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GEOTHERMAL APPLICATIONS WITH GRAPE

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ANSALDO ENERGIA

ABSTRACT

To cope with the increasing market demands in terms of higher efficiency, reduced time to complete the job and lower costs, Ansaldo Energia has developed a pre-engineered standard plant for geothermal applications named GRAPE. The product line comprises a nominal 5, 20, 30, 55 and 110 MW plants which covers most of the needs of the present geothermal industry. The basic performances and components are reviewed: in particular the steam turbine, the condenser and the cooling circuit.

INTRODUCTION

Ansaldo has designed and manufactured about one third of the total world capacity in the geothermal field starting from the very early experiments by the Prince Ginori Conti in southern and central Tuscany at the beginning of the century.

In the process of standardizing the plants as much as possible in order to reduce costs, delivery time and improve the plant quality, Ansaldo has decided to develop a series of reference plants ranging from 4 to about 160 MW as shown in the following table:

TABLE 1
GRAPE LINE CHARACTERISTICS

GRAPE 5	4 - 7 MW
GRAPE 20	15 - 23 MW
GRAPE 30	25 - 35 MW
GRAPE 55	45 - 80 MW
GRAPE 110	90 -160 MW

The power range depends on the available field conditions although the reference conditions were established as follows:

TABLE 2
REFERENCE DATA

Inlet Pressure/Temp	6.5 bar/saturated
Exhaust Pressure	0.1 bar
Cooling Air Temp	21° C
Gas Content (weight)	1.5 %
Consumption (net)	~ 19,000 - 20,000 Kj/KW

However, as it will be shown later on, the plant design and in particular the turbine design is characterized by an operational phase so that it can accommodate quite different working conditions. The aim of the design was then to develop a plant with the following characteristics:

- high flexibility in the inlet conditions
- delivery time : 18 to 23 months
- cost of the electromechanical island (depending upon the size and extent of supply) : \$400 to \$700/KW.

GRAPE MAIN CHARACTERISTICS

In order to show what the basic characteristics of the GRAPE plants are, the most common one has been chosen and the characteristics have been summed up in Table 3. The characteristics of each component shall be reviewed in the following paragraphs.

TABLE 3
GRAPE 55 CHARACTERISTICS

Steam Turbine
Discharge Humidity : typically 10% up to 12%
Inlet Pressure (bar) : 5 to 9 (up to 20 w/ UP specials)
Outlet Pressure (bar) : 0.07 to 0.15 - Double Flow
Number of Stages : 6 to 9 with UP arrangement
Type of Stage : Mainly impulse
(possible partialization)
Dewatering : 20-35% efficiency per stage
Last Bucket : 23" and 26"

Condenser
Basic Option : contact type, compact
Alternate : surface, external condensation

Gas Extraction
Steam Ejectors : under 1% gas weight content
Liquid Ring Hybrid : 0.5% to 2.5%
Compressor : over 2%

Cooling
Cooling Towers : generally splash type crossflow to counterflow.

THE STEAM TURBINE

The steam turbine operates under the conditions specified in Table 3 although the reference conditions are as specified in Table 2. Basically the mass flow (typically 100 to 110 kg/s) for GRAPE 55 is proportional to the mass flow as stipulated by the so called Stodola Elliptical law (see/2.3/) hence:

(1) $$\frac{m\sqrt{T}}{P} = \text{constant}$$

where m is mass flow (kg/s)
T is absolute temp (K) and
P is pressure (bar)

The value of the constant is a function of the stage characteristics (in particular the first one), small changes of the first rotor (load factor) and possibly of the partialization which may accommodate changes in Stodola constant. In particular a typical effect is shown in Fig. 1 where a constant flow situation is examined.

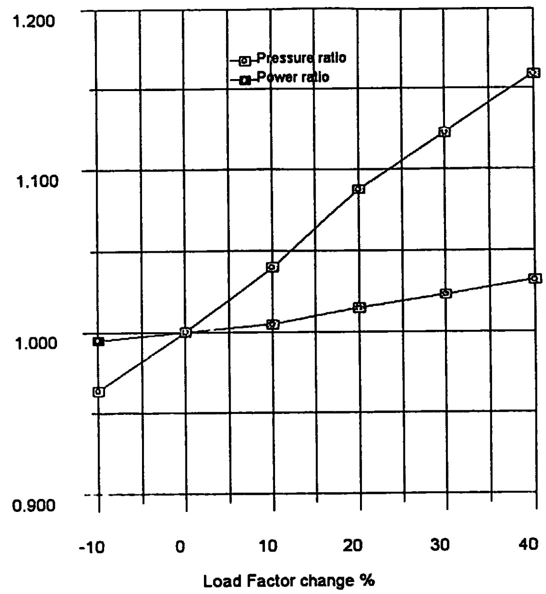

Figure 1. Relative Effect of First Stage Deflection Change

In the UP arrangement, a nine stage configuration is the basic one, however the slots for one to three stages can be left vacant to accommodate lesser pressure ratios. So far the design conditions have been examined and it has been shown that by a careful selection of the first stage characteristics (deflection angle and partialization) the turbine can accommodate different field conditions (without large changes in the efficiency which is however dominated by last stage losses, humidity removal and stage leak control). The extreme cases can be dealt within the UP arrangement. Changes however occur either in the field or in the turbine characteristics (e.g. due to scaling), which affects the flow area and hence the Stodola constant.

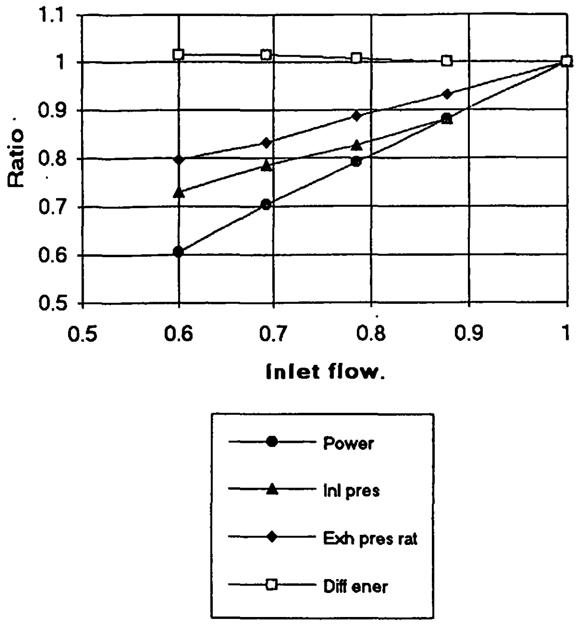

Figure 2. Effects of Inlet Flow

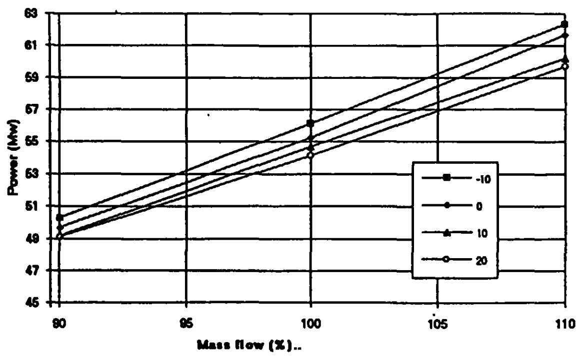

Figure 3. Effect of Part and Mass Flow on Power

The effect is clearly shown in a reference case in Fig. 2, where the effect of the mass flow (or inlet pressure) is depicted; or in Fig 3, where the effect of the change of partialization is also shown. Basically to reduce the power degradation consequent to these effects, the first stage partialization can be modified by means of an externally driven /4/ device in order to either increase or decrease the flow area to accommodate for scaling or field pressure fluctuations. This device however may be responsible for some efficiency losses in the first stage, which are more than compensated if the above circumstances prevail. Typically last stage buckets are the 23" and 26" blades. In the case of a 55 MW turbine, the larger one is recommended for vacuum values better than 0.09 to 0.10 bars. A comparison is schematically shown in Fig. 4.

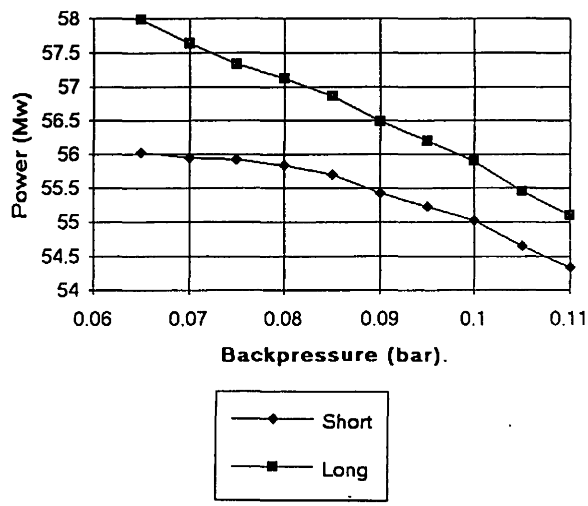

Figure 4. Backpressure and Bucket Influence Over Power

CONDENSER AND COOLING CIRCUIT DESIGN

As mentioned in Table 2, the basic option features the direct contact type of condenser, where the steam is directly quenched by the cooling water. In this arrangement it is essential to correctly evaluate the heat transfer between the steam and the cooling water drops. Most of the analyses so far have been centered upon the classical heat conduction theory (Fourier type analysis - see/5/), where the heat transfer efficiency is calculated as:

(2) $$Z = 1 - \sum_n (6/n^2.\Pi^2).\exp(-n^2.\Pi^2 + F)$$

where F(Fourier) is $F = a\frac{t}{r^2}$

and t = time, a = thermal difference, and r = drop radius

Further corrections may be introduced to account for the finite Biot number effects (see as an example ref. /7/). However comparisons with experimental data /8/ show that this correlation is by far too pessimistic /9/. Better agreement is found if one uses the spray fan jet theory or uses the corrective theory in ref. /6/, where the heat transfer efficiency is:

(3) $$Z = 1 - 3/8 . \sum_n A_n^2.\exp(-16.\mu_n.F)$$

$$A_1 = 1.32, A_2 = 0.73$$
$$\mu_1 = 1.678, \mu_2 = 9.83$$

Another important factor (as apparent from the definition of the Fourier number) is the drop of the average volume radius (sauter value) as shown in Fig. 5. In turn this value is dominated by the drop size distribution function (an example is shown in Fig. 6, from the characterization data of a particular nozzle).

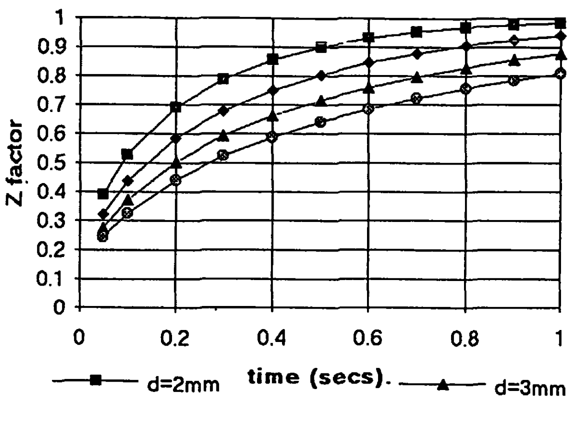

Figure 5. Fully Convective Model

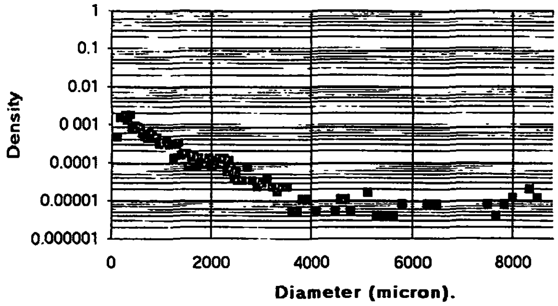

Figure 6. Typical Density Function

It should be also noted that the bigger drops even though quite uncommon may have some influence on the total sauter diameter. Since they are unstable, their influence can be neglected. The effect of the gas is small if the weight concentration is of the order of magnitude of a few percents, however the condensation tends to increase the gas weight fraction in the steam strongly degrading the heat transfer mechanism, then a counterflow trays stripping section is generally present at the end of the process. Globally the total efficiency of this type of condensers is about 80% to 90%, depending from the average gas fraction.

This relatively high efficiency can be somewhat difficult to achieve for a surface condenser, due to the presence of a few F/TTD (small values - say under 5°F might be difficult or very expensive to achieve). This fact and the cost are the main difficulties in the use of such type of arrangement. Another important point is the fact that theoretically the use of a surface condenser might lead to the use of a more efficient cooling tower. Geothermal plants generally use splashing type cooling towers, possibly with a cross flow type of arrangement. Such solution leads to very low values of the volumetric mass transfer coefficient (say less than 100 BTU/cuft.hr) and consequently to very big and expensive cooling towers. The use of a surface condenser might make possible the use of a more efficient type of filling, with a factor two on the efficiency of the mass transfer coefficient. However such arrangement needs a large quantity of make-up water (a substantial portion of the steam flow), which may not be easily available. Consequently the real advantage in the use of surface condenser is environmental, as it makes the effluents treatment easier (being limited to basically the off gas). The solutions are compared in the following table:

TABLE 4
COMPARISON BETWEEN SURFACE AND DIRECT CONDENSERS

	Surface Condensers	Direct Condensers
Efficiency %	70-80	80-90
Relative Cost	~ 2 - 3	1
Cool. Towers	0.5-1.0*	1
Environment	better	

* depending on the availability and quality of makeup water

EXTRACTION SYSTEM

The basic extraction system for the non condensable and low quantity gases is a three steam ejector with intermediate condensers. This solution is very economical, however a consume of about 3.0 kg of incondensables is required. This strongly affects the efficiency of the system as one kg/sec delivers about 500 KW of power. To the other extreme the energy consumption for a compressor is indicating about 500 KW/kg/sec gas), i.e. much less. In between there is the hybrid system where the ejectors are used to raise the pressure by a factor of say three (depending from cooling water temperature) and then the liquid ring pump takes over. Typically the steam consumption is about 1.0 to 1.5 kg steam/kg gas and the energy consumption about 250 KW/(kg/sec gas). The three solutions are compared in Table 5.

TABLE 5
EXTRACTION SYSTEM CONSUMPTION COMPARISON
(per kg/sec of incondensables)

	Steam Kg/s	Energy KW	Total KW*	Cost
Ejector	3.0	0	~ 1500	low
Hybrid	1.25	250	~ 900	med.
Comp.	0	500	500	high

* based on 500 KW/(kg/sec steam)

The results clearly show that the compressor solution is the best from the energetic point of view and it is the one to be preferred if the amount of gas is large or the energy cost gives the data already discussed in the Table 2.

CONCLUSIONS

GRAPE is a family of standardized geothermal plants with reference conditions of fluid, but where provisions have been made in order to encompass a variety of conditions typical of most of the field available (with the exception of low energy fields where new developments are under way). The basic characteristics of the plants have been described aiming at the following basic goals (for the 55 MW unit under reference condition):

- efficiency : better than 19000 to 20000 Kj/Kwh (net)
- cost : better than $500/KW (electromechanical only)
- delivery : better than 21 months

Different architectural schemes have been discussed and comparisons have been made.

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