OPTIMISATION STUDIES FOR OHAKI GEOTHERMAL POWER STATION

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ABSTRACT

Ohaki Geothermal Power Station, to be located some 20 km north of the existing Wairakei Geothermal Station, will consist of two 40-45 MW condensing turbo-generators operating in conjunction with two 11 MW back-pressure turbines being transferred from Wairakei. Direct contact barometric condensers will operate with a single natural draft cooling tower. Incondensible gas content of the steam will be fifteen times greater than at Wairakei and will necessitate large centrifugal exhausters. An optimisation programme has been developed to determine the response of the total turbine, condenser, gas exhauster, cooling tower, circulating pump system over the full range of climatic conditions. The programme has been used to establish design data for the cooling tower, turbo generators, condensers, gas exhausters and the main cooling water pumps. Other uses include determination of the value of using variable flow cooling water pumps and the value of changes in condenser and intercooler design parameters.

1 INTRODUCTION

The aim of the study was to determine optimum design parameters for the turbo generators, condensers, gas exhausters and cooling tower. These were defined as the design parameters which maximised the net present worth of the station - that is the capitalised value of net station output less the construction cost of plant items being investigated, all present worthed to the commissioning date.

The steam flow into the IP turbines was set at 680 tonnes/hour (not including gas flow). This decision was based on reservoir capacity. Variations in station efficiency therefore affect station output and can be evaluated using a capitalised value of power.

The figures presented in this paper are based on a capitalised value of power of $2000/kW. This assumes that some capital investment is deferred by efficiency improvements. The capitalised value of power is directly related to the load factor. The high load factor of geothermal stations (87% assumed) gives a high premium on efficiency.

There are several features of Ohaki Power Station that affect the optimisation study.

(i) The use of a natural draught cooling tower. A natural draught tower is responsive to variation in ambient conditions. It is therefore necessary to consider these variations when determining overall station output.

(ii) The presence of an estimated 5.6% of gas in the steam supplied to the intermediate pressure (IP) turbines. The high gas content makes the power consumption of the centrifugal exhausters a significant factor.

(iii) The use of a direct contact condenser which requires a system of control that minimises the amount of geothermal gas in the cooling water.

(iv) The corrosive environment in the turbine. This imposes a limitation on material selection and allowable stress levels particularly for the last blading stages. The purchase specification will give emphasis to our concern in this area because, if the value of an increment of output was the only consideration, it could lead to a risky design under the stress corrosion conditions existing in the turbine. The study therefore assumed a moderate final stage blade length for the turbine. For similar reasons the operation of the station was constrained so that the theoretical wetness at the turbine exhaust did not exceed 13%. This limited the turbine exhaust temperature to a minimum of 41.5°C.

There are several different possible arrangements for the cooling water circulation. Two have been examined.

(i) A system using variable output cooling water pumps as shown in diagram 1. In this system the full pump flow goes through the condensers and intercoolers and generator coolers (the latter not shown).

(ii) A system using fixed flow pumps as shown in diagram 2. The surplus water above that required for the condensers etc. passes through the overflow to the hot-well. The system is shown with two flow rates.

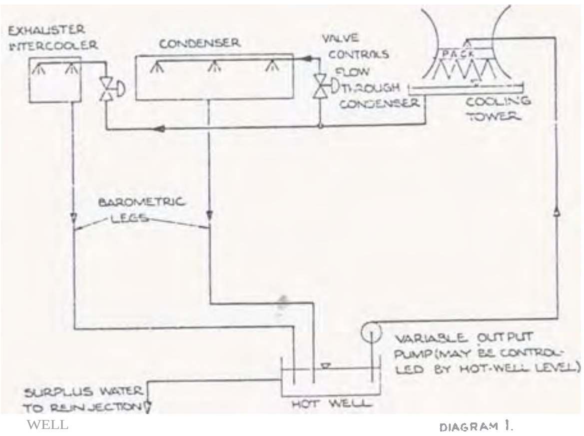

DIAGRAM 1.

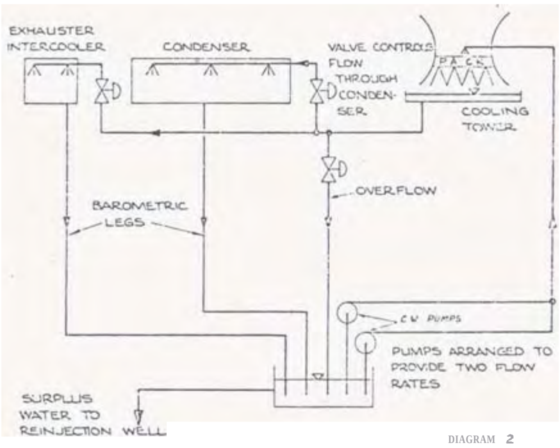

DIAGRAM 2

2 INPUT DATA

The study was made possible by the fortunate acquisition of adequate input data. Had there not been detailed hourly climate data available, or if manufacturers had not made available generous budget price and technical information, the study would have been much less complete.

2.1 Climate Data

The Meteorological Service and Ministry of Works and Development have made available data from a climate station at Grid Reference 782550 which is the near the site of the Power Station. This data was in the form of hourly values of dry bulb temperature and relative humidity, covering, with some gaps', the period January 1976 to July 1980. Using a formula provided by the Meteorological Service the relative humidity values were converted to wet bulb temperatures and the resulting temperatures were analysed to give a frequency of occurrence table.

The correlation between the climate station data and data measured at the station site was checked and found to be good.

2.2 Tower Costs and Performance

Initially tower size and response to changes in ambient conditions were calculated using formulae developed from a paper by Chilton. (Performance of Natural Draught Cooling Towers Proc. Inst. Mech. Engr. 1952). However it was discovered that this approach was not accurate particularly in determining the performance of the tower at widely varying climatic conditions.

The State Electricity Commission of Victoria (SECV) supplied data on the performance of cooling tower packing and on resistance to air flow. This information was used to determine tower size and performance. The tower cost was taken as:

$\$(1590\ D^2 + 36\ D^2\ H_p)$

where D and H_p are pack diameter and depth. The basic data supplied by SECV is detailed below.

Cooling Tower Data

Symbols are as defined in BS4485. The following additional symbols are used.

D	pack diameter	(m)
H_a	air opening height	(m)
H_p	pack height	(m)
D_t	throat diameter	(m)
ρ_m	mean density of air (inlet air and air above pack)	(kg/m^3)
$\Delta\rho$	change in density of air from inlet to above pack	(kg/m^3)
g	acceleration due to gravity	(m/sec^2)
H	effective height of tower taken as the height from the mid point of the pack to the top of the tower	(m)

Heat Transfer

The heat transfer of splash type packing is expressed as:

$$KaV/L = \lambda(L/G)^n = \int \frac{cdt}{h_L - h_g}$$

where: $\lambda = .252L^{-.23}$ per metre height of packing

$n = .154L^{1.41}$

This gives:

$$(KaV/L)_{pack} = .252L^{-.23}\ (L/G)^{.154\ L^{1.41}}$$

per metre height of pack.

The spray below the pack also provides heat transfer capability given by -

$(KaV/L)_{spray} = .07\ (L/G)^{.34}$ per metre of spray height.

If H_p is the pack height and (Ha-1) the height of the spray (H_a is the air opening height and it is assumed that the pack extends into this by 1 metre). The total heat transfer is given by:

$$KaV/L = .252\ H_p\ L^{-.23}\ (L/G)^{.154}L^{1.41}$$

$$+ .07(H_a-1)\ (L/G)^{.34}$$

A de-rate factor of 15% is applied to λ(pack) which changes 0.252 to 0.215. In the calculation KaV/L is evaluated using Tchebycheff's approximation as described in BS 4485 part 2.

Resistance to air flow

Pack 4.059 $L^{.73}\ G^{1.17}$ per metre of pack height

Spray 1.05 $(L/G)^{1.4}$ per metre of spray height

Shell .167 $(D/Ha)^2$

Eliminator 3.9 (Constant)

Columns etc. 2.3 (Constant)

Throat $(D/D_t)^4$

A de-rate factor of 15% is applied to increase the pack and spray values. The above resistances are in velocity heads. The total resistance in velocity heads then equals: $2H\rho_m\Delta\rho g/G^2$

2.3 Pump Costs and Pumping Power

Based on costs from one manufacturer a cost per pump of 145,000 + 77 x (flow in litres per sec) was derived. Assuming three 50% pumps this becomes

435,000 + 116 x (total flow in litres/second) ($)

The pumping costs assumed a head of 9 m plus the air opening height plus the pack depth. The power in kw, assuming an overall pump efficiency of 80%, was taken as .0123 x head x total flow in litres/second.

2.4 Condenser Costs

Data from several manufacturers of the cost of condensers at various design turbine exhaust temperatures was available. However, as it became apparent that it would always be economic to operate a considerable percentage of the time at the exhaust pressure determined by a 13% limit on turbine theoretical wetness the condenser cost estimate was deleted from the optimising programme.

2.5 Exhauster size, cost, power consumption and intercooler heat load

Information was given by several exhauster manufacturers on the variation in attainable suction pressure with suction temperatures.

Design suction pressure (bar)		0.07	0.10	0.13
Attainable suction pressure (bar) at suction temp of:	33°C	0.07	0.10	0.13
	23°C	0.051	0.078	0.016
	13°C	0.035	0.060	0.086

A calculation procedure, based on the proposition that the head developed by a centrifugal compressor was independent of the nature of the fluid, its inlet temperature and whether it is cooled during compression, was developed and correlated well with the information from manufacturers. The results were incorporated by curve fitting into the formula.

$$P = (0.006367e^{18.57S})\ e^{(.0689t_s e^{-9.6645\ S})}$$

where P is attainable suction pressure in bar
and t_s suction temperature
S exhauster size

The exhauster 'size' S is the suction pressure in bar that the exhauster can attain with a suction temperature of 32.5°C.

Using normal centrifugal compressor formulae and a curve fitting procedure the exhauster power required was expressed in kW as:

$$((0.02866S^{-2.7951} + 10)e^{((9.992S^{-0.002876}-10)t_s)}$$

$$+ 3647.2e^{-26.28S} + 440) \times 2.857$$

The heat rejected in the intercoolers of the gas exhausters is given in kJ/sec by:

$$(28.8131e^{.276ts+9600})e^{-((1.85239e^{.08536ts}+18)p)}$$

$$+102.452e^{.10354t_s} + 700$$

This formula assumes 2 intercoolers per exhauster but in terms of system optimisation the difference with 1 intercooler per exhauster is not significant. The exhauster cost was taken as ($M):

$$.5195\ S^{-0.586}$$

The above formulae are for a gas flow of 20 t/hr.

2.6 Turbo-generator costs and output

One manufacturer made available back-pressure/generator output curves for three different turbines These turbines were designed for different design-point back-pressures and varied in exhaust annulus area. Using these curves and a curve-fitting process the following formula for MW output was developed.

MW (per turbo-generator)

$$= 35.7 + (244.1575e^{.033414695A} - 270) \times$$

$$(.1451-V)^{(12.5037218\ (A-3)^{.0251925}\ -\ 12.5)}$$

Where A is exhaust annulus area (m^2) and V is the back pressure (bar). The formula is for 5.6% gas content.

The turbo-generator cost was derived as

1.785 $A^{.471}$ in $m per turbo-generator.

2.7 Heat Content of Steam/gas entering the Condenser

This was taken as equal to -

4.1868 (123747 - 242.18(MW) kJ/sec where MW is the

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total MW output from the two IP turbines. The energy in the gas entering the turbine was allowed for in setting the figure 242.18 by assuming that 2 kg of gas was equivalent in turbine power to 1 kg of steam. (This approximation has been proposed by R James).

2.8 Basic Thermodynamic Data

The calculation on the cooling tower requires data for the heat content of air and its density. The vapour pressure of water is required for the turbo-generator calculations.

The heat content of air was taken as equal to the heat content of the saturated air at the wet bulb temperature.; This was represented by the formula

$$1.732 \times 10^{-32} (273.33 \times WB)^{13.633093} \quad kJ/kg$$

where WB is the wet bulb temperature (°C).
The density of saturated air was taken as

$$1.29 - .00485\ WB\ kg/m^3$$

For non-saturated air the formula used was

$$(1.29 - .00485\ WB)(273.33 + WB)/(273.33+DB)\ kg/m^3$$

where DB is the dry bulb temperature (in °C)

The vapour pressure of water was determined by Antonio's formula.

$$V = e^{12 - 4002/(t + 234)}\ bar$$

where t is the water temperature in °C.
The specific volume of saturated steam was taken as

$$193.67\ e^{-.064685t} + 5$$

where t is the saturation temperature-
The partial pressure of geothermal gas at turbine inlet was calculated from:

$$P_g R(t+273.33)/4300(193.67e^{-.064685t} + 5)bar$$

where P_g is the flow of gas as a proportion of steam flow.

3 SYSTEM OPERATING MODES AND METHOD OF CALCULATION

The programme that has been developed calculates system performance for each climate condition and computes a net station value. The main parameters are adjusted stepwise to arrive at a maximum net station value.

The two main modes of operation are 'exhauster limited' and "tower limited'. For the exhauster limited mode, if the exhauster suction capacity was increased the turbine exhaust pressure would fall. For the tower limited mode the turbine exhaust pressure would not be reduced by an increase in exhauster suction capacity.

The tower equations (refer to 2.5) can be used to determine the air rate (kg of dry air/m²sec) and the hot water temperature. The calculations also require, besides the ambient wet and dry bulb temperatures, the water rate (kg/m²sec) the heat rejected in the tower (expressed in the programme as kg°C/sec) and the cooling range. A solution of the tower equations is obtained iteratively and each time the air rate or hot water temperature is changed a series of equations is solved in a subroutine to give the water rate, cooling range and heat rejected. These equations are derived from condenser, hotwell and intercooler heat balances. (Changes in the heat rejected are small but are included so that the programme can determine the value of changes in intercooler design parameters and also because it seemed easier to include them than to do otherwise).

The subroutine itself is iterative. It starts with the existing value of the cooling range and corrects it to a value that matches the values of hot water temperature and air rate. For the exhauster limited case the structure of the subroutine is as follows:

Calculate:

1 - exhauster suction pressure (a function of exhauster suction temperature)
2 - turbine exhaust pressure
3 - turbine exhaust temperature
4 - MW output
5 - heat rejected in condenser, intercooler and generator cooler
6 - evaporation in cooling tower
7 - temperature rise of water in condenser
8 - cooling range
9 - $F(\Delta T)$ = calculated cooling range - assumed cooling range

A Newtons approximation method of solution is used to get $F(\Delta T) = 0$.

For the tower limited case the equivalent subroutine structure (again starting with an assumed value of cooling range) is:

Calculate:

1 - exhauster suction temperature
2 - with an assumed value of turbine exhaust temperature calculate -
  (a) MW output
  (b) heat rejected
  (c) evaporation
  (d) temperature rise of water in condenser
  (e) turbine exhaust temperature and pressure
  (f) exhauster suction pressure

  This calculation is repeated until the change in turbine exhaust temperature is <.001°C.

3 - cooling range
4 - $F(\Delta T)$ = calculated cooling range - assumed cooling range

Again a Newtons approximation method is used to get $F(\Delta T) = 0$.

Both subroutines produce a value of heat rejected and the water rate can easily be calculated (only required in the exhauster limited condition when variable flow pumps are assumed). For fixed flow pumps the overflow (cumecs) can be calculated.

The exhauster limited mode has a constraint in the programme which prevents the turbine exhaust temperature from going below the wetness limited figure.

4 RESULTS

Sample printouts from the programme are given below. Table 1 is a montage of 3 studies to show the variation in design points arising from different cooling water systems. It can be seen that there is an economic margin in favour of variable flow pumps. This margin however needs to be discounted by the extra cost of such pumps and by the extra down-time caused by their electrical or mechanical complexity. Also, changes in pump efficiency with flow variations were not considered. The margin in favour of two flow rates as against one is not so great and it is doubtful whether the additional complexity is justified (6 pumps, 3 large and 3 smaller as against 3 large pumps).

Table 2 is an abbreviated run from the single flow rate option referred to above. FRQ is the cumulative frequency of the tabulated data which is ordered according to the re-cooled water temperature RWT. From the table it can be seen that 32.2% of the time RWT is equal to or less than 21.8°C. DB and WB are the ambient wet and dry bulb conditions for which the programme calculated other data on each line. CR and HWT are the cooling range and hot water temperature to tower respectively. Other columns give turbine exhaust conditions, net MW output (after allowing for exhausters and CW pumps) and cooling water flow rates including evaporation and re-injection from the hot-well.

The difficulty of specifying a single design point for the turbo generator can be seen by considering the frequency distribution of the turbine exhaust pressure (X PRESS). The turbine will in fact run over a range of exhaust pressures determined by ambient conditions and any other control constraints.

TABLE 1

		FIXED FLOW CW PUHPS ONE FLOW RATE		FIXED FLOW CW FUHPS TWO FLOW RATES		VARIABLE FLOW CW PUMPS
CALCULATED TOWER DIMENSIONS						
PACK DIAMETER (M)		59.700		60.10		59.90
HEIGHT (M)		83.580		84.14		83.C6
PACK DEPTH (M)		6.600		7.10		7.00
RATIO DIAM/HGT AIR OFENING		7.80		7.80		7.80
TOWER DETIGN DATA						
FLOW IO TOWER AT DESIGN POINT (CUMECS)		6.50		5.98		6.08
MAX FLOW TO TOWFR (CUMECS)		6.50		6.50		7.00
RATIO MAIN PUMPS/TOTAL FLOW				0.92		
RATIO FOR SMALL PUHP CUT IN				1.02		
WATER RATE AT DESIGN POINT (KG/SQ. M SEC)		2.322		2.108		2.156
AIR RATE AT DESIGN POINT (KG/SQ. M SEC)		1.702		1.717		1.712
DESIGN DRY BULB (DEG)		10.00		10.00		10.00
" WET BULB (DEL)		9.50		9.50		9.50
APPROACH (DEG)		13.33		12.87		12.99
RECLD WATER TEHP (DEG)		22.83		22.37		22.49
DESIGN COOLING RANGE (DEG)		15.32		16.63		16.37
HOT WATER TEMP (DEG)		38.14		39.00		36.87
HEAT REJECTED (GJ/HOUR)		1471.75		1467.90		1468.74
TURBO-GENERATOR DESIGN DATA						
ASSUMED EXHAUST ANNULUS AREA PER TURBO-GENERATOR (SQ M)		6.66		6.66		6.66
MAXIMUM OUTPUT PER T/G (MW)		45.00		45.00		45.00
GENERATOR COOLER FLOW (CUMECS)		.12		.12		.12
EXHAUSTER DESIGN DATA						
EXHAUSTER RATING AT 32.5 DEG SUCTION TEMP (BAR)		0.0992		0.1000		0.0996
MAXIMUM EXHAUSTER FOWER (MW)		4.95		4.89		4.95
EXH SUCT TEMP DESIGN F'GINT (DEG)		24.03		24.37		24.49
EXH SUCT PRESSURE DESIGN POINT (BAR)		0.077		0.077		0.077
INTERCOOLER TEMP RISE (DEG)		7.00		7.00		7.00
ASSUHED CONDENSER DESIGN DATA						
TERMINAL TEMP DIFFERENCE (DEG C)		3.00		3.00		3.00
GAS APPROACH TEMP DIFFERENCE (DEG)		2.00		2.00		2.00
EXHAUSTER SUCTION LOSS (BAR)		0.010		0.010		0.010
ECONOMIC FACTORS						
DISCOUNT RATE AS DECIMAL		0.10		0.10		0.10
CAPITALISED VALUE OF POWER ($/KW)		2000.00		2000.00		2000.00
CALCULATION OF NET STATION VALUE						
TOWER COST ($M)	5.536		5.665		5.617	
+ PUMP COST ($M)	1.286		1.286		1.349	
+ EXHAUSTER COST ($M)	4.355		4.340		4.340	
+ TURBO-GEN COST ($M)	9.435		9.435		9.435	
= TOTAL CAPITAL COST ($M)		20.612		20.726		20.741
CUMULATIVE T/L OUTPUT (MW)	87.889		87.923		80.030	
- " PUMPING POWER (MW)	1.859		1.820		1.783	
- " EXHAUSTER POWER (MW)	4.659		4.600		4.629	
= NET STATION OUTPUT (MW)		81.371		81.503		81.618
NET OUTPUT CAPITALISED AT $2000/KW ($M)		162.742		163.006		163.237
- TOTAL CAPITAL COST						
= NET STATION VALUE ($M)		142.130		142.206		142.490

Table 3 is compiled from a number of runs based on the data above for the single flow rate case. It shows the effect of varying the condenser design parameters, TTD, GATD and ESL, the turbogenerator exhaust annulus area EAA and the exhauster intercooler temperature rise DELTI.

TTD (Terminal temperature difference) is the difference between turbine exhaust temperature and the mean temperature of the cooling water leaving the condenser. GATD (Gas approach temperature difference) is the difference between the re-cooled water temperature and the temperature of the non-condensible gas leaving the condenser. ESL (Exhauster suction loss) is the pressure difference between turbine exhaust and the non-condensible gas outlet.

The table show the extent of variation from the 'gn point assumptions, the value of the change, the components of the change.

5 CONCLUSIONS

The results indicate that at the value of power assumed ($2000/kW) it may be economic to use turbine exhaust pressures lower than in a number of recently constructed stations. The turbine manufacturer should be advised of the proportion of time the turbine is expected to operate at different exhaust conditions.

Both variable flow and fixed flow cooling water pumps should be called for *so* that a decision can be made when assessing tenders on the type of cooling water pumps to be used.

Table 2 shows that the condenser will work for a considerable time at low pressures and should be designed accordingly. Table 3 indicates significant benefits from reducing the terminal temperature difference, gas approach temperature difference and exhauster suction loss as much as possible. W. Hart in his "Final Report on Direct Contact Condenser Tests Carried out on the Ohaki/Broadlands Pilot Plant" shows that a minimum terminal temperature difference is advantageous for reasons of condensate chemistry. This is then one of the few occasions in engineering where one desired result does not work against another.

Table 3 also shows some significant advantage in increasing the temperature rise of intercooler cooling water.

The increase in value from an increase in turbine exhaust annulus is most marked and underlines the difficulty in balancing economic factors against practical and prudent design realities particularly in geothermal plant where stress orrosion problems are significant.

6. ACKNOWLEDGEMENTS

The State Electricity Commission of Victoria has given generous assistance in providing basic cooling tower design data.

The several firms that supplied preliminary design data and budget prices will be aware of their contribution, which is appreciated.

Discussions with engineers and scientists in the Meteorological Service, Department of Scientific and Industrial Research, Ministry of Works and Development and the Electricity Division of the Ministry of Energy have contributed essential elements to the study. The method of reducing the climate data to a frequency table *is* due to Mr D. Willis. Mr R. Broer evolved computer programmes as new requirements became apparent.

The author also wishes to thank the General Manager of the Electricity Division of the Ministry of Energy for permission to publish this paper.

TABLE 2

FRQ (CUM)	DB DEG	WB DEG	IND	RWT DEG	CR DEG	HWT DEG	TURBINE EXHAUST TEMF DEG	X PRES BAR	GAS PP BAR	OUTPUT (NET) MW	TOWER CUMECS	EVAP CUMECS	O/FLOW CUMECS	CONI) CUMECS	INTER CUMECS	GEN CUMECS	RE-INJ CUMECS
0.004	-4.0	-4.5	1.0	17.16	15.32	32.48	41.50	0.0819	0.0020	83.69	6.500	0.108	1.592	4.43'7	0.243	0.120	0.001
0.075	1.0	0.5	1.3	18.93	15.31	34.24	41.50	0.0019	0.0020	83.64	6.500	0.114	1.159	4.838	C.269	0.129	0.075
0.134	6.0	2.5	2.0	20.87	15.33	35.39	41.51	0.0820	0.0020	83.60	6.508	0.124	0.031	5.134	0.291	0.120	0.065
0.233	6.0	4.5	2.0	20.67	15.32	35.98	41.77	0.0831	0.0020	83.43	6.500	0.121	0.733	5.228	0.298	0.123	0.068
0.260	9.0	5.5	2.0	21.35	19.33	36.68	42.07	0.6944	0.0020	83.24	6.500	0.127	0.605	5.340	0.307	0.120	0.062
0.322	10.0	6.5	2.0	21.80	15.33	37.13	42.27	0.0855	0.0021	03.11	6.500	0.129	0.522	5.416	0.314	0.128	0.060
0.377	11.0	7.5	2.0	22.26	15.33	37.59	-42.48	0.8863	0.0021	82.98	6.500	0.130	0.433	5.496	0.321	0.120	0.059
0.462	11.0	8.5	2.0	22.60	15.33	37.93	42.63	0.0869	0.0021	82.88	6.500	0.129	0.370	5.555	0.326	0.120	0.060
0.500	10.0	9.5	2.0	22.83	15.33	38.14	42.73	0.0874	0.0021	09.32	6.500	0.125	0.329	5.576	0.330	0.120	0.064
0.526	14.0	8.5	2.0	23.03	15.34	38.37	42.82	0.0878	0.0021	02.74	6.500	0.136	0.270	5.634	0.333	0.128	0.053
0.597	15.0	9.5	2.0	23.52	15.34	38.86	43.04	0.0888	0.0021	82.57	6.500	0.137	0.177	5.724	0.341	0.120	0.052
0.667	16.0	10.5	3.0	1'4.01	15.35	37.36	43.27	0.0899	0.0022	82.43	6.508	0.135'	8.072	5.820	0.349	0.120	0.050
0.719	16.0	11.5	4.0	24.37	15.35	39.71	43.43	0.0905	0.0021	82.33	6.500	0.138	0.000	5.886	0.356	0.120	0.051
0.798	18.0	13.5	4.0	25.04	15.35	40.39	44.11	0.0930	0.0021	81.86	6.500	0.142	0.000	5.084	0.354	0.120	0.047
0.864	18.0	14.5	4.0	25.79	15.34	41 .13	44.85	0.0974	0.0022	01.31	6.500	0.141	0.000	5.888	0.352	0.120	0.048
0.931	18.0	16.5	4.0	26.60	15.33	41.94	45.65	0.1015	0.0023	80.66	6.500	0.140	0.000	5.893	0.348	0.120	0.049
1.000	26.0	19.5	4.0	29.13	15.36.	44.49	48.18	0.1153	0.0026	78.16	6.500	0.156	0.000	5.895	0.329	0.120	0.033

NOTE, IND = 1.0 INDICATES WETNESS LIMITED, 2.0 EXHAUSTER LIMITED AND 4.0 TOWER LIMITED MODE OF OPERATION.

TABLE 3

VARIABLE	TTD	GATD	ESL.	EAA	DELTI
'DESIGN' VALUE ASSUMED FOR STUDY	3.03 DEG	2.00 DEG	0.010 BAR	6.66 SQ M	7.00 DEG
VARIATION IN ASSUMED VALUE	1.00 "	1.00 "	0.002 "	0.56 "	1.00 "
	REDUCTION	REDUCTION	REDUCTION	INCREASE	INCREASE
INCREASE IN NET STATION VALUE PER TURBOGENERATOR	$130,000	$1 80,000	$180,000	$605,500	$21,500
COMPONENTS OF INCREASE IN VALUE					
INCREASED GENERATION	100.0%	70.3%	100.0%	100.0%	100.0%
REDUCED EXHAUSTER POWER		21.7%			