TECHNO-ECONOMIC EVALUATION OF MECHANICAL CHILLER FOR POWER RECOVERY IN A GAS TURBINE UNIT IN NIGERIA

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ABSTRACT

The rated power output of a gas turbine is not realised when it operates at ambient conditions different from the rating conditions. Probably the most significant ambient condition in this regard is temperature with substantial losses in gas turbine performance attributable to it hence the need for gas turbine inlet air cooling in hot climes. In this paper, operation data from a gas turbine plant in Nigeria has been used to determine the variation of gas turbine power output with ambient temperature. Also, a techno-economic analysis of a suitable mechanical chiller for reducing intake air temperature has been carried out. It is found that for every 1^{0}C increment in ambient air temperature above ISO standard temperature, gas turbine power output drops by 0.94%. Retrofitting the gas turbine with a mechanical chiller can potentially improve power output up to 20% with substantial financial gains including the capability of the system to pay for itself in under three years. Further, the upper limit of capital expenditure for the mechanical chiller installation to be profitable was determined. There is, therefore, an economically viable case for expanding the present power capacity of gas turbine plants in Nigeria through mechanical chilling of inlet air.

Keywords: Gas turbine performance, ambient temperature, inlet air cooling, economics.

INTRODUCTION

There is no gainsaying that energy generation and utilisation is both the spur and reward of national growth and development. This reliance on energy for economic growth is true globally as it is in Nigeria [1, 2]. For continued economic expansion then, the world's energy needs keeps on increasing with the more developed nations consuming more than their counterparts that are not quite as developed. However, the annual percent increment in energy consumption, particularly projections into the future, does not follow the same pattern. It has been predicted by [3] that, between 2012 and 2040, the African continent will witness an annual 2.6% increment in energy consumption – six times more than the projection for the US and the second largest estimated energy demand increment worldwide in the same time. In Nigeria, one of the top economies in Africa, the current demand for energy far outstrips supply. Less than 60% of electrical power demand in Nigeria has ever been generated at any time [4] with thermal power plants accounting for 86% of generated power [5]. In this regard, the gas turbine (GT) has gained popularity largely because of its relatively low initial capital cost, flexibility, reliability and compactness.

In recent times, those in charge of power generating utilities in the country are striving towards better profits by clamping down on power theft – the criminal practice of stealing electrical power – that has been identified as one of the major challenges in power distribution in the country [6]. However, attention must likewise be paid to improving efficiency and performance in the generation stage. This, for GT-powered stations, would

involve investigating power losses from the thermodynamic viewpoint. In this regard, the GT air inlet temperature which is fixed by the ambient air temperature is known to have dominating influence on plant efficiency and performance particularly in hotter climes [7-13]. A number of studies have therefore been conducted to measure the impact of ambient air temperature on GT performance. For instance, [14] reported a loss of about 0.6% in rated power of heavy duty GT units located in Dubai for every 1^0C rise in ambient temperature. A 0.74% decrement in power output was reported by [15] for every 1^0C rise in ambient temperature for Alstom frame-5 GT machines operating in Chabahar, southern Iran where the average temperature is 27^0C. The effect of ambient air temperature on gas turbine power output is probably best summed up by [13] namely that 0.6 – 1.0% power loss is encountered for every degree rise in GT intake air temperature above the ISO standard condition (15^0C).
The reason for the loss in power output for GT operating in hot environment is well known. Gas turbines are constant volume engines for which shaft power is proportional to inlet air mass flow rate so that a reduction in air density at high temperatures reduces the specific volume of air in the engine resulting in power loss. To recover power lost this way, several GT inlet air cooling (GTIAC) techniques are available at present and they can be classed either as water evaporation systems or heat transfer systems. The former includes evaporative cooling and inlet fogging while the latter includes mechanical chillers, absorption chillers and cold thermal storage systems.

A review and comparison of the different GTIAC methods and how they affect GT performance profiles have been conducted by [10, 16, 17]. Other papers focused on individual cooling techniques such as fogging [18], evaporative cooling [19], mechanical chiller [20, 21], absorption chiller [15, 22] and thermal storage [23]. GT performance improvement offered by GTIAC has been shown to be attractive not only from thermodynamic assessments but also from the economic standpoint in different locations [15, 19-30].

The ambient air conditions of every location is different and the decision to embark on a capital intensive venture like GTIAC for a particular location requires a good knowledge of turbine performance in the area. In Nigeria, there is not only a dearth of literature on the subject of power losses as a result of ambient conditions but also there is hardly any studies on GTIAC for the region. [19] is possibly the only published study in that regard and in it the potential benefit of evaporative cooling in gas turbines in Nigeria is evaluated. However, the cooling effect obtainable from evaporative cooling systems is limited by the air wet bulb temperature which is mainly a function of air relative humidity. Therefore, evaporative cooling despite being a relatively low cost venture will suffer from poor efficiency for long periods in the Nigeria climate which is predominantly high in humidity. Consequently, it is important to evaluate the economics of installing mechanical chillers (whose performance is independent of air humidity) in gas turbines in Nigeria. To achieve this aim, the variation of electrical power as a result of ambient temperature changes in the location of the gas turbine has been empirically modelled using real operation data. This knowledge was used in estimating the power gains achievable by inlet air cooling via mechanical chilling thereby permitting an economic analysis of such modification to the gas turbine. As design choices affect project profitability, a parametric analysis by independently varying fuel and electricity costs as well as overall capital expenditure, has also been carried out.

PLANT AND SITE DESCRIPTION

This study was carried out with data from a GE Frame 9 engine that is part of the Gbarain National Integrated Power Project (NIPP) power plant situated at Gbarain, in the southern

ISSN 2056-5860

Nigeria state of Bayelsa. The turbine works on a simple cycle mode (illustrated in Fig. 1) and is composed of an axial compressor with 17 stages and a maximum pressure ratio of 12.6, with an exhaust gas temperature of about 537 ^{0}C and a three-stage axial turbine. It has a simple cycle thermal efficiency of 37% and an inlet air mass flow rate of about 407 kg/s. The Gbarain power station is fuelled with natural gas provided by Shell Petroleum Development Company (SPDC) through its Gbarain-Ubie integrated oil and gas plant in Bayelsa State. One of two units in the Gbarain power station (each rated 123 MW) started operations in 2016 and it was from this that data was collected for a period of six months. The six months for which data were collected was June, July, September, October, December (all in 2016) and January 2017. Operation data, including ambient temperature and power output, which are recorded on an hourly interval for each day was collected. In this study, a filtrate of the thousands of data collected was utilised. Anomalous data which may have been due to part-load operation occasioned by maintenance or grid requirement were discarded.

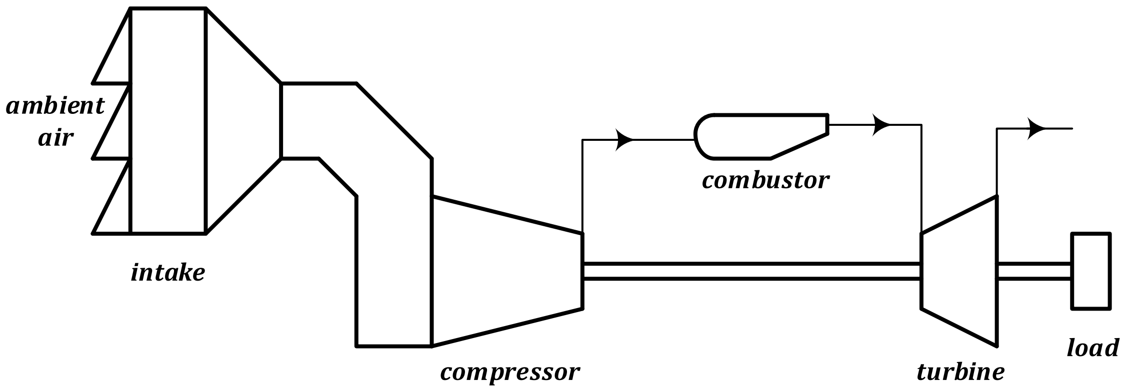

Fig. 1. Schematic of simple cycle gas turbine engine

Bayelsa, like all of southern Nigeria, typically has two seasons: the wet and dry seasons. In the thick of the wet season, rain volumes are high and at the height of the dry season, there is no rainfall. However, there is not only an overlapping of the seasons but also brief periods where there is interference between seasons. For instance, after a long rainy season from March to July, there is a short spell of dry season in August lasting for 3-4 weeks. Thereafter, a short rainy reason, in which the rains are not as heavy as in the long rainy season, follows from September to mid-October. The long dry season begins in late October and carries on till February. The climate of southern Nigeria is such that the mean monthly temperature range is within 24.4^{0}C – 28.9^{0}C and relative humidity values range from 55.6% to 84.2% [31].

MECHANICAL CHILLER THEORY

A mechanical chiller works on the well-known vapour compression cycle: gaseous refrigerant from the evaporator is compressed and then liquefied in a condenser by exchanging heat with cooling water. Thereafter, its high pressure is reduced through an expansion valve by expanding into an evaporator where the low pressure liquefied refrigerant converts into gas by removing heat from the chilled water in the GT inlet air cooling coils. Heat collected from the chilled water together with that due to compression is rejected to the atmosphere in the cooling tower. Therefore, the main components of a chilling system of this kind are: an air cooling coil, a centrifugal chiller, a cooling tower, a chilled water pump (for circulating chilled water from/to chiller to coils), a cooling water pump (for circulating cooling water from/to the chiller and cooling tower) and electrical facilities (to drive chiller compressor, tower fan and pumps).

Cooling Load Estimation

The cooling load capacity Q_{cl} of the mechanical chiller in order to achieve air temperature reduction is equal to the sum of sensible heat loss from the air and the latent heat of condensation of associated water vapour which the air contains.

$$Q_{cl} = \dot{m}_{air}(q_{sensible} + q_{latent}) \tag{8}$$

where h_a, h_c and h_d are the air enthalpy values at points a, c and d respectively as shown in Fig. 2 which is an illustration of the path taken by the inlet air on a psychrometric chart as it changes from state a (the ambient air condition) to state c, the desired cooled state along the path $a - b - c$ plotted in heavy lines. The sensible cooling load, $q_{sensible}$ is shown as $d - c$ and is equal to $h_d - h_c$ while the latent cooling requirement, q_{latent} is shown as $a - d$ and is the value of $h_a - h_d$; $\dot{m}_{air}$ is the inlet air mass flow rate.

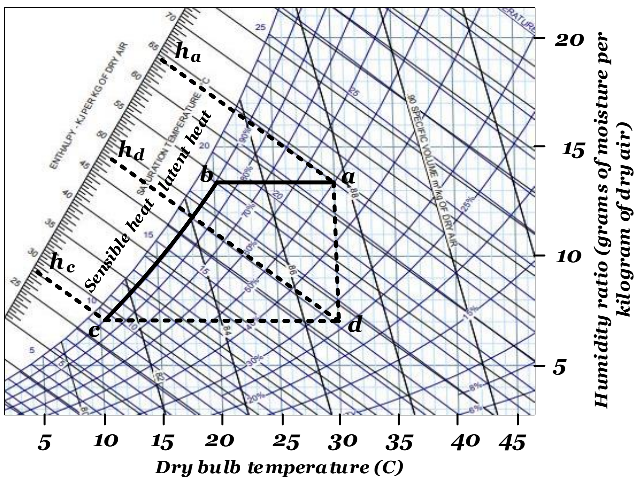

Fig. 2. Illustration of cooling process on psychrometric chart

Chiller Design

The electric power consumption of the chiller $\dot{W}_{chiller}$ is obtained using the relation:

$$\dot{W}_{chiller} = Q_{cl}/COP_{chiller} \tag{9}$$

where $COP_{chiller}$ refers to the coefficient of performance of the mechanical chiller for which a value of 6.5 is applicable to modern chillers [22, 32]. Other chiller design considerations are chilled and cooling water entry and exit temperature as well as their respective flow rates. Standard values for these parameters have been extracted from [33] and provided in Table 1 where 'gpm' and 'RT' are US gallons per minute and refrigeration ton respectively. 1US RT refers to a refrigeration capacity that is equivalent to 3.52 kW.

Table 1. Standard conditions for mechanical chillers [22, 32]

Parameter	Value
Chilled water entry/exit temperature (^{0}C)	12/7
Cooling water entry/exit temperature (^{0}C)	29/34
Chilled water flow rate	2.4 gpm/RT
Cooling water flow rate	3 gpm/RT

The evaporator and condenser of the chiller are shell and tube heat exchangers, in which refrigerant goes in the shell side and water in the tube side. The pumping power consumption for the evaporator, $\dot{W}_{evap\ pump}$ and condenser, $\dot{W}_{cond\ pump}$ were calculated using equations (10) and (11) as in [28] where the familiar symbols have their usual meanings and $\dot{V}_w$ represents the volumetric flow rate in m^3/s with the subscript w referring to water wherever it appears; ΔH_{ct} the difference between the height of cooling tower inlet and outlet connections.

$$\dot{W}_{evap\ pump} = \dot{V}_w \Delta P_{evap}/\eta_{pump} \quad (10)$$

$$\dot{W}_{cond\ pump} = \dot{V}_w(\rho_w g \Delta H_{ct} + \Delta P_{cond})/\eta_{pump} \quad (11)$$

The pressure in either the evaporator or condenser is essentially the saturation pressure for the given temperature and can be found on pressure-temperature charts. For R-134a, a common refrigerant for this sort of processes, the condenser pressure at 34^0C is 0.37 MPa and the evaporator pressure at 7^0C is 0.86 MPa. With a known chiller compressor outlet pressure which is typically 1 MPa [32], the pressure changes in condenser and evaporator ΔP_{cond} and ΔP_{evap} can be calculated as can the pump sizes assuming a pump efficiency, η_{pump} of 0.9.

The capital cost of pumps according to [28] is given by equation (12) where $\dot{W}_{pump}$ is in kW and k depends on the electric power consumption of the pump. For pump power in the range of 0.02 – 0.3 kW, $k = 0.25$, while for a range of 0.3 – 20 kW, $k = 0.45$ and for a power range of 20-200 kW, $k = 0.84$.

$$C_{pump} = 308.9 \times \dot{W}_{pump}{}^{k} \quad (12)$$

Cooling Tower Design

The cooling tower may be designed in line with [34]. However, the purchased equipment cost for a cooling tower, C_{ct} may be estimated using [28]:

$$C_{ct} = a_1 \dot{m}_w{}^{a_2} \times 10^{a_3 A.R + a_4 A + a_5 R + a_6} \quad (13)$$

This requires a knowledge of only the water flow rate the tower handles, $\dot{m}_w$. All terms in the equation are constants except A, the temperature difference between water outlet temperature and ambient wet bulb temperature and R, the temperature range in cooling tower. The values of the constants are: $a_1 = 3950.9, a_2 = 0.58729, a_3 = -0.0032091, a_4 = -0.026719, a_5 = 0.043654$ and $a_6 = -0.1026$.

The capital cost of a cooling tower fan including the electric motor cost according to [25] is given by equation (14) in which p and q are constants with values of 21.5644 and 20.1554 respectively and $\dot{W}_{fan}$ is the electric power consumed by the fan in kW.

$$C_{fan} = \left(p + q \times \dot{W}_{fan}{}^{0.5}\right)^2 \quad (14)$$

The fan power consumption, $\dot{W}_{fan}$ is estimated by [35] to be 1.05% of the nominal cooling tower capacity $\left(\dot{m}_w C_{p,water} \Delta\theta\right)$ with $\Delta\theta$ being the temperature difference between cooling tower water inlet and outlet.

The operation of the cooling fan requires that water be introduced to make up for that lost due to evaporation and blowdown. From [36], the evaporation loss, $Evap_{loss}$ is given by $(0.00085 \times 1.8 \times \dot{m}_w \Delta\theta)$ whereas the blowdown loss is $Evap_{loss}/(COC - 1)$. The cycles of concentration, COC, is the ratio of dissolved solids in circulating water to the dissolved solids in make-up water and is often given a value of about 3. The make-up water requirement is the sum of the evaporation loss and blowdown loss. The cost of water for meeting this requirement is given by:

$$Cost\ of\ water = C_w \times t \times m_{w,loss} \times \frac{3600}{1000} \quad (15)$$

where $\dot{m}_{w,loss}$ is the mass flow rate of make-up water (in *kg/s*) required in the cooling tower, t is the annual operating hours and C_w is the price of water per cubic meter.

Electricity cost to meet the power requirements of the system is given by equation (16) where $\dot{W}_{total}$ is the total power requirement of the system which is a sum of the electric power consumption of the chiller, fan and pumps.

$$Cost\ of\ electricity = C_{elect} \times t \times \dot{W}_{total} \quad (16)$$

RESULTS AND DISCUSSION

GT Power Output Variation with Ambient Air Temperature

Fig. 3 shows how power output from the gas turbine unit varies with ambient temperature for the six months in which data was collected. From Fig. 3, an empirical relationship between power output and ambient temperature, T has been established as: $\text{Power (MW)} = -1.1563T + 143.28$. This implies that there is a drop of 1.16 MW in power output as ambient temperature rises by 1^0C in the location of the gas turbine. This corresponds to a 0.94% loss in the rated power of the machine and is in agreement with previous studies like [13].

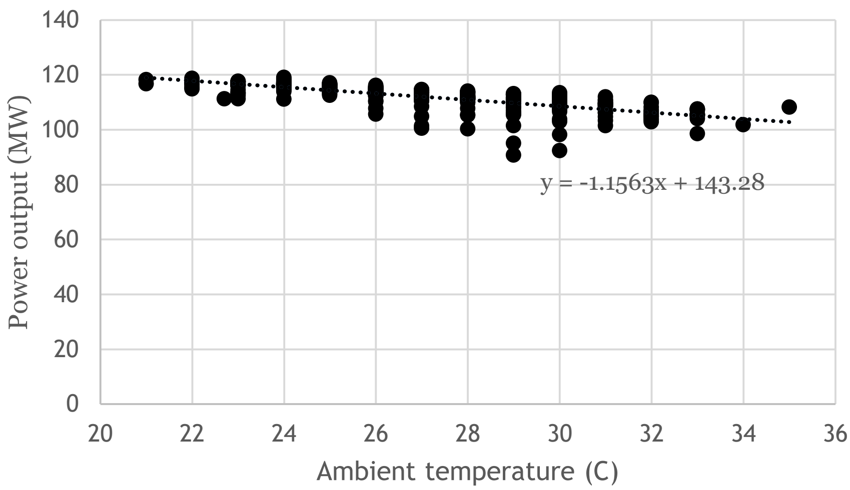

Fig. 3. Power output variation with ambient temperature

Economic Analyses

The overall capital cost of the system based on the above considerations and a capital cost estimation of $231.4/kW for a compression chiller and $195/kW for a heat exchanger [29] is presented in Table 2. The overall capital cost includes the capital investment and the operational and maintenance (O&M) costs assumed to be 2.5% of the overall capital investment [29].

Table 2. Investment cost (k$)

Component	Cost (k$)
Chiller	536.1
Cooling tower	177.6
Pumps	72.8
Cooling tower fan	84.3
Cooling coil	2936.5
O&M	95.2
Overall capital cost	3902.4

For the techno-economic analysis displayed in Table 3, a gas turbine availability of 91% was assumed amounting to 8000 hours of operation per year. The cost of electric power was assumed to be $0.04/kWh, the dollar equivalent of the cost of electricity in Nigeria [4, 19] while the cost of water was set at $1.5/m^3. Natural gas, the fuel burnt in the turbine, was assumed to have a LHV of 50 MJ/kg and to cost $0.1275/kg [37]. The profitability indices considered are the net present value, NPV, internal rate of return, IRR and the payback period. NPV represents the difference between the present value of future cash flows as a result of an investment and the cost of the investment and calculated using:

$$NPV = \sum_{n=0}^{N} \frac{CF_n}{(1+r)^n} \tag{17}$$

In which CF_n represents the net cash flow at year n; r is the rate at which future cash flows are discounted to render them in present value (this has been assumed to be 7%, a typical average cost of capital); IRR is computed as the discount rate r for which the NPV function equals zero thereby offering a means, in addition to the NPV, of justifying the undertaking of a project. The payback period is simply the time period required to recoup the original investment made in the project. Based on the values in Table 3, obtained for an investment return of 10 years, introducing the mechanical chiller inlet air cooling system appears to be a profitable investment.

Table 3. Annual techno-economic analysis of the proposed chiller

Description	Unit	Value
Technical indices		
Annual additional power	GWh	185.0
Annual power consumed	GWh	24.8
Annual additional fuel consumed	million kg/year	36.0
Annual water consumed	million kg/year	267.6
Cost indices		
Annual cost of additional power	million kg/year	7.4
Annual cost of power consumed	million $/year	1.0
Annual cost of added fuel consumed	million $/year	4.6
Annual cost of water consumed	million $/year	0.4
Net annual revenue	million $/year	1.4
Capital cost	million $	3.9
Profitability indices		
Payback period	years	2.75
NPV	million $	5.34
IRR		34%

However, given that the cost of natural gas could vary considerably over any given period of time and also that the cost of fuel (gas) consumed is a significant portion of the annual expenditure on the modified turbine (Table 3), it is imperative to consider the effect of natural gas cost variation on the different profitability indices. Fig. 4, Fig. 5 and Fig. 6 show how natural gas price variation affects the payback period, NPV and IRR respectively. In each of these figures, the effect of the cost of electricity on the measures of profitability have also been shown.

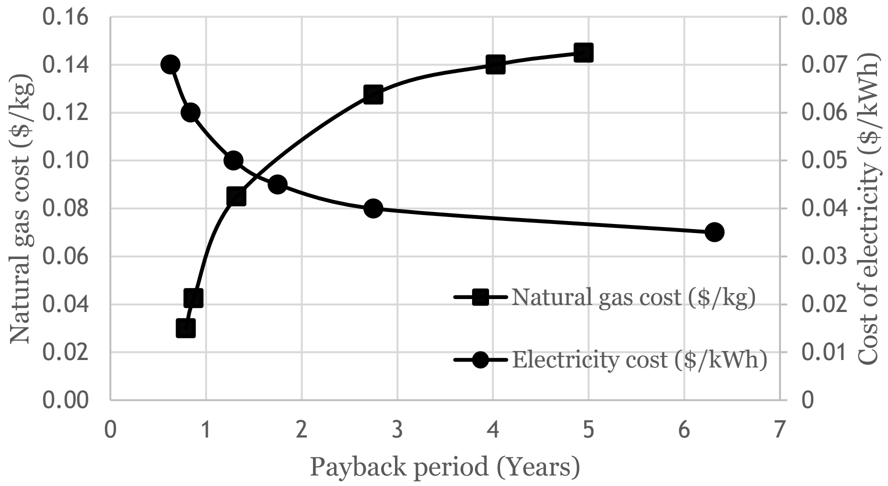

Fig. 4. Variation of payback time with gas and electricity cost

Payback period increases very rapidly at gas costs greater than about $0.13/kg. In fact, from Fig. 4, a 13.7% increase in gas price (from $0.1275/kg) results in 80% increase in payback period. Further, Fig. 4 suggests that it is hardly attractive to go below $0.04/kWh on electricity cost as a mere 12.5% decrement below this value of unit electricity cost causes a 130% increase in payback time.

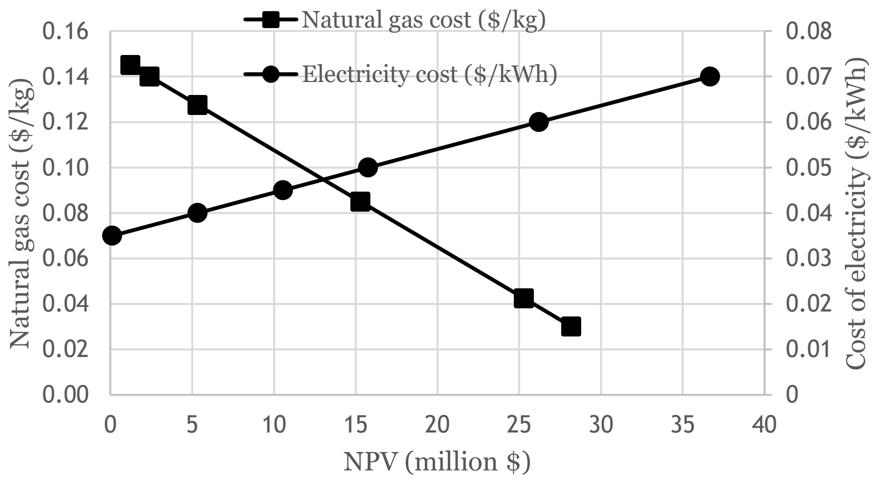

Fig. 5. Variation of NPV with gas and electricity cost

In Fig. 5 and Fig. 6 showing the effect of gas and electricity cost on the two main determinants of project profitability, NPV and IRR, it is clear (from the slopes of the two lines) that the cost of natural gas is more influential than that of electricity. Following on from the comparisons made in the discussion of the payback period, a 13.7% increase in natural gas cost (from $0.1275/kg) reduces NPV and IRR by 0.05% and 1.22% respectively. From Fig. 5 and Fig. 6, these reductions in NPV and IRR values would still guarantee a profitable project. Further, decreasing electricity cost by 12.5% from $0.04/kWh causes an increase in both NPV and IRR by 0.01% and 0.35% respectively. These are not very significant financial gains compared to the huge negative impact on payback time earlier highlighted hence it will be unattractive to value generated electricity below $0.04/kWh.

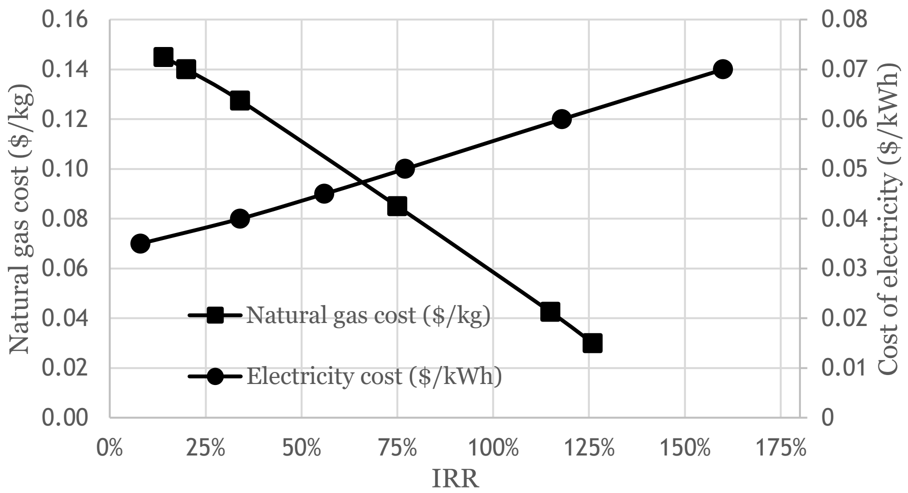

Fig. 6. Variation of IRR with gas and electricity cost

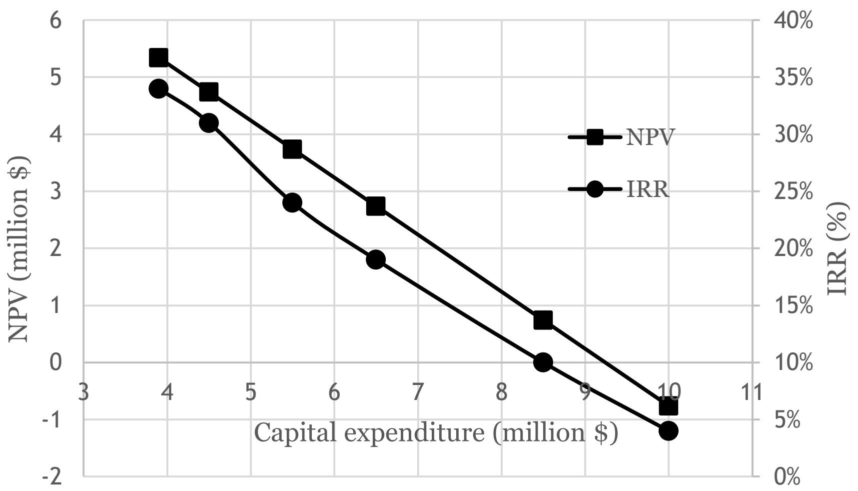

Fig. 7. Value analysis of the mechanical chiller system

In Fig. 7, NPV and IRR are plotted as a function of the capital cost of the project starting at the capital cost arrived at in earlier calculation (Table 2). It can be deduced from Fig. 7 that application of the mechanical chiller is profitable only if the cost of the project is about $9 million or less. At about that capital cost value NPV is greater than zero and IRR exceeds a rate of return of 7% typical of capital investments.

CONCLUSIONS

Gas turbine power and efficiency variation due to ambient temperature changes in a particular site in southern Nigeria have been investigated using real operation data. The potential of improving electrical power generation in Nigeria by expanding the capacity of existing GT-powered power plants through mechanical chilling of inlet air was highlighted. Also, economic analyses evaluating the benefits of having the system in place has been conducted in terms of payback period, NPV and IRR. The main findings are:

1. There is a 0.94% loss in power output for every 1^0C rise in ambient temperature at site location

ISSN 2056-5860

2. Operating a mechanical chiller at 10^0C can improve power output of the selected turbine by about 20% compared to normal operation (without inlet air cooling) at a typical ambient temperature of 29^0C.
3. A mechanical chiller cooling system can result in a positive net revenue of about $1.4 million per year at reasonably priced natural gas and electricity costs for the selected gas turbine and site location
4. The inlet air cooling equipment can pay for itself in under three years as the calculated payback period using reasonable gas and electricity prices is about 2.75 years.
5. $9 million is the recommended upper limit for capital expenditure for a mechanical chiller inlet air cooling system for a gas turbine plant in southern Nigeria if the application of the system is to result in financial gains.

ACKNOWLEDGEMENTS

The authors express gratitude to Joshua Osedaghene and others at Gbarain NIPP Power Plant, Nigeria for necessary access to gas turbine operation data. Ogbonnaya Agwu is grateful to the PTDF for sponsoring his PhD studies in Cardiff University.

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