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PERFORMANCE REVIEW: PBMR CLOSED CYCLE GAS TURBINE POWER PLANT

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Abstract:

Helium is considered as one of the ideal working fluid for closed cycle using nuclear heat source due to its low neutron absorption as well as high thermodynamic properties. The commercial viability of the Helium turbo machinery depends on operational success. The past attempts failed due to poor performances manifested in the form of drop in efficiency, inability to reach maximum load, slow response to the transients etc. Radical changes in the basic design were suggested in some instances as possible solutions. A better understanding of the operational performance is necessary for the detailed design of the plant and the control systems. This paper describes the theory behind the off design and transient modelling of a closed cycle gas turbine plant. A computer simulation model has been created specifically for this cycle. The model has been tested for various turbine entry temperatures along the steady state and its replications at various locations were observed. The paper also looks at the various control methods available for a closed cycle and some of the options were simulated.

Introduction

It is believed in the power generation circle that a modular gas cooled reactor using helium turbo machineries will have place in the market due to its unique qualities such as inherent safety, short construction time, low operational logistic requirements, efficient part load performance, quick response to sudden load changes, low environmental impact etc.

The PBMR (Pebble Bed Modular Reactor), a subsidiary of the South African Power Utility, Eskom, is engaged in the developmental work for the design and construction of a Closed cycle helium turbine plant using Nuclear heat source. Several local and international companies have shown interest in this exercise and two companies, one from US and one from Europe, have already joined PBMR as share holders. The first proto type, which is a full-scale version of the PBMR having generation capacity of 116 MW of electricity, will be commissioned in 2004.

A team of engineers and scientists in South Africa, identifying and analysing various technical issues associated with the closed cycle gas turbine plants and also coordinating various knowledge bases for finding solutions. Some of the key problem areas are material selection, modular design, testing and commissioning, prediction of plant performance, controllability and control system design etc. The lack of real test data and the inability to make scaled test version of the equipments are real challenges in this project. High levels of computational predictions are being used to help the various phases of this project from detailed design to the commissioning.

Performance prediction is one of the complex issues associated with the gas turbine plants. This is due to the cyclic nature of the plant, where the performance of the compressors depends on the driving turbines and these turbine's performances depend on the compressors. Hence iterative procedures are required to predict the plant behaviour at any point in time. In this process of iteration either the continuity of the mass flow or the energy balance can used. When using the energy balance, the compressor work is equated to the turbine work for a particular speed along the steady state operation. When the turbine work is more than the compressor work, the shaft accelerates and when it is less, it decelerates. These are transient operations. Any change in load setting will achieve only through a transient operation, which may be very short or long in duration depending on the initial and final points of operation. The steady state and transient performance prediction are important because any load setting or its change can be achieved only through a transient and the control devices which is effecting that load change should be designed based on that knowledge. Performance predictions of conventional power plants are relatively easier because each component's behaviour is independent of others and can be predicted separately. The performances of components are interlinked in the closed cycle plant and a system approach is necessary to predict it.

Nomenclature

N- Rotational speed M- Mach Number K- Kelvin P-Pressure T-Temperature
W -Mass Flow rate ω- Angular speed γ- Ratio of Specific Heats (C_p/C_v)
w -Non Dim: Sp.work τ,θ -Temperature ratios

The PBMR Plant Details

The power conversion unit (PCU) consists of two turbo compressors, power turbine, recuperator, pre-cooler and intercooler. The PCU is designed for any Generator output setting from 2.5 MW to 116.3 MW electricity.

The turbo compressors are placed in a chamber called pressure vessel, which will also be used as the duct between the HPC outlet and the recuperator cold side inlet. Hence there will be an external pressure of 70 bar on the casings of the turbo compressors. All the rotating equipments will be running on magnetic bearings primarily due to the cold welding nature of Helium, which prevents the use of conventional mechanical bearings. Magnetic bearings also improve the rotor dynamics. The total quantity of Helium in the circuit is 2500 kg and same amount is kept in the residual storage as well.

The PCU inventory control strategy is to control the Helium inventory over a range from 20 to 100% of nominal power at the same time ensuring a minimum pressure of 1 Mpa. Lower power levels are obtained by the use of Reactor bypass valve. This valve takes the high-pressure helium from HP compressor outlet and supplies it at the pre-cooler inlet. This can be used for fast acting load variations.

Interrupt valves are placed between HP compressor outlet and the cold side inlet of the recuperator. The interrupt system will be designed for over speed protection and it will act as an emergency stop of the Brayton Cycle. These interrupt valves will be stop valves without much control characteristics

Position Parameter	*Value*
Cycle core mass-flow	140 kg/s
Reactor thermal power	265 MW
Reactor inlet temperature	536°C
Reactor outlet temperature	900 °C
Generator electrical output	116.3 MW
Overall pressure ratio	2.7
Generator efficiency	98.5%
Cycle efficiency (generator excluded)	45.3%
System efficiency (generator included)	44.1%
Plant net efficiency	42.7%

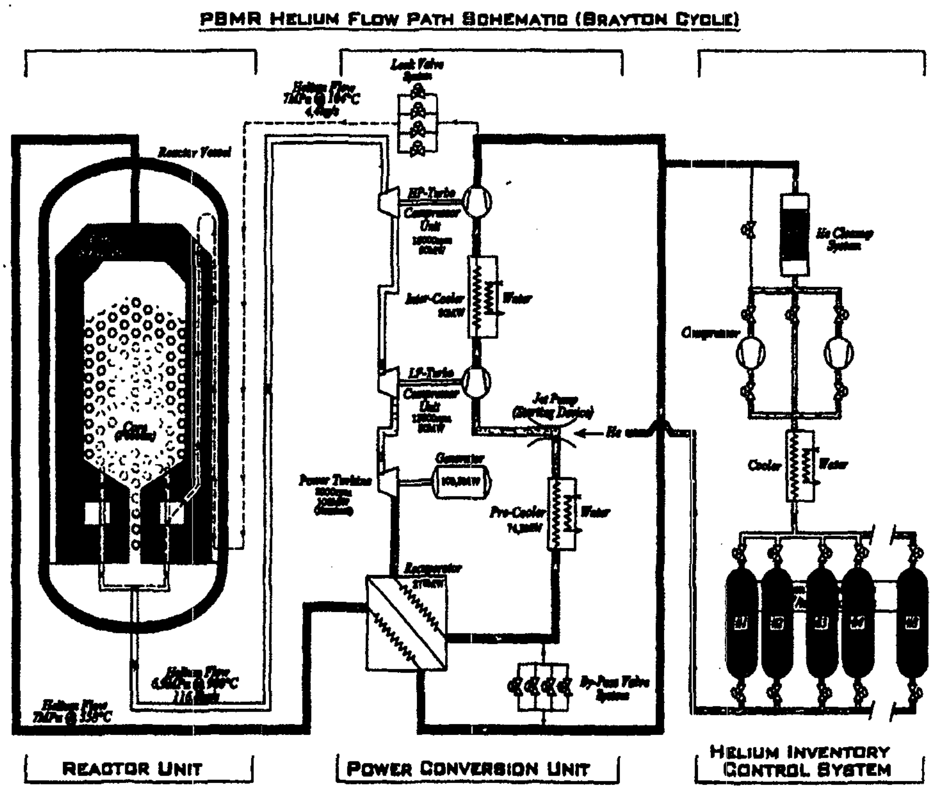

FIG. 1. Flow Diagram of Pebble Bed Modular Reactor with By-pass and Inventory control system.

System Position	P. MPa	T °C	System Position	P. MPa	T °C
1	2.59	27.9	4	6.72	900
2A	4.24	104.4	5A	5.46	812
2a	4.23	27.6	5a	4.34	721
2	7.0	104	5	2.61	553
3	6.955	536	6	2.59	138

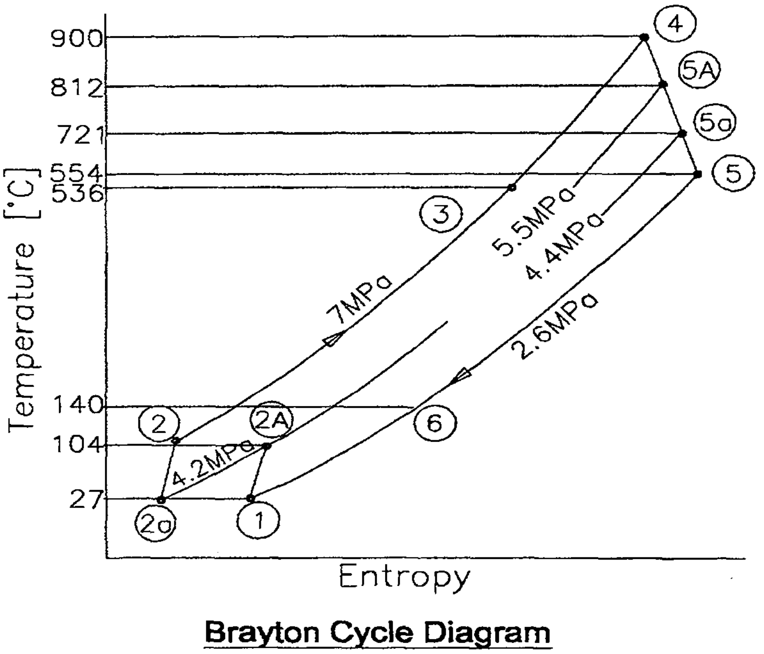

FIG. 2. PBMR Brayton Cycle.

Principle of power control

Closed cycles present unique issues and opportunities for modulating power output from power systems. Two methods, bypass control and inventory control are described and their features discussed here. Temperature modulation is also an option but it has to be done with the reactivity control and hence not described in detail here. These systems have been of interest for use in the nuclear power industry. It is important to be able to produce the part load power at high efficiency and in such a way as to minimize the thermal stress impact on the heat source, especially if the source is a gas-cooled nuclear reactor. For a given machine, the cycle pressure ratio is nominally fixed by the compressor

1. *Heat Source Bypass*

The bypass control is exercised through the bleed of high-pressure gas to short-circuit the heat source and the turbine. The throttling process is obviously a source of irreversibility so that use of such a scheme results in reduced part power efficiency.

The cycle temperatures can be held constant with just the thermal power input matching that required maintaining the cycle temperatures at the reduced mass flow through the reactor. This has the advantage in that stresses associated with temperature gradients in the metals may be held close to constant.

The impact on performance is readily calculated since the cycle temperature may be taken to remain fixed The regenerator will process equal masses on both sides at all times, which implies that the ideal design situation of $T_3 = T_6$ is maintained at part power. This is due to the fact that with constant T_4 and constant compressor pressure ratio, $T_3 = T_5$. The enthalpy balance on the mixer gives the temperature at state 6, which is, for an ideal and perfect gas, trivial. The cycle analysis is merely a work accounting with the full mass flow processed by the compressor and less in the heater and turbine An ideal cycle analysis yields

$$\eta_{th} = \left[1 + \frac{w}{w_{\max}}\left(\frac{\theta_4}{\tau_c} - 1\right)\right]^{-1} \ldots\ldots\ldots\ldots\ldots\ldots\ldots 1$$

$$w_{\max} = \left(\frac{\theta_4}{\tau_c} - 1\right)(\tau_c - 1) \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots 2$$

For values of these parameters of 4 and 1.5, respectively, the efficiency is as shown in *FIG. 3.*

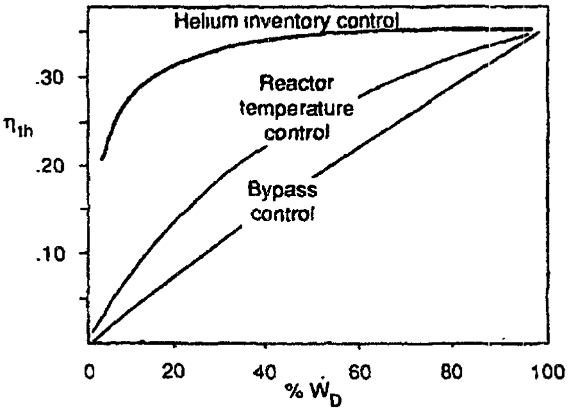

FIG. 3. Performance of the inventory and bypass controlled closed Brayton cycle[5].

The modulation of T_4 in an ideal cycle gives efficiency results that are identical to those of bypass control. The implication is that bypass and peak temperature reduction have the same thermodynamic merit. Temperature modulation and bypass may, therefore, be used together if the resulting performance is acceptable. In practice, accurate part power performance is evaluated with significantly greater consideration of the irreversibilities.

2. *Inventory Control*

A good method of producing part-load power is available to closed cycle engines where the pressure and thus the density of the working fluid may be controlled by connecting the cycle fluid to a storage vessel.

A compressor is used to pump the working fluid out of the system of working components. The reduced mass of the circulating fluid results in a smaller mass flow rate, which, in turn, reduces power output from the system. Means are also provided to allow the return of the fluid to the cycle when power is to be increased. In order to minimize heat transfer in the storage component, the fluid is removed from the lowest temperature point in the cycle with appropriate means for cooling.

The operation of the cycle at reduced mass flow rate allows operation with the same temperatures and pressure ratio This means that the heat engine operates with the same thermodynamic cycle, resulting in approximately constant efficiency and specific work.

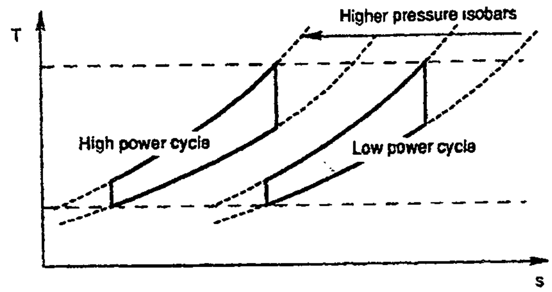

FIG. 4. T-S diagrams for part and full power for the inventory controlled closed cycle [5].

The fact that the temperatures remain invariant, as the mass flow rate is reduced, implies that the local sound speeds are constant. Blading and flow-passage geometric design fix the local Mach numbers so that local flow velocities are everywhere constant to first order. With velocities constant, the mass flow rate is proportional to the density, which, for constant temperature, is also proportional to the absolute pressure. The thermodynamic cycle operating at various pressure levels can be shown as in *FIG. 4*. The variation of cycle performance is a function of the working fluid properties. These properties are insensitive to absolute pressure when the gas is monatomic. For other gases, the effect of changing pressure may be significant in affecting specific work and efficiency.

It is expected, therefore, that the relation between efficiency and fractional power is relatively flat, and a small fractional power output can be obtained by operating at low absolute pressure. In practice, the fluid frictional losses are slightly altered because the decreased density also decreases the flow Reynolds numbers. This increases the importance of viscous losses. The effect is to reduce efficiency slightly as the power output is reduced because component efficiencies are reduced.

The peak and part-power cycle efficiency noted for bypass and temperature modulation control on a realistic analysis are about half the value noted for an idealized analysis for the same temperature extremes, showing the importance of the irreversibilities. Further, the serious degradation of efficiency with bypass control is noted as a disadvantage in relation to inventory control. It should be noted, however, that the severity is important only if the fraction of time spent at less than full power is significant[8].

Theory of Dynamic Behaviour

The physical properties of the dynamic behaviour can be classified into 3 groups according to their velocity.

The first group, which runs with small characteristic times belong to the mass inertia due to the pressure changes caused by gas oscillations. The period of these oscillations amounts to some deciseconds [8]. This can be considered as the running time of the gas particles in passing the turbo machines, heat exchangers and ducts.

The second group is due to the mass storing in the volumes of the gas ducting components and the energy storing of the rotating masses causing a change in rotor speed. This charging time and the machine inertia time are in the range of couple of seconds.

The third group belong to the heat storage in the heat exchangers and the corresponding changes in temperature associated with it. Normally these time constants range at least one-tenth power over those of the second group. Particularly Helium cooled high temperature reactors have great thermal inertias according to their large masses.

There are some other physical phenomenon faster than the above mentioned, like the one caused by the wake dents behind the trailing edges which strike a blade at the frequency of rotor speed time the number of blade per stage. These can be in the order of 10^{-6} seconds. Such dynamic behaviours can be regarded as quasistationery.

With regard to the focus of a calculation, the physical phenomena of one or more groups must be considered [8]. When focusing on the pressure gradient in the hot gas duct during opening or shutting the by-pass valve, the third group, which deal with the heat storage of heat exchangers, can be neglected. Similarly the same third group has to be considered if means for keeping reactor inlet temperature constant during start up. The second group must be considered for the speed control calculations. Here the inclusion of the procedures belonging to the first group can be necessary in case of a greater control operation. The dynamic behaviour of the reactor and the heat exchangers can be neglected based on an assumption that the turbine entry temperature is kept constant by the reactor control. The mass storing in the volumes and the energy storing of shafts have a certain priority and it must be considered for all practical simulation purposes.

Transient Performance Analysis and Control System Design

Transient performance and control system design are inseparable. During transient manoeuvres engine operation is inherently more prone to undesirable events than running steady state. These must be avoided by engine and control system design [6]. Some of the transient operations of a typical power generating unit are load ramping, load rejection, islanding etc. If the unit is connected to a major grid, the frequency has to be maintained to avoid the isolation of the unit from the grid also the unit should keep running for the house load if it isolated from the grid due to trouble in the grid. If it is an independent power producer, not connected to a grid, more rapid power changes are required as there are fewer load devices, which may be switched on and off instantaneously.

The PBMR unit is targeting for a 10% load change in 1 second, which will put heavy target on the control system design. Control system design for a conventional gas turbine or steam turbine driven generator is an established technology and hence it is readily available in the market. But closed cycle gas turbines with helium bringing a different set of issues and hence the transient behaviour of the turbo machineries have to be analysed properly for developing an accurate control system. Various phenomena particular to transient performance are identified and their impact has to be studied [6]. The various phenomena are

a) Heat soakage

During transient operation there are significant net heat fluxes between the working fluid and the engine metal, unlike for steady state operation. This can be up to 30% for acceleration from idling to full load. The impact of the soakage during transient can be very dramatic in this CCGT due to the presence of the heat exchangers. The modelling of the Heat soakage can be done by calculating the heat flux to or from the metal component.

Heat Soakage = f(heat transfer(kW), heat transfer coefficient(kW/m^{2}K), gas and metal temperatures(K), gas mass flow(kg/s), area of metal(m^2), mass of metal(kg), CP of metal (kJ/kgK)

Heat Soakage, $Q = h.\ A\ (T_{gas}\text{-}T_{metl})$
$dT_{gas} = -Q/W\ CP_{gas}$
$dT_{metal}/dt = Q/\ (Mass.metal.\ Cpmetal)$

The T_{gas} *and* T_{metal} *are at time t. Hence for a given heat transfer Q is derived and then* dT_{gas} *and* dT_{metal} *are evaluated for the given time step dt as follows*

Approximate heat soakage = f(max. heat soakage(kW) , time(s), time constant(s))

$Q = Q_{max}\ e^{(-t/TC)}$

TC , time constant,ranges from 5s for a 200kg engine to 40s for a 2 tonne engine.[6].

The component geometric data, thermal masses and heat transfer coefficients are assumed to be available.

b) Volume packing

During steady state operation the mass flow entering a given volume is equal to that leaving. This is no longer true under transient operation as the pressure, temperature and hence density of the fluid changes with time. This is known as volume packing and can have a notable impact upon an engine's transient performance, especially for the largest volumes such as ducts and heat exchangers. The volume dynamics should be accounted for ducts, the combustor and heat exchangers. The following equation allows the change in mass flow leaving the volume, relative to that entering to be calculated.

Rate of mass storage in a volume (kg/s) = f(γ, Mach number, gas constant (kJ/kgK), mean temperature(K), mean pressure(kPa), volume (m^3))

$$W_{in} - W_{out} = \frac{V\, dP/dt}{RT\left(1+\left(\frac{\gamma-1}{2}\right)M^2\right)^{\frac{1}{\gamma-1}}} \quad \text{……………………………………….(3)}$$

dP/dt is calculated from the known values of P at time t and time t-1.

Volume packing has a special significance in the PBMR model. The HP compressor is discharging the pressurised helium into a chamber, which houses the turbo machineries. The cold side of the recuperator inlet, where to the gas is flowing, is at the other end of the chamber and hence the gas is taking a complex passage over the pipelines and the equipments. A CFD modelling of this volume with all the internals is planned at a later stage. The output from this CFD analysis will be used in the computer models as an exercise to improve the accuracy.

c) Tip clearance changes

During the acceleration, the thermal growth of the compressor or turbine disc is slower than the pressure and the thermal growth of the casings, causing blade tip clearances to be temporarily increased. The converse is true during a deceleration, which can lead to rubs. This change in compressor geometry affects its map, the main issue being lower surge lines. There is also second order reduction in flow and efficiency at a speed [6].

This is for the conventional and generic geometrical design. In the PBMR design, the compressors and the turbines are situated in a pressurised chamber and hence the pressure growth of the casings will be different from that of a conventional nature during a transient. Hence the tip clearance change during transient operation has to be modelled or calculated separately to be incorporated in to the computer models. Changing Tip clearances and interstage heating may significantly lower the surge line. The changes to the compressor map are of second order and may be ignored for simple transient performance models.

d) Heat source (Reactor) delay

This is the time delay between the fuel supply and the actual heat rejection. Compared to conventional GTs using liquid or gaseous fuels, gas cooled reactors have more heat source delay due to the slow reaction time and also due to the high heat carrying capacity of the graphite in the reactors. The time delay from reactivity change to the release of heat in the reactor can be obtained from the Reactor designers.

e) Heat transfer within multi-stage components

Where a single map is used to model a multi stage component such as an axial flow compressor, net heat transfer will have a second order effect upon the map during a transient. This is due to its effect upon gas temperature through the component and hence stage matching, as it changes the referred speed and hence flow capabilities of the rear stages.

e) Control system delay and lags

Control hard wares such as valves, variable guide vane actuation rings etc. take a finite time to move to new positions demanded by the controller during a transient. This finite time may comprise a delay, where there is no movement for a given time and or a lag where the device is moving but lagging behind the demanded signal. Control system sensors measuring parameters such as pressures and temperatures will also show delays and lags relative to conditions. The following equations can be used for delays and lags in movement of control system hardware components. The delays and time constants need to be provided by control system component system manufacturers.

Lagged value of parameter = f (lagged value at previous time step, time step, time constant)
*Lagged Value(time = T) = (Lagged value (time = T-DT) + (Actual value (time = T))/(TC + DT) (*e.g.: Actual value of a gas temperature, and lagged value of a thermocouple)

Delayed value of a parameter,
Value (time = T) = Value (time = (T – delay))

More analysis of this aspect should go hand in hand with the suppliers of control hard wares. However a good forecast of the allowable limit of the delay caused by control hardware, for a better transient performance, can be very useful in the detailed design of control hardware. The computer models have to be run for various durations of delays originating from the control system and the effect on the outputs have to be analysed. But a totality of the physics of the transients have to be established by incorporating all the influencing factors mentioned earlier to get a complete picture.

The computer model and the results

A computer model has been developed specifically for this particular project. This is capable for testing the steady state and transient operations. This model carrying only the elementary features at present, but more will be added to it, as detailed designs of the project are available. The various leakages and other pressure losses are based on empirical approximations due to the lack of specific information on the physical design and hence this paper deals with only the steady state performance and detailed transient model will be made available in the future.

Theory of the Simulation Model

To simulate the transient or off design performance of gas turbine engines, the transient period is segmented into time intervals. For each time interval, the calculation of the thermodynamic parameters of the gas path has to be carried out. Once these parameters (temperature, pressures, mass flows etc.) have been found, the power input and output for each component can be calculated. Then a power balance can be carried out for each shaft, and hence the accelerating torque can be calculated. This accelerating torque is then integrated over the time interval, and the change in shaft speeds is obtained. This process of thermodynamic variable calculation and torque integration is repeated over several time intervals as required.

In this procedure, the difficult task of thermodynamic variable calculation can be accomplished by either constant mass flow method (CMF) or inter component volume method (ICV). Inter component volume method is chosen for this exercise. This is the more realistic, since it includes allowance for gas mass storage, which is ignored in the CMF method

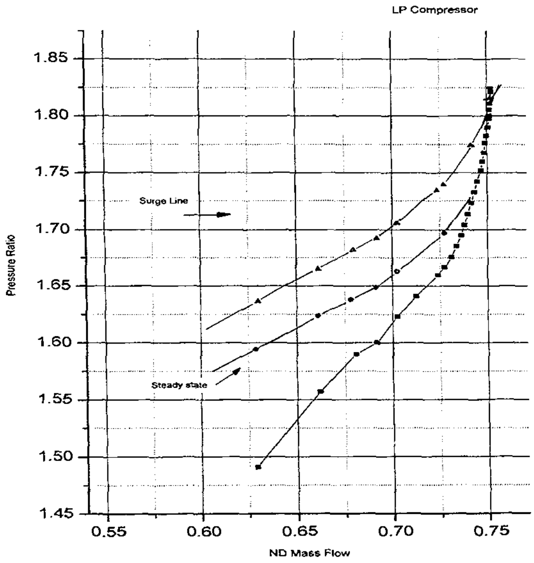

FIG. 5. Compressor performance on a rising TET.

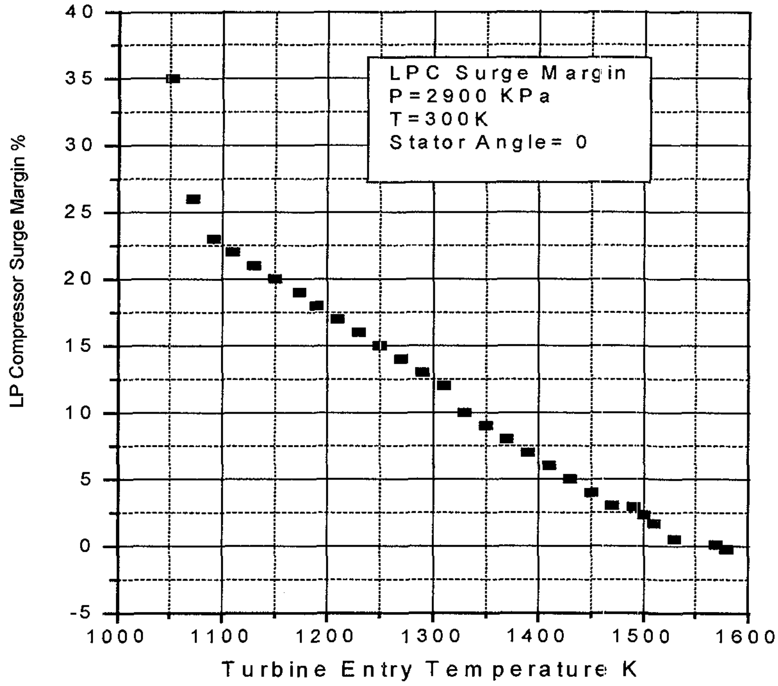

FIG. 6. Change in LP Compressor Surge margin as the TET goes up, keeping the compressor inlet Pressure and Temperature constant.

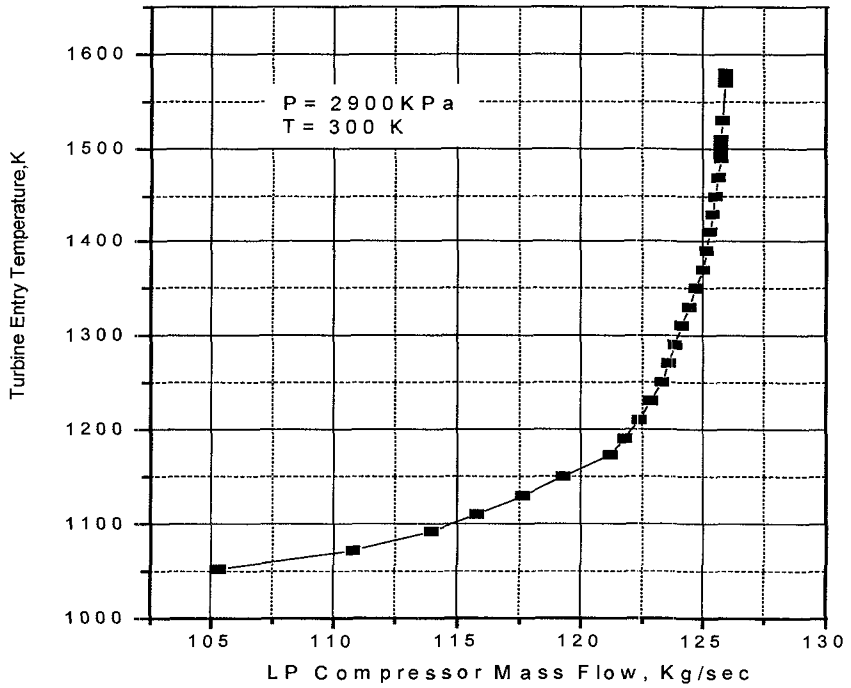

FIG. 7. Mass flow through the compressor at various TET.

For each time interval in the segmented transient period, the calculation of the thermodynamic parameters of the gas path has been carried out. Once these parameters have been found, the power input and output of each component can be calculated. Then a power balance has been carried for each shaft and thus accelerating torque can be calculated. This accelerating torque is then integrated over the time interval, and the change in shaft speed is obtained. This process is repeated over several time intervals as required [11].

The method of inter component volume method (ICV) is used in this model. The ICV method is more realistic, since it includes allowance for gas mass storage, which is ignored in the constant mass flow (CMF) method.

The PBMR plan to use the inventory control system to do the load variation at the rate of 10 MW/min. The *FIG. 9* shows that in order to achieve that target the pressure (absolute) change required is around 3.4 bar/min (calculated from the slope). Also the change in mass flow rate is 8 kg/sec/min. This type of information can be useful for the design of Inventory control valves.

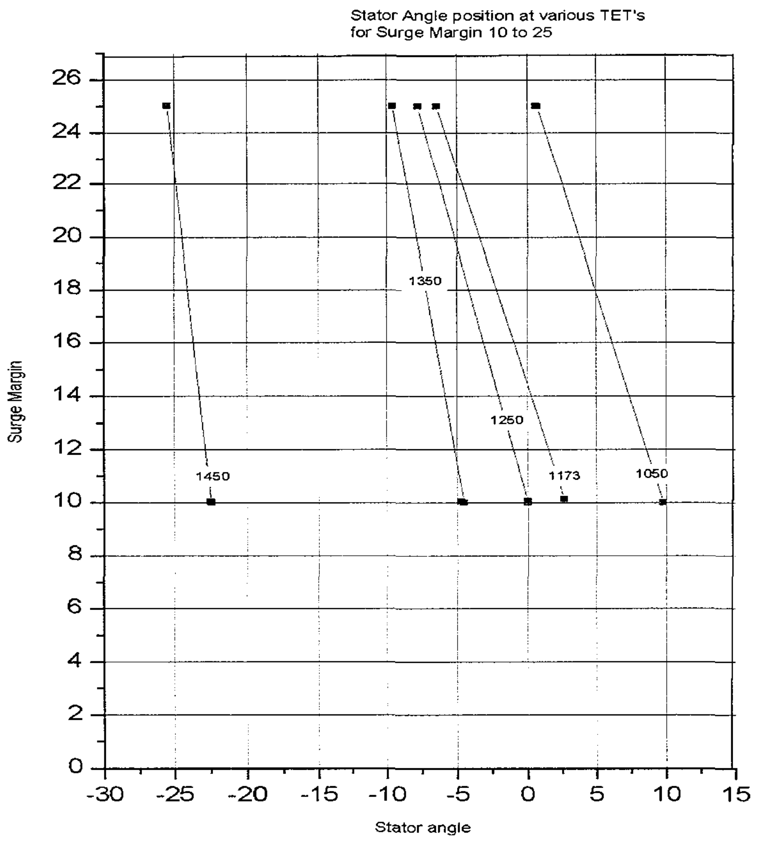

FIG.8. LP Compressor Stator Angles at different TET's for maintaining the Surge Margin between 10% and 25%.

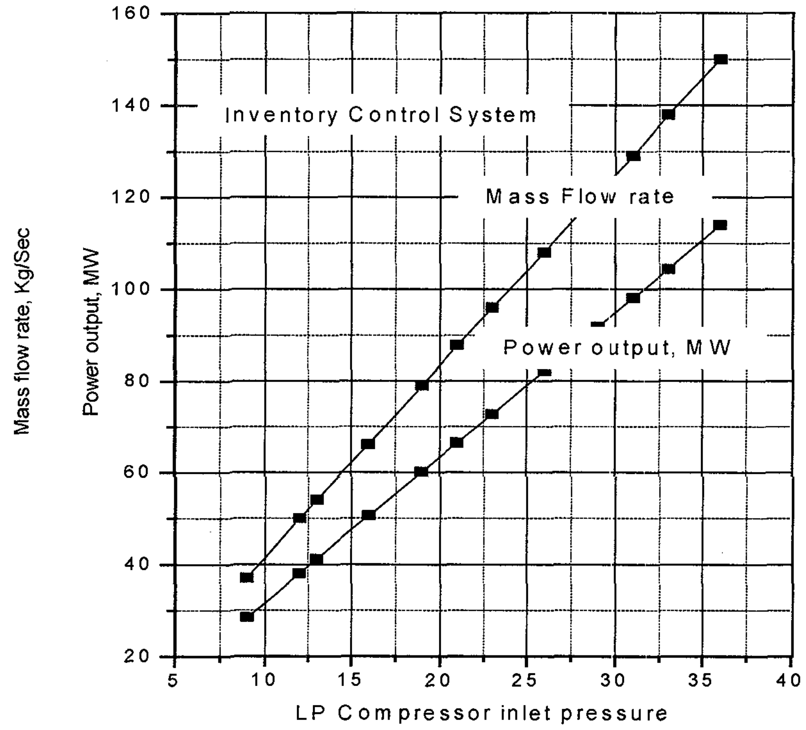

FIG. 9. Effect of change in the system pressure (LPC inlet Pressure shown) on the plant output. The efficiency, Pressure Ratio, TET are theoretically constant for the whole range.

Overall Gas Turbine Performance

The gas turbine part load performance using by pass control is an attractive option due to its rapid action. But an efficiency penalty has to be paid for this. A rough estimate shows that 35% reduction in the flow will be sufficient to bring the unit to synchronous idling. Thesudden decrease in mass flow rate through the reactor may cause in an increase in Turbine Entry Temperature beyond the design value. Another operational problem will be the possibility of surge due to the rising TETs as seen in the *FIG. 5.* However introducing a change of setting for the compressor stator vanes may push the operating line towards the steady state to avoid the surge due to rise in turbine entry temperature. The PBMR high-pressure turbine blades are cooled blades and hence it may have the capability to withstand temperature excursion for a short duration.

Helium inventory control is the most efficient part load control method. The system efficiency will almost same for a wide range of loading from 40% to 100%. The potential for thermal stresses are also minimal. The response time of this method can be considered as moderate and hence it can be used for normal and pre calculated load variations. The drawback of this system is that the logistics required for this method can be space consuming.

Conclusion

The operational performance is subjected to the finding of successful solutions of hard-core technical issues associated with Helium Turbines such as cold welding effects, difficulty in containing helium with the seals etc. More elaborate experimental study coupled with computational work is being carried out to reduce the uncertainty in this area. The future exercises are planned to focus on the time delay of each control system. The final selection of equipments can only be done after considering the transient effects of each component and the system. However it is convinced that there is a market place for modular high temperature reactor for remote load centres where establishing logistics for conventional power plants is not viable and also for urban load centres where there is a need for efficient part load performance.

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