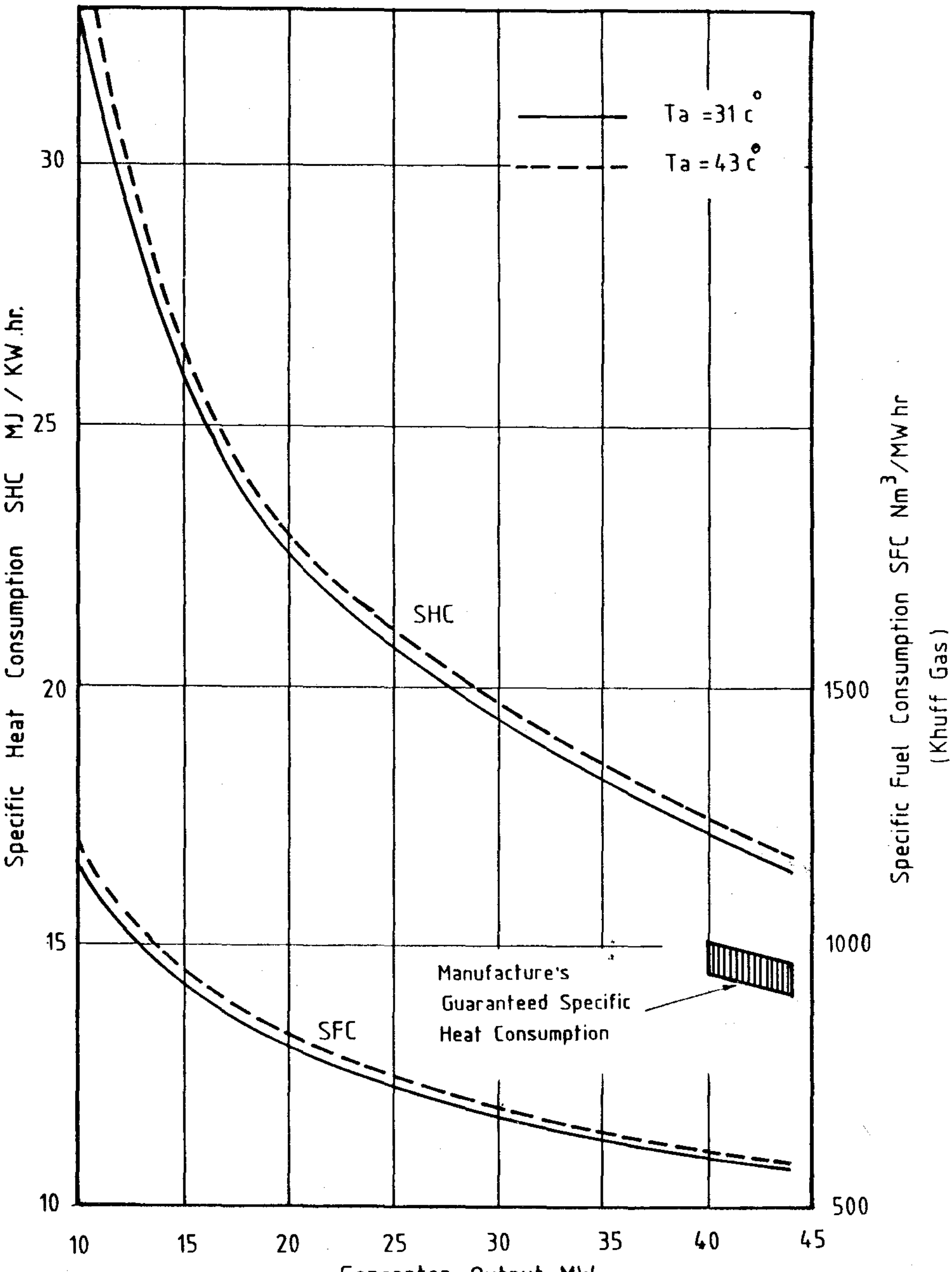


Fig. 13: Variation of Specific Heat Consumption SHC and Specific Fuel Consumption SFC with Power Output and Ambient Temperature.

characteristics.

Although recognizing that the excess heat energy supplied to gas turbines may be utilized in the waste heat boilers, there is still reason to believe that an increase in specific heat consumption should be seriously considered. For one reason, electricity is not only the primary product of the plant, but it is also the most expensive form of energy: the efficient conversion of heat into electricity implies successful operation of the plant. Secondly SHC is a measure of gas turbine performance, and unusual values of SHC indicate some sort of engine problems that should be investigated.

Excessive specific heat consumption can be classified according to its causes into two major parts. The first part is due to engine degradation of performance. This means that with all variables of the cycle set according to manufacturer's recommendations, the engine fails to achieve the expected overall efficiency. Therefore an appropriate remedial action should be taken based on the results of engine diagnosis and inspection. Excessive specific heat consumption due to engine degradation is shown in Figure (14). In the analysis of turbo-generator unit 2, this part of excess energy is approximately 1.7 MJ/KW. hr at full load.

The second part of excessive specific heat consumption is due to engine part loading as indicated in Figure (14), this depending on general load demands, load schedule of the plant, number of units in operation at a time as well as few other factors.

Simple calculations of the annual plant factor, based on the rated capacity of the plant and the total electrical energy generated during 1984 indicate a value of 0.4. Although this value appears to be low, it can be justified if the electrical power system of Qatar as a whole is taken into consideration.

The load on an electrical power system is apportioned among the power plants that comprise the system. The generating capability of the system must be sufficient to meet peak or maximum demand of the year, normally during the hot summer, with some reserve generation to provide for possible failure of equipment and outage for regular maintenance. The spinning reserve consists of those machines synchronized with the electrical network and ready for loading. In the electrical power system of Qatar, all plants except Ras Abu-Fontas operate as base load plants. On the other hand, Ras Abu-Fontas is the station which adjusts the total output power of the system to meet the demand. While six turbo-generator units may be sufficient to provide the required power in winter time, however, the twelve units must be in

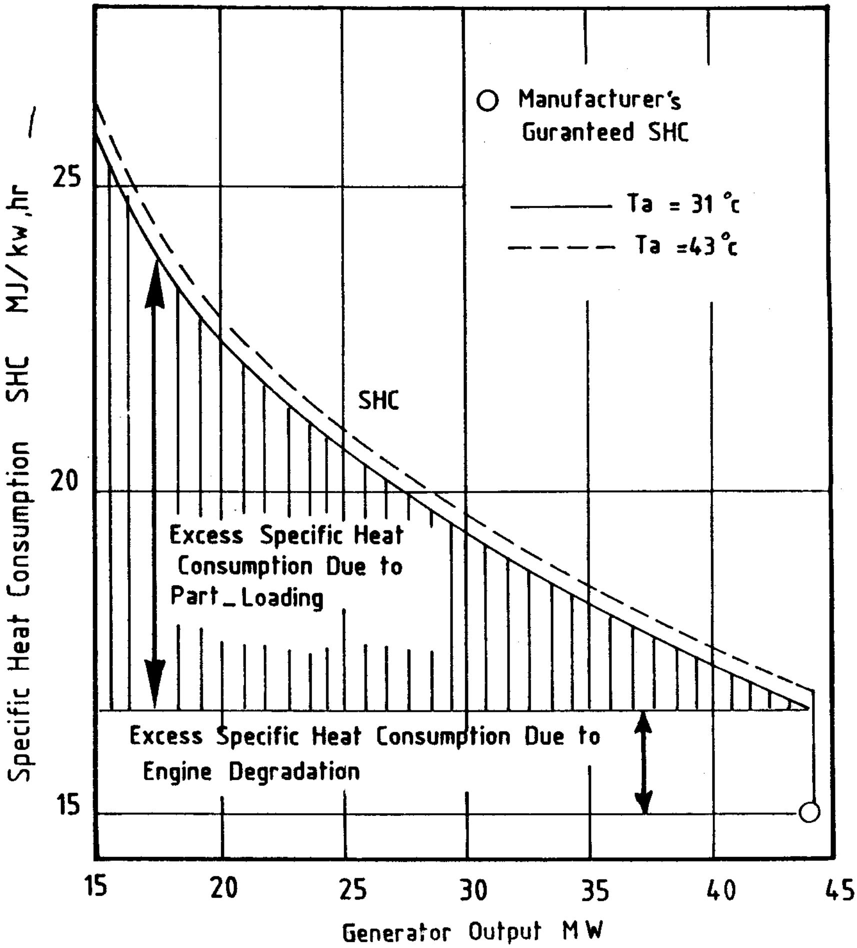


Fig. 14: Waste Energy Due to Part-Loading and Engine Degradation.

operation during peak load conditions in summer time.

The system spinning reserve of 120 MW minimum is also assigned to Ras Abu-Fontas power station. This spinning reserve may well exceed 180 MW for a good portion of the year due to sharp load variations. For example, there is an industrial consumer who has a 60 MW furnace which is turned on and off twelve times a day!. To maintain he minimum spinning reserve, two turbo-generator units need to be started up and shut down every two hours; this is not acceptable due to excessive maintenance cost.

It is worth mentioning that among the 622 MW, which is the installed capacity of Ras Abu-Fontas station, there are 30 MW allocated for black start of the electrical power system and are not contributing to the load capability of the plant. With these factors in mind, an annual plant factor of 0.4 seems to be quite reasonable. Consequently, turbo-generator units are working at almost 50% load for most of their operating hours. For the purpose of this study, it may be interesting to see how much excess heat is needed per unit power generated due to part-loading. Figure (14) shows excess SHC due to part loading. Figure (15) presents, in a simple way, the economic side of the problem. Based on the government-highly-subsidized fuel prices (0.8 QR per 1000 St. Cu. ft.), simple calculations result in a yearly excess fuel for generating electricity, due to both part-loading and engine degradation of unit 2 alone, of the value of 1.3 million QR. In the era when most natural gas was to be flared, the previous conclusion may seem to be of secondary importance. But nowadays, with most of natural gas produced is being utilized (10), and furthermore the prospects of exporting it are seriouslynsidered, we should not ignore such an amount of excess fuel expended in the process of generating power, even though it might be used either totally or partially, in water desalination, especially when international gas prices are taken into account.

The performance of the waste heat boiler-desalination units and the extent to which we can successfully utilize additional heat associated with excess specific fuel consumption are thought to be an important area for future study. It should be pointed out however, that the present study is mainly concerned with gas turbine-generator unit alone.

10. CONCLUSION

A discussion of energy consumption of 44 MW gas-fired turbo-generator unit led to

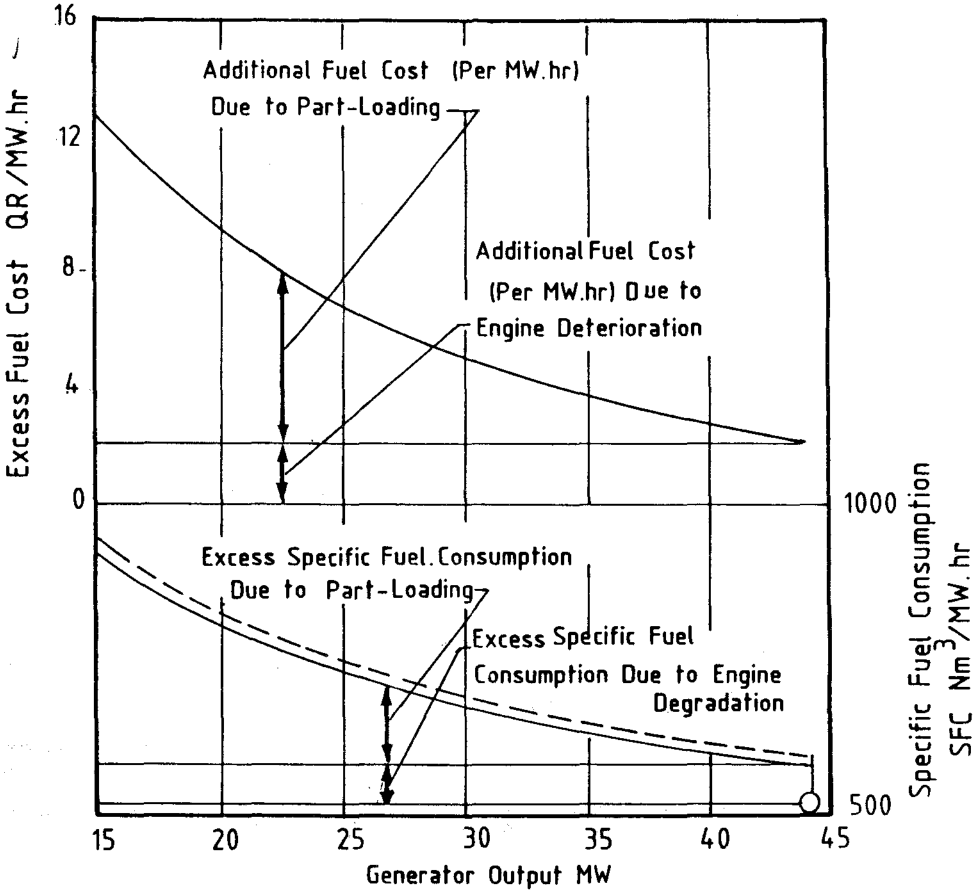


Fig. 15: Excess Fuel Cost and SFC Due to Part-Loading and Engine Degradation.

the conclusion that excessive specific fuel consumption, due to performance degradation and part load operation of the unit, costs some 1.3 million Qatari Riyals annually.

A rise of ambient temperature slightly increases specific heat consumption; it reduces however, power output substantially.

The method of analysis is explained and relevant results validated by comparison-with selected measured variables.

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