Proceedings World Geothermal Congress 2005
Antalya, Turkey, 24-29 April 2005

Binary Systems with Geothermal Fluid Pressurization to Avoid Flashing: Energy Evaluation of Down-hole Pump Cooling below Geothermal Fluid Temperature

Paola Bombarda, Mario Gaia

Dipartimento di Energetica, Politecnico di Milano, Piazza Leonardo da Vinci 32, 20133 Milano, Italy

paola.bombarda@polimi.it; mario.gaia@polimi.it;

Keywords: Binary power plant, submersible down-hole pump, fluid re-injection, wellbore scaling, medium-high geothermal fluid temperature.

ABSTRACT

The pressurization of the liquid geothermal fluid at a suitable depth in the well, by means of an electric powered pump, is an established technique to avoid flashing in the well. However this technique is limited by the maximum temperature which can be sustained by the windings of the pump motor. In the present paper, the basic feasibility and energy cost of cooling the windings are evaluated. The proposed solution consists of a cooling loop made by an evaporator fed with water, which removes the heat generated in the motor windings, a downward water pipe, which feeds the evaporator, an upward steam pipe which removes the evaporated steam and brings it at well head and a condenser which rejects the removed heat into the ambient; a pump is also needed in order to pressurize the cooling loop and avoid evaporation in the downward water pipe. The size of the pipes to be put into the well for the cooling system is preliminarily dealt with: when feasible, this technique can broaden the application range of the binary cycle technology. Concerns about the environmental impact and reservoir exploitation point out that the adoption of the binary cycle is particularly advantageous in many aspects: it is well known that emissions in the atmosphere of harmful or undesired components (for example: H_2S, CO_2) are to be avoided: in binary power plants the all-liquid geothermal brine can be easily re-injected after use. This solution therefore gives the chance to treat the noncondensables gases as dissolved in the liquid brine instead of as gasified in the vapor phase. Re-injection has also a beneficial effect on the reservoir exploitation.

1. INTRODUCTION

The pressurization of the liquid geothermal fluid at a suitable depth in the well, by means of an electric powered pump, is an established technique to avoid flashing in the well. Though in this case external mechanical power is used instead of the natural drift of the steam in the upward flow, the use of a pump can assure a constant discharge flow at sufficient pressure to avoid flashing and to strongly reduce wellbore scaling problems, which could otherwise occur after flashing. This technique is limited by the maximum temperature which can be sustained by the windings of the pump motor.

Geothermal power plant configuration depends strictly on several factors: the most important are the geothermal brine characteristics and the site features. Although complex power plant configurations are possible (see for example Bombarda and Macchi, (2000)), the geothermal power generation systems are mostly based on conventional steam technology; besides this, the binary cycle technology is applied for the exploitation of medium-low temperature liquid–phase geothermal resources.

The geothermal fluid is normally a mixture of liquid, vapor and noncondensable gases, with relative fractions depending on site characteristics. As already stated, the adoption of the binary cycle is usually considered for liquid brine. If the steam quantity is not negligible, the application of the binary cycle is still possible, but much more complicated; in this situation the flash cycle is often preferred. The mass fraction of noncondensables, mostly CO_2, is also an important parameter: a high noncondensable content would suggest adopting a steam turbine with open cycle, i.e. with geothermal fluid release in the ambient at the end of the expansion. The noncondensable gases in fact act as a supplementary thermal resistance in the condensation process, and a non-condensable gas compressor is required to eliminate the gases for a proper condenser operation: the compression power absorbed increases therefore with noncondensable content and it becomes much greater if gas re-injection is considered. However, open cycle is not favored because it is well known that CO_2 emissions are to be maintained as low as possible due to greenhouse effect and other substances which may be contained in the geothermal fluid (H_2S for example) are also harmful to the ambient.

Concern about the environmental impact and reservoir exploitation points therefore out that the adoption of the binary cycle is particularly advantageous in many aspects: in binary power plants the all liquid geothermal brine may be straightforwardly re-injected after use. This fact has also a beneficial effect on the reservoir exploitation.

In order to apply the binary cycle technology with subsequent brine re-injection for the case of two phase brine at well head, different options may be considered:

- the first option is simple: the liquid and the gaseous content of the brine are split in two separate flows which feed two parallel sequences of Organic Rankine Cycle heat exchangers; in this case, however, the vapor condensation occurs and the problems caused by noncondensables presence are encountered, as mentioned. A single turbine and a single pump are then considered to complete the cycle.

- an interesting option consists in the pressurization of the brine so as to maintain it in the liquid state up to the power plant outlet: a pump is in this case placed at convenient depth in the well. On the suction side, the pump receives the geothermal fluid at the local reservoir conditions (all liquid, temperature and pressure depending on reservoir characteristics) and it discharges it

at a greater pressure, sufficient to hinder the flash process which would otherwise occur during the upward flow of the brine inside the well. This solution is practically applicable only if the geothermal fluid temperature is low or, alternatively, if the pump is conveniently cooled.

The former solution, though complicated, is easily obtainable with the state of the art technology; the latter solution is less conventional, but, when feasible, it allows the adoption of a much simpler and more compact Organic Rankine Cycle, and gives the chance to treat the noncondensables gases as dissolved in the liquid brine instead of as gasified in the vapor phase.

It is generally expected that the adoption of a down-hole submersible pump can boost the application of binary cycle technology, and ensure the exploitation of low temperature liquid – phase geothermal resources too, thus significantly increasing geothermal power generation potential. If the motor is properly cooled, the application of binary cycle technology can be extended also to medium-high temperature resources, both in situations in which the flash cycle is not appealing (see the Salavatli example hereafter) and/or when environmental concern suggests the adoption of an almost zero emission plant instead of a flash plant. The problem of cooling the motor winding is therefore of remarkable interest.

2. THE DOWN-HOLE PUMP

The down-hole pump is nowadays seen as a key component for the extensive developing of binary power plant technology. Several studies and research programs are therefore focused at developing such pumps.

2.1Current technology

An important research program has been conducted by NEDO, as described by S. Ichikawa, H. Yasuga, T. Tosha, H. Karasawa (2000). Several pump configurations were conceived at beginning of the study but it was concluded that the most suitable version is the classical one realized with an impeller driven by an electric motor. Alternatives were considered in order to possibly avoid the coupling with an electric motor down-hole: water or steam turbine driven pumps and line-shaft solution, with motor outside the well were however regarded as less promising solutions. The positioning of the pump-motor in the well is imagined at a depth lying between 300 and 600 m, in order to process geothermal water with a maximum temperature of 200°C. Materials selected are high nickel stainless steel, cobalt-base alloys, hard metal coated (material thermal resistance is of the utmost importance). The authors declare that the motor is cooled, though they do not give any information about the cooling system. Electric submersible power cables are covered with metal sheath to prevent water invasion into the cables.

The U.S. Department of Energy has supported research on down-hole pumps as well: survey results are reported by Pritchett (2000) and, with more details, by Pritchett (1999). A comparison between line-shaft and submersible pumps is outlined: according to this study, in a conventional plant, with a well internal diameter equal or greater than 250 mm, a line-shaft pump provides somewhat better fluid delivery capacity; but for smaller internal diameters submersible pumps provide higher fluid discharge rates and are applicable with higher reservoir temperature. Line-shaft technology is seen as mature and static, while submersible pump technology has reported to have achieved significant advances and is expected to have more in the future. Submersible pumps could allow also the diffusion of small relatively slim hole geothermal power plants (typically 100 to 1000 kW capacity), which could be a promising application for distributed power generation. Following Pritchett, limits exist for the well internal diameter if a pump is to be used: if the well is very small (75mm diameter) no down-hole pump may be applied at the moment. With a 150 mm or larger internal diameter a down-hole submersible pump may be used; with even bigger diameter (standard production wells, 300 mm) a line-shaft pump may be adopted. Until quite recently, virtually all geothermal application of down-hole pumps have involved line-shaft pumps, but in the last few years the possibilities of submersible pumps have started to be explored. According to Pritchett, line-shaft pumps operate usually at 1800 rpm, while submersible pumps can operate at 3600 rpm; that means that for a considered impeller the specific work will be higher for the submersible pump than for a line-shaft one. As a final point, the submersible pump is again seen as more promising with respect to the line-shaft pump: its application is however limited by the maximum temperature allowable for the geothermal fluid elaborated by the pump. According to the materials and the technology used in the pump, different limits exist: Pritchett reports that the best pumps can work at the moment with geothermal fluids up to about 205 °C; the cooling motor problem is a matter of crucial interest. When no reference is made to re-injection, the advantage of the down-hole pump + binary cycle plant in respect of flash plants depends on the reservoir features, being the reservoir temperature the most important one. Generally Pritchett states that the down-hole pumps are advantageous compared to self-discharge wells below 200°C for standard production wells and below 160°C for slim holes.

2.2 Choice of the submersible pump location and pump head

Vapour formation must be avoided both at pump intake, so as to prevent the pump from being damaged by cavitation (the formation of vapour bubbles which collapse afterwards causing impeller blade erosion), and in the fluid column flowing upwards to well head, so as to avoid scaling of the pipe.

From the former condition pump position is determined; from the latter condition pump head is evaluated. In both case it is assumed that

$$p \geq p_{bubble} + 1 \qquad (1)$$

where all values are in *bar*, p is the local pressure, p_{bubble} is the bubble point pressure, i.e. the pressure at which a bubble of vapour is formed, and, following Pritchett, 1 *bar* has to be added as a safety margin which allows for minor fluctuations.

If dissolved CO_2 is present in the geothermal fluid, care must be taken in considering the partial pressure of CO_2 in the liquid-vapour equilibrium, and p_{bubble} must be properly evaluated. Note that p_{bubble} increases with temperature and CO_2 content, and this fact has a strong influence on pump position and pump head.

2.2.1 Pump location

The pump suction pressure p_s will be given by

$$p_s = p_r - \rho g(D - h) - \Delta p_{wf-pi} \qquad (2)$$

where p_r, ρ, g, D, h, $\Delta p_{wf\text{-}pi}$, are respectively reservoir pressure, geothermal fluid density, gravitational acceleration, well depth, pump depth, and fluid flow friction pressure drop from well feeding to pump intake. By calculating $\Delta p_{wf\text{-}pi}$ and imposing the relation (1) at pump suction, pump depth can be evaluated. For the sake of simplicity, no reference was made to the pump NPSH (Net Positive Suction Head), which is strongly dependent on the particular impeller design and may vary greatly from pump to pump: if NPSH is to be accounted for, pump location will be somewhat deeper than the one found by means of the simplified procedure. From (1) it is also clear that if the geothermal fluid temperatures or CO_2 content increases, the pump depth increases as well.

2.2.2 Pump head

Imposing that delivery pump pressure must be enough to guarantee that at wellhead the relation (1) is still valid, i.e. that single-phase liquid state is maintained up to the power unit, and detracting the pump suction pressure, the total pump pressure increase is easily found:

$$\Delta p_{pump} = \rho g D + \Delta p_{wf-wh} + p_{bubble} + 1 - p_r \qquad (3)$$

Pump pressure increase must therefore be enough to balance the gravitational pressure drop, the fluid flow friction pressure drop $\Delta p_{wf\text{-}wh}$ from well feeding to well head and to maintain the fluid in the liquid phase; the reservoir pressure is detracted from pump pressure increase. For the sake of simplicity the ORC heat exchanger pressure drop is not considered, but when added it would slightly modify the total pump pressure drop having little impact on final result.

2.3 Power consumption

Once mass flow rate and pump head are known, pump hydraulic power consumption is easily calculated by considering hydraulic efficiency and the electric power consumption is then evaluated by considering shaft and electric motor efficiency. If the stage works at its optimum volumetric flow rate, the hydraulic efficiency can be quite high: both Pritchett (2000) and Ichikawa et al. (2000) report efficiencies in the range 0.74-0.8.

3. COOLING LOOP

As reported before, depending on materials and technology involved, for every submersible pump there is a maximum tolerable geothermal fluid temperature which may reach 200°C, or may be even lower if standard components are adopted. No matter the value of the limit, it is important to underline that a limit exists above which this simple submersible pump is no longer usable. To broaden its utilization range, a system which can take away the heat generated by the electric motor must be employed.

The adoption of a whole separate conventional refrigerating machine to cool the windings is not strictly required: moreover, conventional refrigerating machine utilize as refrigerant fluid halogenated hydrocarbons or hydrocarbons which have normally an operating temperature range lower than the one required in this application.

It is then necessary to split the ideal refrigerating machine in its basic components, to choose a proper thermal carrier fluid and reassemble the refrigerating machine on an "*ad hoc*" basis. For the present study, the selected thermal carrier fluid is pure water: it evaporates in the right range of temperature, it has very favourable transport characteristics, and its use is not restricted by law. The evaporator, which is the component where heat removal occurs, constitutes a supplementary motor case, such that the liquid water is circulated between the motor envelope and the evaporator case. If the water is circulated at convenient pressure and temperature, it evaporates, thus efficiently removing heat and cooling the windings. To permit water circulation in the evaporator, a feeding water tube and a discharge steam tube going from the evaporator to wellhead are connected to the evaporator. At wellhead the loop is closed by means of a condenser, which rejects the heat removed from the windings into the ambient and provides liquid water to the feeding downward tube. Two components, which are always present in a refrigeration cycle, are still to be discussed: an expansion valve, and a compression device. The throttling valve must be located at the evaporator inlet, in order to take the cooling water to the right evaporator operating pressure; the compression device, which is usually a gas compressor, is on the contrary not present at the evaporator outlet: compression is in this case operated in liquid phase with a pump placed after the condenser. Overall, the proposed device can be assimilated to a heat pipe, rather then a refrigeration unit. During operation, cooling water must be maintained in liquid phase when flowing downward: being surrounded by geothermal fluid flow, it will soon reach geothermal fluid temperature; if it is assumed that water reaches immediately this temperature, this means that pressure at tube inlet must be greater than or at least equal to the saturation pressure corresponding to geothermal fluid temperature. Actually, the system can operate with inlet tube pressure exactly equal to this saturation pressure: when flowing downwards, the gravitational pressure drop acts as an additional safety margin. The heat removal and hence the evaporation must occur at the maximum motor winding temperature considered $T_{mw.}$ Consequently the pressure must be equal to the saturation pressure corresponding to T_{mw}, which is easily obtained by means of a calibrated flash process in the throttling valve. In the upward flow, vapour flow is maintained thanks to evaporation pressure and a proper tube sizing.

With this cooling device, the application of the submersible pump can be extended over a wider application range and the opportunity of the binary cycle may be investigated over a broader range of geothermal fluid conditions.

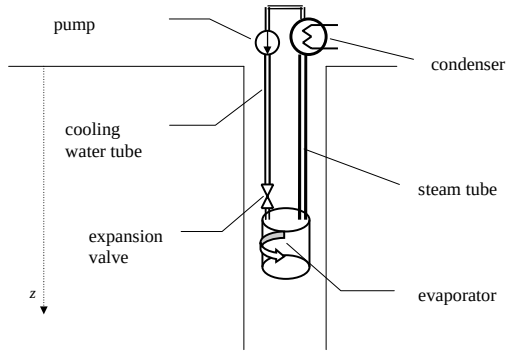

Figure 1: Cooling loop: two phase water is circulated around the motor case; it evaporates thus removing heat. Generated steam is removed via a discharge tube; heat removed is transferred to the ambient via a condenser at ground level.

4. PERFORMANCE EVALUATION

The cooling system described was sized and calculated for a test case, which refers to a particular geothermal well, and

subsequently systematically studied by resizing it for extended operating conditions over a wide range of geothermal source temperatures.

4.1 Basic assumptions

The down-hole pump is operated at a flow rate corresponding to maximum hydraulic efficiency; no limitation on the pump maximum ΔP is assumed (it is therefore assumed that the number of stages can be varied without any constraint); no limitation on the pump maximum power is considered (the pump is refrigerated and it is assumed that pump design can be modified so that no problems with mechanical resistance arise). Power consumption and dissipated heat to be removed are calculated by assuming values for pump hydraulic efficiency, shaft efficiency and motor electric efficiency corresponding to state of the art technology. Values considered are reported in Table 1.

The maximum temperature of motor casing containing the winding (T_{mw}) is regarded as a parameter variable between 140°C and the source temperature; though the best materials and technology allow higher temperatures, the analysis concerns also more conventional motor windings and thus the lower limit is set as low as 140°C; the higher limit is not really fixed, being the technology in continuous progress, and the analysis is therefore extended up to a value equal to the geothermal source temperature (actually in this case it would be sufficient to position the pump in the flow to have it cooled, without any evaporation process).

Downward flow: cooling water temperature is assumed constant in the downward flow, with the same value of the reservoir temperature; the pressure is on the contrary variable with position (vertical coordinate *z*) according to gravitational head and friction pressure loss:

$$p(z) = p_{pd} + \rho g z - \lambda \rho \frac{v^2}{2} \frac{z}{d_w} \qquad (4)$$

where p_{pd} is the pressure at pump delivery (which correspond to saturation pressure at geothermal fluid temperature), and d_w is the cooling water tube diameter. At throttling valve inlet, pressure is obtained from (4) for *z*=pump position. In all calculations the liquid water velocity is set to 1.5 m/s.

Upward flow: in the upward flow, no steam velocity is assumed *a priori*: steam velocity (and consequently steam tube diameter) is found by imposing (eq.5) which states that initial pressure (evaporation pressure) compensates for friction loss and gravitational head, leaving final steam pressure at condenser inlet greater than or equal to atmospheric pressure. (The cooling loop must not reach sub-atmospheric pressure in order to avoid problems with air infiltration).

$$\lambda \rho \frac{v^2}{2} \frac{h}{d_v} = p_{sat} - \rho g h - 1 \qquad (5)$$

where h is pump depth and d_v is vapour tube diameter. The pressure drop of condenser, which is not expressly accounted for, does not alter the procedure and would only result in a slightly larger pipe diameter.

Reservoir and production well: following the test case, the same reservoir features and production well were assumed for the parametric analysis: reservoir pressure is estimated at 132 *bar*, well depth is 1400 *m*, and well diameter is 250 *mm*. CO_2 content, is assumed as high as 1%, which is a value representative for geothermal resources with medium-high carbon dioxide content.

Table 1: Down-hole pump and well major assumptions.

Major assumptions		
pump hydraulic efficiency		0.75
shaft efficiency		0.96
motor efficiency		0.90
geothermal discharge rate	m^3/s	0.077
water velocity	*m/s*	1.5
minimum pressure in the cooling loop	*bar*	1
reservoir pressure	*bar*	132
well depth	*m*	1400
well diameter	*mm*	250
CO_2 content	*%*	1

4.2 Calculation results

The test case proofed encouraging results as far as the feasibility of the cooling solution is concerned; the calculations were thus extended to parametric analysis.

4.2.1 Case test: Salavatli field

Salavatli geothermal field is located in a promising area in Turkey, in the region of the Menderes Massif. Two wells AS-1 and AS-2, were drilled in the 80's; the test case in the present paper refers to the first one, AS-1. Geothermal fluid characteristics, reported by Serpen and Tüfekçioglu (2003), are shown in Table 2.

Table 2: Characteristics of Salavatli test case.

Salavatli well AS-1		
mass flow	*kg/s*	69.44
fluid temperature	*°C*	169.5
CO_2 content	*%*	1.2
bubble pressure	*bar*	38.5
boron content	*mg/l*	40-60

The CO_2 content has a value slightly higher than that assumed in the parametric analysis: it is representative of geothermal fields in Turkey, and it is a relatively high value: this means that severe scaling problems occur if a flash process takes place. Boron content is also very high and this implies that re-injection must be performed in order to avoid ambient pollution. Both aspects point out the adoption of a binary cycle with down-hole pump and re-injection as a very interesting technical solution. Down-hole

pump characteristics calculated on the basis of values reported in Table 2 are shown in Table 3.

Table 3: Down-hole pump characteristics.

Salavatli down-hole pump		
Depth	*m*	350
pump head	*bar*	32
power consumption	*MW*	0.367

In this example, the maximum allowable temperature for the motor casing is considered as low as 140°C, in order to evaluate the feasibility of the solution for a conventional down-hole pump. Main cooling loop features are reported in Table 4.

Table 4: Main Salavatli cooling loop features.

Salavatli cooling loop			
heat removal		*kW*	50
cooling flow		*kg/s*	0.025
pressure at pump delivery		*bar*	7.8
pressure at the liquid expansion valve inlet		*bar*	20.6
evap. temperature		*°C*	140
evap. pressure		*bar*	3.61
		downward flow	upward flow
diameter	*mm*	4.8	24
velocity	*m/s*	1.5	28
friction Δp	*bar*	18	2.5
gravitation.Δp	*bar*	30.8	0.07

It can be seen that section required by cooling system tubes is very small, and the tubes can be easily accommodated in the main well pipe without a large reduction in the free flow area. The highest pressure in the cooling loop is reached at throttling valve inlet, and it is about 20 bar.

Binary plant power produced is calculated by means of the model described in Bombarda and Macchi (2000), assuming isobutane as the working fluid, saturated cycle and air cooled condenser. As a final result the following power balance is obtained (see Table 5).

Table 5: Salavatli power plant balance.

Salavatli power plant		
binary cycle power	*MW*	3.37
down-hole pump consumption	*MW*	0.37
net power	*MW*	3.00

Binary cycle power calculated, which correspond to 44.3% of the power which would be produced by an ideal reversible cycle operating with the given geothermal fluid, is representative of the electric power obtainable with a standard solution: higher values may be obtained with more complicated cycle configurations.

4.2.2 Parametric analysis

In the parametric analysis both the geothermal fluid temperature and maximum motor winding parameter are considered as variable. Calculation results are presented in the following figures, which give respectively the pump position, the cooling mass flow, the cooling loop diameters, the cooling loop pressures and the net power produced as a function of the geothermal source temperature:

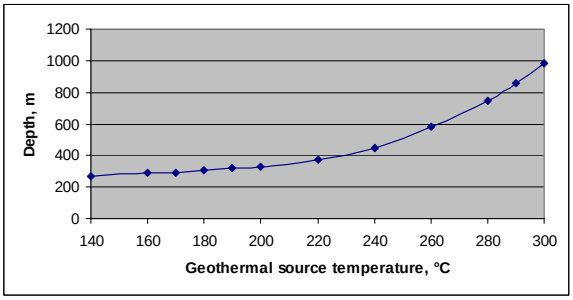

Figure 2: Down-hole pump position in the well for a well of 1400 m depth and 132 bar reservoir pressure.

It is clear (see Eq. 2) that pump depth depends only on geothermal source temperature and CO_2 content.

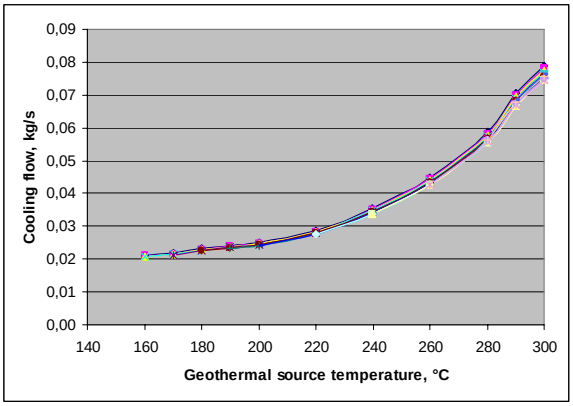

Figure 3: Cooling flow required for the motor windings as a function of geothermal source temperature at different maximum winding temperature.

As the curve corresponding to different value of T_{mw} are all overlapped Figure 3 shows that T_{mw} has no practical influence on the cooling flow required. The useful enthalpy drop decreases with increasing geothermal source temperature, hence the cooling mass flow increases with the temperature of the geothermal source.

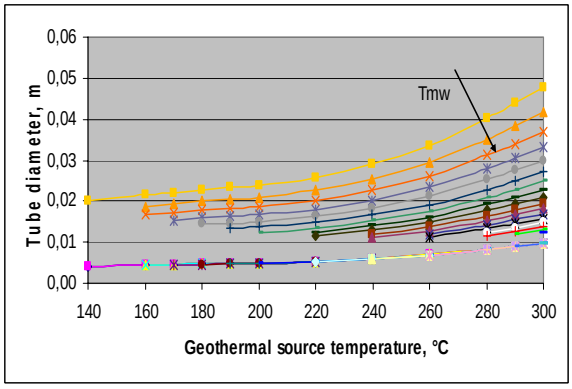

Figure 4: Cooling loop diameters as a function of geothermal source temperature at different maximum casing temperature.

Cooling loop diameters are in all cases small enough to be considered not relevant as far as cross sectional area in the main flow of the well in concerned. As the cooling mass flow increases with the geothermal source temperature, tube diameters increase as well with the geothermal source temperature. Water tube diameter is nearly constant with T_{mw},; the steam tube diameter is on the contrary decreasing when motor winding temperature is increasing (evaporation occurs at a higher pressure and steam flow is more compressed).

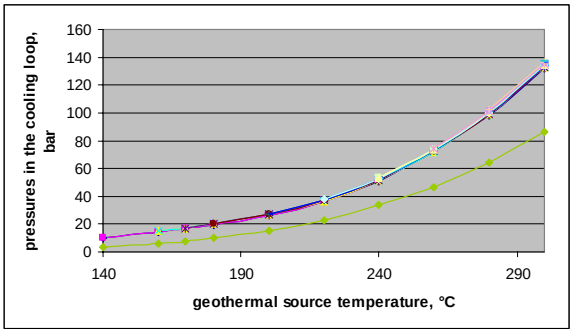

Figure 5: Pressures in the cooling loop.

Figure 5 is representative of the operating pressures in the cooling loop: the upper curve represents the highest loop pressure, which is located at throttling valve inlet and has no practical dependence on the maximum windings temperature, the lower curve represents pressure at downward tube inlet. Note that the highest loop pressure increases almost exponentially with the geothermal source temperature.

The power consumption of loop pump, though greatly variable with maximum loop pressure, is in all cases low because the cooling mass flow is always low. Values range from a few W up to 1.5 kW.

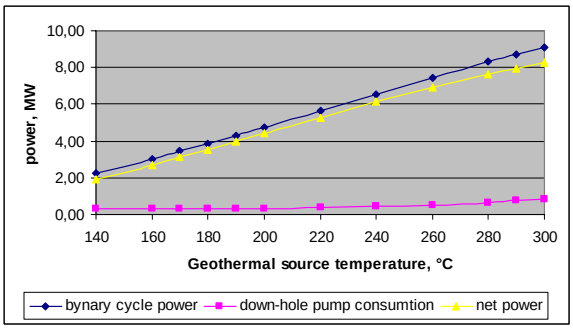

Figure 6: Binary cycle electric power, down-hole pump power consumption and net power produced.

Figure 6 summarizes the plant performance: binary cycle electric power produced is estimated by decreasing the reversible cycle efficiency by a factor which is set equal to 45%. It is easily seen that net power produced increases with geothermal fluid temperature.

5 CONCLUSIONS

The calculations performed show that an external cooling of the down-hole pump motor is feasible, at least from the energy consumption and from the well occupation point of view. The solution proposed consists of a pressurized water loop with an evaporator and a condenser. The most important variables for the loop design are the geothermal source temperature and the CO_2 content; the maximum motor winding temperature has only a minor influence. The energy cost of operating the cooling loop is negligible compared to down-hole pump consumption.

The convenience of the proposed solution is verified over a wide range of geothermal fluid temperatures and shows that the binary cycle solution could be a profitable solution even for medium–high temperature which are usually typical for flash cycle: moreover this solution may allow field exploitation in cases where, because of ambient pollution and scaling problems, flash cycle technology cannot be adopted.

NOMENCLATURE

d	tube diameter	*mm*
D	well depth	*m*
g	gravitational acceleration	m/s^2
h	pump depth	*m*
p	pressure	*bar*
v	velocity	*m/s*
λ	friction factor	
ρ	fluid density	kg/m^3

Subscripts

mw	motor winding
pd	pump delivery
pi	pump intake
r	reservoir
s	suction
sat	saturation
wf	well feeding

REFERENCES

Bombarda, P.., and Macchi, E.: Optimum cycles for Geothermal Power Plants, *Proceedings* World Geothermal Congress 2000, Tokyo, Japan (2000)

Ichikawa, S., Yasuga H., Tosha T., Karasawa H.: Development of Downhole Pump for Binary Cycle Power generation using Geothermal Water, *Proceedings* World Geothermal Congress 2000, Tokyo, Japan (2000)

Pritchett, J.W.: Electrical Generating Capacities of Geothermal Slim Holes, *Proceedings* World Geothermal Congress 2000, Tokyo, Japan (2000)

Pritchett, J.W.: Electrical Generating Capacities of Geothermal Slim Holes, *Final Report*, DOE/ID/13455, (1999)

Serpen, U., Tüfekçioglu, H.: Developments in Salavatli Geothermal Field of Turkey, *European Geothermal Conference 2003*, Szeged, Hungary (2003)