THE UNITED NATIONS UNIVERSITY

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LECTURES ON GEOTHERMAL ENGINEERING

Russell James

Geothermal Training Programme
Reykjavík, Iceland
Report 13, 1986

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PREFACE

From the start of the UNU Geothermal Training Programme in 1979, it has been customary each year to invite a geothermal expert to Reykjavík as a Visiting Lecturer. The lecturers have stayed at the Geothermal Training Programme from two to eight weeks. During this time they give a one-week lecture series on their speciality, and have discussion sessions with the Fellows attending the Training Programme. The lecture series are open to the geothermal community in Iceland.

The Visiting Lecturers have added an extra dimension to what the UNU Geothermal Training Programme can offer to its Fellows. It has also been an important opportunity for the Training Programme to contribute new understanding to the geothermal engineers and scientists in Iceland, through the lecture series and discussions with a distinguished expert from another country. The following geothermalists have been Visiting Lectures at the UNU Geothermal Training Programme:

1979	Donald E. White	United States
1980	H. Christopher H. Armstead	United Kingdom
1981	Derek H. Freeston	New Zealand
1982	Stanley H. Ward	United States
1983	Patrick Browne	New Zealand
1984	Enrico Barbier	Italy
1985	Bernardo S. Tolentino	Philippines
1986	C. Russell James	New Zealand

This year's Visiting Lecturer was Russell James, Geothermal Research Engineer, D.S.I.R. at Wairakei, New Zealand. About twenty-five years ago he figured-out how to measure the output of two-phase geothermal wells. He calls it Lip Pressure; the Icelandic term is "aðferð" (Method) Russell James; in the United States they say James Tube. James' 1962 paper on "Steam-Water Critical Flow Through Pipes" is probably the most useful geothermal paper published in the literature. Russell James has been highly productive in his carreer as a geothermal engineer, as evident in his many writings.

The present report consists of five papers that formed the basis of Russell James' lecture series in Reykjavik, September 22-26, 1986. The papers are on the subject of steam and water flow in geothermal wells and pipelines - the very subject Russell James has contributed to the most.

Jón-Steinar Gudmundsson
Director
Geothermal Training Programme

MAXIMUM DISCHARGE OF GEOTHERMAL WELLS

Russell James
DSIR, Wairakei, Taupo, New Zealand.

ABSTRACT

We cannot tell how "good" a well is unless we can estimate the maximum flow possible under such ideal conditions as complete permeability at the production horizon and boiling point throughout the depth of the reservoir. Calculated Lip pressures for vertical wide-open discharge under these conditions are surprisingly independent of the kind of fluid tapped by the well, whether dry saturated steam or saturated hot water.

The status of an actual well can be established by comparing the measured Lip pressure with the calculated theoretical maximum.

Discharges are simply determined from the values of Lip pressure and supply fluid enthalpy.

INTRODUCTION

Some years ago, I was working on a Wairakei well which tapped a supply of saturated hot water (water at the boiling point for its pressure). During a period of about a month, the fluid changed to dry saturated steam at the same temperature and pressure, but no change was observed in the Lip pressure attached to the vertically discharging well when blown wide-open. Of course, the actual flow-rate would have decreased considerably as this is related to Lip pressure and fluid enthalpy in the following equation, James (1962).

$$G = \frac{1839 \; P_c^{0.96}}{h_o^{1.102}} \qquad (1)$$

G Flow, t/m^2s

P_c Lip pressure, bars

h_o Fluid enthalpy, kJ/kg

Obviously the driving force and controlling factor was the presence of the compressible vapour phase, with the water being merely dragged along as a passenger; at least that was the superficial hypothesis advanced at the time. It would, of course, be extremely difficult to experimentally verify this phenomenon over a range of well depths and fluid temperatures and types. Hence, the approach undertaken here is to calculate Lip pressures over a range of well depths and bore diameters for (a) dry saturated steam, and (b) saturated hot water. This is accomplished specifically for the condition shown in Figure 1 where the well is discharged wide-open vertically and where supply horizon permeability is considered as perfect with no restriction on flow into the well at depth.

As an unmanageable mix of well depths and bottom hole conditions is possible to envisage, it was decided to simplify matters by imposing a relationship between these factors. Fortunately, such a relationship exists in practice as it appears that geothermal reservoirs either are, or tend towards Boiling Point with Depth (BPD), so that both pressure and temperature increase progressively with depth from the ground surface, often down to a so-called Base temperature. Over the depth at which BPD obtains, boiling water and steam co-exist, and depending on the permeability-porosity of the rocks, the well may draw either of these fluids from the supply horizon or even a mixture of both. When supplied with saturated hot water, steam generation (flashing) starts immediately and continues as the fluid ascends to the wellhead. Supplementary steam from the rock matrix may increase the fluid enthalpy above that expected from the horizon water temperature but we shall only consider the extreme conditions here, of all-water or all-steam entering the well.

For the case where water is boiling at the ground surface at 100°C and at greater temperatures at depth due to the increasing hydrostatic head implsed by boiling water, we have the following equation derived by James (1970) and in the metric form:

$$C = 69.56\ H^{0.2085} \quad \text{for } 30 < H < 3000 \qquad (2)$$

C Reservoir temperature, °Celcius

H Depth in metres

So for any particular depth of well, we may take the supply fluid temperature from the above equation and hence obtain from published Steam Tables, the associated pressure, specific volumes of steam and water as well as enthalpies and other data.

DRY SATURATED STEAM CALCULATION

Lapple (1943) theoretically estimated the flow of compressible fluid through long pipes to the atmosphere and this was later experimentally confirmed by James (1964) specifically for dry saturated steam. The method of calculation is given in detail by James (1970) where charts are presented of flow, viscosity and specific volume together with formulas to estimate Reynold Numbers of flows and friction factors in commercial steel pipes.

The approach is to select a temperature, say 250°C and, from Steam Tables, obtain the steam pressure of 39.73 bars, and from equation (2) the depth of 462 m. The steam flow to atmosphere is now calculated and converted to Lip pressure employing equation (1) in which the steam enthalpy h_o = 2801.5 kJ/kg at 250°. A trial method is required after initial guessing of the friction factor, and assumption of a well bore diameter. Results are charted on Table 1 against $(P_c/d_c^{0.602})$ which gave a reasonable straight line on log-log paper when plotted against supply fluid as shown on Figure 2.

Table 1. Plot relating supply steam temperature to Lip pressure and well bore diamete

°C	$\frac{P_c}{d_c^{0.602}}$
175	7.46
200	9.60
225	12.13
250	15.08
275	18.56
300	22.75
320	26.88
340	32.25

SATURED HOT WATER CALCULATION

As for dry saturated steam, detailed calculations are presented by James (1970) together with charts of viscosity and specific volume for homogeneous mixtures of steam-water substance, at various pressures and enthalpies. The acceptance of no-slip between the steam and water is assumed valid for the case of maximum unrestricted vertical flow to the atmosphere as it agrees with measured values on powerful wells. However, when a well is restricted by throttling the wellhead valve, the homogeneous assumption appears invalid as flow-rates are no longer in accordance with actual measured values.

As in the case for dry saturated steam, a downhole temperature is selected which permits the depth to be calculated from equation (2) and thermodynamic data derived from Steam Tables, but here we have all-water entering the well and increasing in steam fraction as it rises to be discharged to the atmosphere. This discharge takes place at the speed of sound at the Lip pressure located on the rim o the pipe outlet. A trial method is necessary in which both Lip pressure and pipe friction factor have to be initially guessed. Overall pressure-drop is the sum of hydrostatic pressure-drop, frictional pressure-drop and pressure-drop due to the increase in kinetic energy within the pipe from bottom entry to top exit.

Results are charted on Table 2 similar to that for dry saturated steam, and then plotted on Figure 2.

Table 2. Plot relating supply water temperature to Lip pressure and well bore diameter

°C	$\frac{P_c}{d_c^{0.602}}$
200	9.35
250	15.60
300	22.71
330	27.75
350	31.00
360	33.70

CONCLUSIONS

It should be pointed out that the extraordinary agreement shown on Figure 2 for both steam flow and flashing hot water would most probably not have been investigated if unobserved on a geothermal well at Wairakei, where hot water at the bottom changed over to steam at the same temperature and pressure.

As both bottom hole and exit pressures are identical, one might assume that the pressure curve over the well depth is also the same and hence it may be possible to estimate the steam-water pressure-drop at any location by calculating that for the steam curve, but this would need verification by experiment.

If the match is so good for vertical flow, would we expect a similar match for horizontal flow? Provisional calculations indicate what one would suspect, namely an increasing divergence with depth as the weight of the water fraction in the steam-water mixture exerts its dominance. Presumably vertical flow has some compensating factors which bring close agreement for these apparently different modes of flow, at least over the temperature range common to geothermal reservoirs suitable for power exploitation (175° to 350°C).

The straight line on Figure 2 passes through all the plotted points with good agreement and has the following equation:

$$\frac{P_c}{d_c^{0.602}} = \left(\frac{C}{72.2}\right)^{2.195} \qquad (3)$$

<u>EXAMPLE</u>

If a 0.2 m diameter geothermal well is drilled 800 m into reservoir which is at boiling point throughout its depth, what is the maximum flow possible?

Maximum flow occurs at wide-open vertical discharge as shown in Figure 1, and for perfect permeability at the downhole supply horizon, which is here assumed.

The temperature at a depth of 800 m is calculated from equation (2)

$$C = 69.56\ (800)^{0.2085} = 280.32°\text{C}$$

From Figure 2, the equation of the line is now used:

$$\frac{P_c}{d_c^{0.602}} = \frac{P_c}{(0.2)^{0.602}} = \left(\frac{280.32}{72.2}\right)^{2.195}$$

That is, P_c = 7.45 bars. If the fluid entering te well is saturated hot water at 280.32° the enthalpy from Steam Tables is 1238 kJ/kg. Then from equation (1),

$$G = \frac{1839\ (7.45)^{0.96}}{(1238)^{1.102}} = 4.87\ \text{t/m}^2\text{s}$$

Fluid flow in tonnes/hour is

$$4.87\ (3600)\ \frac{\pi}{4}\ (0.2)^2 = 550.31\ \text{t/h}$$

If fluid entering the well is dry saturated steam, the enthalpy from Steam Tables is 2779 kJ/kg. Then from equation (1):

$$G = \frac{1839\ (7.45)^{0.96}}{(2779)^{1.102}} = 2.01\ t/m^2s$$

Steam flow in tonnes/hour is

$$2.01\ (3600)\ \frac{\pi}{4}\ (0.2)^2 = 226.91\ t/h$$

These are the maximum flow-rates possible; actual wells have reduced discharges due principally to relative impermeability of reservoir rocks retarding inflow at the supply horizon (granulated bed, fissure or fractures).

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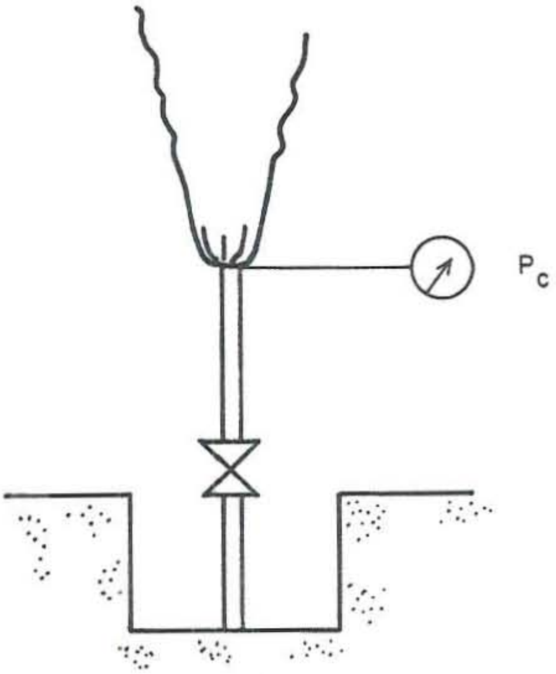

Figure 1. Maximum discharge of a well.
Vertical flow with valve wide-open

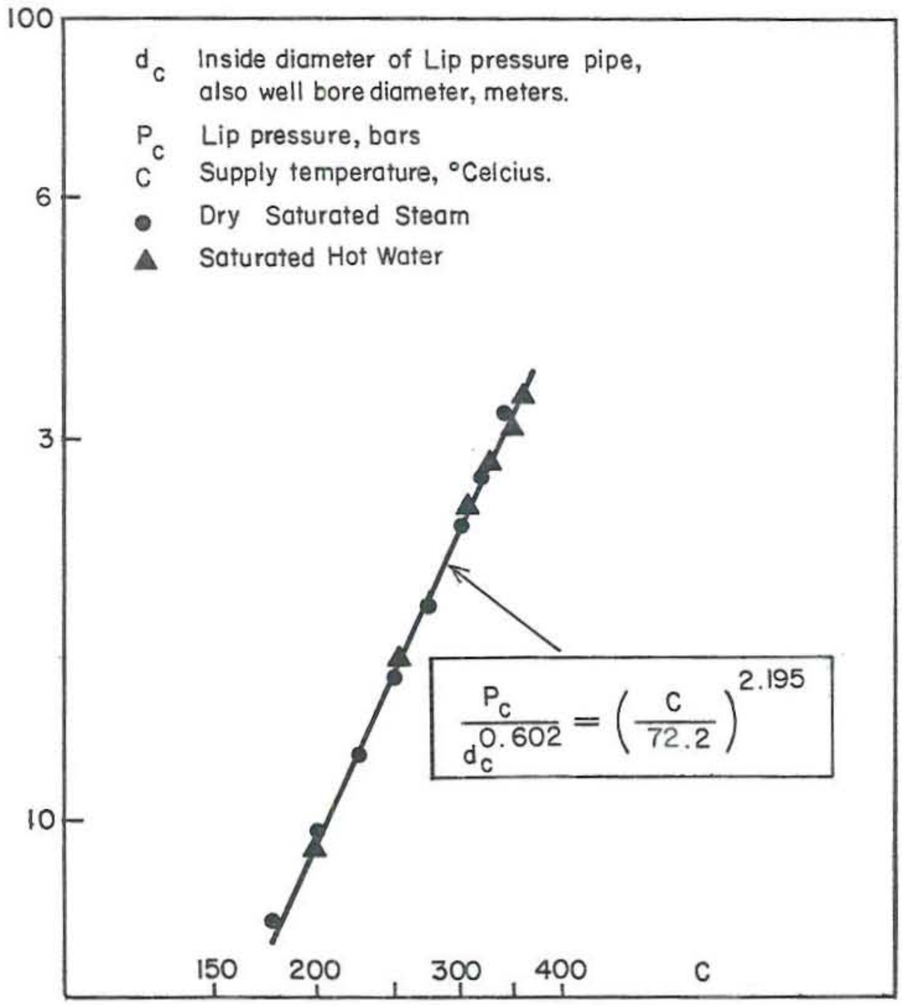

Figure 2. Maximum vertical Lip pressure for Hot Water and Steam Reservoirs.

SIGNIFICANCE OF THE MAXIMUM DISCHARGING-PRESSURE

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ABSTRACT

It would be impressive to raise the well head pressure of a geothermal borehole to the maximum sustainable by flow and from this deduce the supply water temperature, rate of discharge, dryness fraction and density of the wellhead fluid. All this data can, in fact, be obtained from the reading of the pressure gauge installed below the wellhead control valve so long as the flow condition is that given above. The tentative results can be valuable for untested wells drilled in isolated areas; and for monitoring production wells as it permits estimates to be made of changing subterranean conditions.

INTRODUCTION

From the early days at Wairakei, it was noted that wellhead pressures could only be raised to a certain maximum value when throttling discharge. Any attempt to raise it further resulted in collapse of the flowing steam-water mixture and closure of the well. From a wide-open vertical discharge of about 500 t/h, the most productive wells would reduce to about 110 t/h at a Maximum Discharging-Pressure (MDP) of 25.7 bars or thereabouts. This was for 0.2 m diameter boreholes drilled into the Wairakei reservoir, which approximated to Boiling Point with Depth (BPD) down to about 460 m at 250°C. Since then (20 years ago), both reservoir temperatures and values of MDP have decreased with the latter dependent on the former. A correlation between these factors was calculated by James (1970) as:

$$C = 99.75 \quad P_m^{0.283} \quad \text{for } 8 < P_m < 80 \qquad (1)$$

Borehole tests in a number of countries have shown this equation to be surprisingly accurate (confirmed by downhole Kuster measurements on flowing wells) in estimating supply water temperatures from values of MDP; this in spite of steam-water mixtures considered as homogeneous. Also, frictional pressure-drop was ignored, as was kinetic energy increase in the flowing fluid and potential energy requirements to elevate fluid to the wellhead. These were found negligible compared with the hydrostatic head imposed by the column of ascending steam-water mixture.

Hence $$P_s - P_m = \frac{9.81\ L}{100\ V_{sw}} \quad (2)$$

where V_{sw} is the homogeneous steam-water specific volume taken at the average pressure of $(P_s + P_m)/2$ over the flashing length L and at the enthalpy h of the supply water temperature C. The temperature with depth relationship in a reservoir pressurised hot water exists is shown of Figure 1 and has the equation:

$$C = 69.56\ L^{0.2085} \quad \text{for } 30 < L < 3000 \quad (3)$$

Even if a well is drilled to below depth L and then discharged at MDP, boiling will first start within the casing at very close to depth L associated with temperature C in the above equation. This, no doubt, explains the accurate results obtained whether fluid is supplied from the BPD zone of from greater depth within the pressurised hot water underlying it.

Equations (2) and (3) are required to solve for P_m, the results of which are given in equation (1), which is independent of well diameter due the dominance of the hydrostatic head over frictional and kinetic energy effects. Using the latter equation, for various values of P_m, the associated supply water temperatures and enthalpies are given in Table 1.

DRYNESS FRACTION AND SPECIFIC VOLUME AT THE WELLHEAD

When the wellhead pressure equals P_m, it is of interest to see how the dryness fraction and specific volume of the steam-water mixture varies. Values are calculated and given in Table 1 and it is seen that, over a range of P_m up to 70 bar, dryness fraction

$$q\% = \frac{P_m}{4} \tag{4}$$

while V_{sw} is approximately constant at 6 m^3/t.

In other words, whatever the water temperature (up to 332°C) supplying the well at MDP, the density of the steam-water mixture at the wellhead is fairly constant as this is the reciprocal of specific volume. It appears likely that "Bubble" flow takes place over the lower levels of the well with "Churn" flow at higher levels, as described by Taitel et al. (1980). For these conditions, at MDP, both steam and water phases travel at approximately the same velocity, hence the concept of homogeneity adopted here is a realistic one.

FLOW-RATE AT MDP

MDP values of boreholes in New Zealand are recorded by the Ministry of Works and Development, and a study of flow-rates at different supply water enthalpies h and bore diameters d, gives the pragmatic rule

$$W = 2.5\ hd^2 \tag{5}$$

Values of flow-rate are given in Table 1 for a well of 0.2 m diameter.

MIXTURE VELOCITY AT THE WELLHEAD

The flow-rate is required to determine the velocity of the steam-water mixture at the wellhead, upstream of the control valve. At MDP $u_{sw} = \dfrac{W\ V_{sw}}{\frac{\pi}{4}\ d^2\ 3600}$

Substituting W of equation (5), and taking the value of $V_{sw} = 6$ as constant over the range of interest where $P_m \leq 70$ bars.

$$u_{sw} = \frac{h}{188.5}$$

As expected, mixture velocity is independent of borehole diameter and increases with supply water temperature and P_m as shown in Table 1.

Table 1. Physical Factors related to P_m for a Geothermal Well flowing a Maximum Discharge Pressure. W values for d = 0.2 m.

P_m	C	h	q%	V_{sw}	W	U_{sw}
10	191.4	814	2.54	6.03	81.4	4.32
20	232.9	1002	4.93	6.03	100.2	5.32
30	261.2	1140.6	7.36	6.03	114.0	6.05
40	283.3	1254	9.73	5.97	125.4	6.65
50	301.8	1354.1	12.19	5.94	135.4	7.18
60	317.8	1445	15.27	6.07	144.5	7.67
70	332	1535	17.81	5.99	153.5	8.14
80	344.7	1627	21.53	6.15	162.7	8.63
90	356.4	1724.7	26.21	6.41	172.5	9.15
100	367.2	1848	33.4	6.99	184.8	9.8

CONCLUSIONS

From a simple test on a wet geothermal well, the Maximum Discharging-Pressure gives a lot of information, and it is hoped will gain world-wide use. Production wells can be occasionally checked for fall in P_m due to decline in the supply water temperature at depth, as a change as small as 1 degree C will be reflected in a measurable variation in the wellhead pressure gauge as determined by equation (1).

The concept of homogeneity, although not popular in the literature of two-phase flow, appears to apply to the ascent of geothermal steam-water mixtures over large distances and under the restraint of minimum flow at the highest possible wellhead pressure.

NOTATION

C	boiling water temperature at depth L, °Celcius
d	wellbore diameter, metres
h	boiling water enthalpy associated with C, kJ/kg
L	depth, metres
P_m	Maximum Discharging-Pressure (MDP) at wellhead, bars
P_s	boiling water pressure associated with C, bars
q	dryness fraction of steam-water mixture at wellhead
u_{sw}	velocity of steam-water mixture at wellhead, m/s
V_{sw}	specific volume of steam-water mixture at wellhead, m^3/t
W	flow-rate at Maximum Discharging-Pressure, t/h

EXAMPLE

Under discharging conditions, the wellhead pressure of a previously untested borehole is throttled to a maximum value of 37 bars gauge. What provisional deductions can be made, assuming a borehole diameter of 0.2 m, and atmospheric pressure of 1 bar?

Maximum Discharging-Pressure $P_m = 37 + 1 = 38$ bars

From Figure 1 or equation (1), $C = 99.75\ (38)^{0.283} = 279.25\,°C$

This is the temperature of the water supplied to the well at depth, and from Steam Tables has an enthalpy h = 1235 kJ/kg

From equation (5), $W = 2.5\ (1235)\ (0.2)^2 = 123.5$ t/h which is the flow at a wellhead pressure of 38 bars.

Conditions at the wellhead are as follows:

Dryness fraction as a percent $q\% = \frac{P_m}{4} = \frac{38}{4} = 9.5\%$

From Section (4).

As the value of P_m is less than 70 bars, the specific volume of the steam-water mixture is constant at 6 m^3/t and density is the reciprocal 0.167 t/m^3. Wellhead mixture velocity (homogeneous) from equation (6),

$$u_{sw} = \frac{1235}{188.5} = 6.55 \text{ m/s}$$

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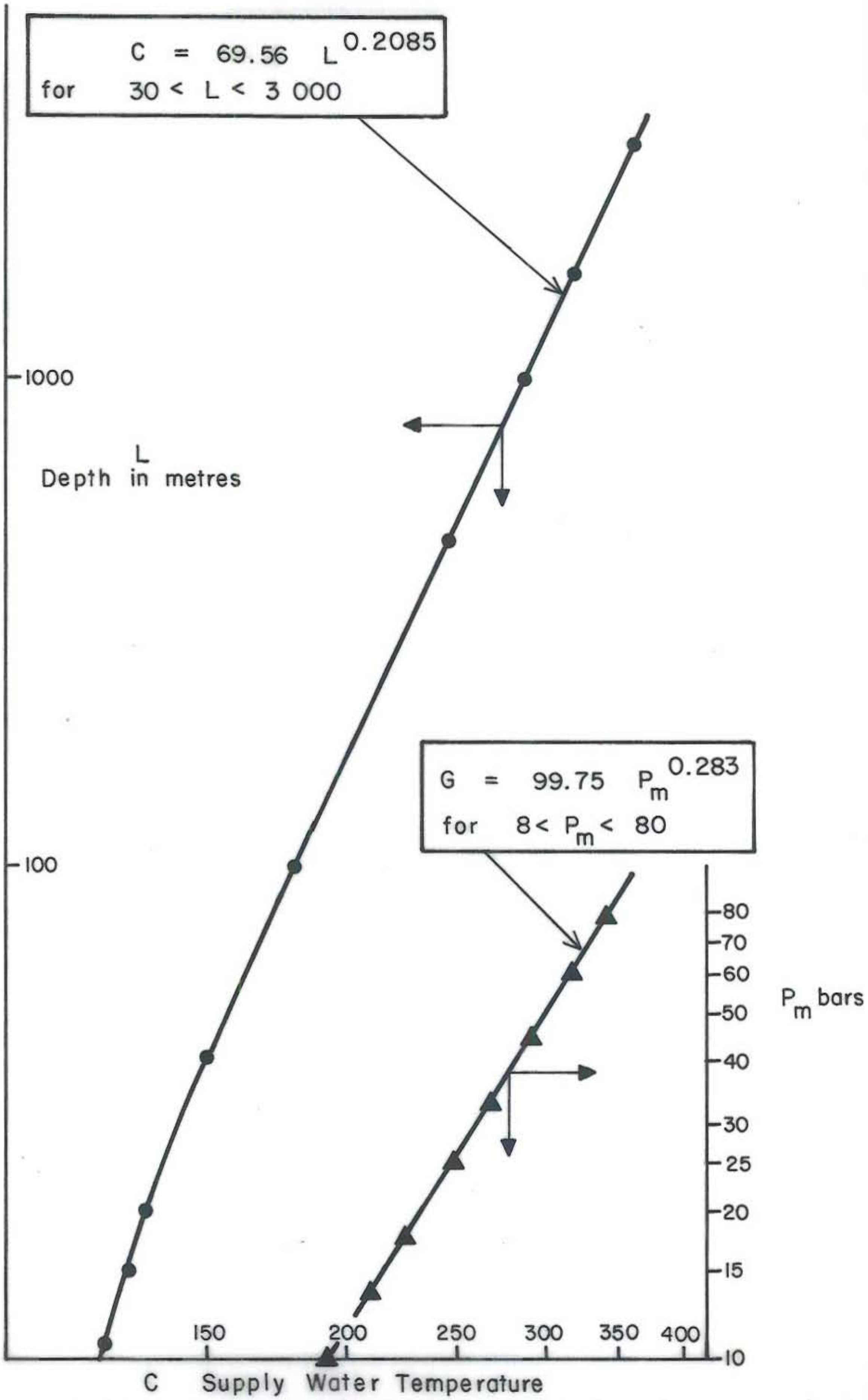

Fig. 1. Temperature versus depth and Maximum discharge Pressure P_m for boiling throughout Reservoir, with 100° at Surface.

POWER POTENTIAL OF GEOTHERMAL WELLS RELATED TO RESERVOIR TEMPERATURE

by

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ABSTRACT

For equal flows of hot water wells, the electric power which can be generated increases with feed water temperature. However, high temperature permeable wells discharge greater flows than that of similar lower temperature wells, with the result of enhanced power potential.

In fact, where fluids are exploited utilising two-stage flash, these factors combine to give a power potential which is porportional to the cube of the feed water temperature. Hence a feed of 315 C would generate twice the power of that of water at 250 C for wells of good permeability and where the reservoir exists under conditions of boiling point with depth.

Higher temperature water (exceeding 300 C) has, however, a commensurate higher tendency to mineral deposition in reinjection water lines and this disposes design to single-stage flash with slightly reduced power, compared with the two-stage alternative.

INTRODUCTION

After the discovery and exploitation of the Larderello field in Italy, drillers have sought reservoirs which produce dry or super-heated steam and have had successes with The Geysers (U.S.A.), Kamojang (Indonesia), Matsukawa (Japan) and others. Most reservoirs penetrated, however, have proved to contain hot water at or near the boiling point for depth and many have been developed since the building of the Wairakei station. Early theory, James (1966), urged shallow drilling (order of 350 m) into hot water reservoirs, so as to exploit the 'top of the boiler' and tap the thin layer of steam believed to exist there. By this means a steadily increasing volume of vapour should spread over the top reaches of the reservoir supplied by the underlying water which boils and draws heat from the rock matrix. The efficiency of such an approach was calculated as far higher than the alternative of simply discharging boiling water as the latter would have less than half the power-life of the former which more effectively utilizes the rock heat. To judge by the power generated at Larderello and The Geysers, this may be so, especially as it is believed that such dry steam reservoirs are indeed under-pinned by large volumes of water either beneath the steam zone, James (1968) or co-mingling with it, Truesdell and White (1973). In the initial development of a hot water field, a putative steam zone may be of small thickness or possibly non-existent and in the latter case a well drilled to about 350 m would merely tap hot water at about 236 C whereas deeper drilling would most probably find higher temperature water capable of greater power potential even though having to discharge through a longer length of borehole. Statistics indicate that this is so, as the average depth of hundreds of wells drilled in geothermal regions in the U.S.A. over the last few years has reached 2 km, which, for a classical boiling point with depth (BPD) reservoir would attain 339 C. This would be an attractive result so long as bottom hole permeability was good and that the mineral concentration in the brine was not too high to present problems with reinjection or in-hole scaling. And nowadays, reinjection is a mandatory part of nearly all world-wide geothermal schemes, even for dry steam reservoirs with their relatively small quantity of condensate to be disposed.

It may be appropriate to reasses drilling strategy and make a determined effort to exploit the top of water reservoirs in the expectation of evolving a steam zone with fewer problems and more energy efficiency potential. However, one cannot be dogmatic, as drilling is an exacting and expensive business with first discharge awaited with nervous anticipation, and reputations dependent, to some extent, on the results. Higher temperature water does, of course, produce more steam than lower temperature water and consequently more electric power can be generated. It is also found that higher temperature water, even though at greater depth, discharges greater flows than lower temperature water for equivalent feed-zone permeabilities, hence the trend to tap deeper horizons is a logical one. Up to now, no quantitative assessment has been possible into the comparative merits of higher water temperature with depth except to be aware that deeper is better. This is because it has not been possible to sensibly compare the discharge of wells which are drilled to various depths without being aware that permeability variation, controls more often than not, whether a well is a good producer, and so a low temperature shallow well can have a greater discharge and power potential than a deeper higher temperature well drilled into tighter formations. Although, as has been pointed out, statistics indicate that the deeper wells are generally a better investment, this may be because unknown factors conspire to increase permeability at around a depth of 2 km in a similar way that good permeability is found at about 350 m depth, James (1984a).

Therefore, for equivalent permeabilities, and assuming that boiling point with depth (BPD) obtains, a good deep well is superior to a good shallow well within the depth common to geothermal drilling, which is down to about 2.5 km. At greater depths, the cost of drilling increases rapidly and begins to exert its influence on the cost-benefit analysis, but in this study that aspect will be ignored, and will only be resurrected if deeper drilling becomes an intrinsic part of geothermal technology.

Maximum Well Discharge

This has been calculated James (1980) for infinite permeability at the feed zone, boiling point with depth, and wide-open stable vertical discharge. The following formula is employed.

$$P_c = d_c^{0.602} \left(\frac{C}{72.2}\right)^{2.195} \quad \text{for } 180 < C < 350 \tag{1}$$

P_c = lip pressure in bar; C = the feed temperature, degree celcius. For the boiling point with depth relationship where H is depth in metres, James (1980).

$$C = 69.56 \; H^{0.2085} \quad \text{for } 30 < H < 3\,000 \tag{2}$$

Formula (1) has been confirmed in practice James (1984b), and applies whether the feed is reservoir water which is just at the boiling point (for the hydrostatic pressure), or whether it is dry saturated steam at the same temperature. The relationship between flow, lip pressure, pipe diameter and fluid enthalpy is derived from James (1962) and in the metric form:-

$$W = 5.2\,(10)^6 \; \frac{P_c^{0.96} \; d_c^{2}}{h_o^{1.102}} \tag{3}$$

W = Flow, t/h $\quad d_c$ = Inside diameter of well and discharge pipe, m.
h_o = Enthalpy of discharge, kJ/kg
We now require a relationship between water enthalpy and temperature and a plot of these factors derived from steam tables, gives:-

$$h_o = 1.475 \; C^{1.197} \quad \text{for } 210 < C < 350 \tag{4}$$

Substituting (1) in (3), we have:-

$$W = 5.2\,(10)^6 \; \frac{d_c^{2.578}}{h_o^{1.102}} \left(\frac{C}{72.2}\right)^{2.1072} \tag{5}$$

Now substitute (4) in (5),

$$W = \frac{5.2\,(10)^6 \; d_c^{2.578}}{\left[1.475 \; C^{1.197}\right]^{1.102}} \left[\frac{C}{72.2}\right]^{2.1072}$$

$$W = 410.82 \; d_c^{2.578} \; C^{0.788} \quad \text{tonne/hr} \tag{6}$$

Although this result gives the maximum flow from a well which can be expected, it can increase with displacement upwards of boiling point with depth in the reservoir; also for low enthalpy wells, high gas content can boost flow to higher values than that determined from equation (6), James (1982). Lower discharge than the maximum can be due to (a) mineral deposition which reduces the well diameter, (b) impermeability of the feed horizon, or (c) displacement downwards of the boiling point with depth relationship.

Equation (6) shows that flow is directly proportional to diameter of well and feed water (or steam) temperature; however, the index of diameter is greater than the expected square law and emphasises the importance of increasing well diameters where excellent permeability exists.

To determine the amount of electric power which can be generated from the flow of equation (6), we require the specific power rate for hot water expanding by two-stage flash into turbo-condensers under optimum design conditions. Fortunately, this has been accomplished, James and Meidav (1977) who present the following relationship:

$$\text{Megawatt (electrical)} = W \left(\frac{C}{1260}\right)^{2.2233} \quad \text{for } 210 < C < 350 \qquad (7)$$

Substituting (6) in (7), we obtain:-

$$\text{Megawatt (electrical)} = \frac{5.26}{(10)^5}\, d_c^{2.578}\; C^{3.0112} \qquad (8)$$

This shows that the electrical power which can be generated from hot water reservoirs is proportional to the cube of the feed temperature, and tapping a reservoir at 315 C for example, should give twice the power of a reservoir at 250 C, all other factors being equal.

Interestingly enough, James (1986) shows that at these reservoir temperatures, the amount of silica transported in the separate steam and water pipelines (for power and injection) is also twice as high for the higher temperature reservoir. Because of potential scaling problems anticipated in the reinjection water lines of higher temperature fields (exceeding about 300 C), it may be that only single-stage flash will be employed rather than the more efficient two-stage flash, in which case the power relationship of equation (7) will have to be replaced with that derived for single-stage, James and Meidav (1977), as:-

$$\text{Megawatt (electrical)} = W\left(\frac{C}{1055}\right)^{2.611} \quad \text{for } 235 < C < 365 \qquad (9)$$

Substituting (6) in (9), we obtain:-

$$\text{Megawatt (electrical)} = \frac{5.247}{(10)^6}\, d_c^{2.578}\, C^{3.399} \qquad (10)$$

For single-stage flash and condensing sets, the relationship of equation (10) shows that a reservoir temperature of 306.6 C would generate twice the power of a reservoir of 250 C. It should be noted that although the index of temperature now exceeds the cube, the power will be less than that of the two-stage alternative, due to the equation constant being smaller by approximately 10. For example, if a well diameter of 0.2 m is taken and a reservoir hot water temperature of 250 C, then the power potential for two-stage flash from equation (8) comes to 13.8 MWe compared with 11.7 MWe for the single-stage alternative. Actual values when flows are reduced by discharging horizontally instead of vertically would lower these figures to about 75%, namely to 10.3 MWe and 8.8 MWe. Further decline would also be expected over the years as reservoir pressures fall with time and exploitation.

Maximum Power from Dry Steam Reservoirs

The shallower depths of dry steam reservoirs are usually at a temperature close to about 236 C and at a horizon which agrees with that calculated from equation (2), namely 350 m and because of this, the lip pressure of equation (1) can be determined independently of whether steam or hot water is the fluid involved. Therefore, inserting 236 C in equation (1) we have:-

$$P_c = d_c^{0.602}\left(\frac{236}{72.2}\right)^{2.195} = 13.46\ d_c^{0.602} \qquad (11)$$

Substituting this value of P_c in equation (3) and letting h_o = 2804.1 kJ/kg which is the enthalpy of dry saturated steam at 236 C, we obtain

$$W = d_c^{2.578}\ (10)^4 \quad \text{tonne/hr} \qquad (12)$$

Taking a steam power specific rate of 10 tonne/MWh, James and Meidav (1977), we can now determine the maximum electric power which can be generated by a dry steam reservoir as

$$\text{Megawatt (electrical)} = \frac{d_c^{2.578} \quad (10)^4}{10} = (10)^3 \, d_c^{2.578} \qquad (13)$$

Hence, for a well diameter of 0.2 m, the power potential = 15.8 MWe. If we substitute this value in equations (8) and (10) for the same well diameter, we can determine the temperature of hot water reservoirs which give identical power to the steam reservoir, as follows:-
Hot water reservoir employing two-stage flash to give 15.8 MWe, and d_c = 0.2 m.

$$15.8 = \frac{5.26}{(10)^5} (0.2)^{2.578} \quad C^{3.0112}$$

Equivalent hot water reservoir temperature = 261.5 C

For hot water reservoir employing single-stage flash to give 15.8 MWe,

$$15.8 = \frac{5.247}{(10)^6} \quad (0.2)^{2.578} \quad C^{3.399}$$

Equivalent hot water reservoir temperature = 273 C.

As has been pointed out, unless there is a technical and economic breakthrough in the control of mineral scaling from the separated injection water, it is likely that the single-stage design will be increasingly prevalent, especially as deeper drilling is gaining momentum and higher temperature water reservoirs discovered.

Although it might be thought from these figures that a hot water reservoir which exceeds 273 C has the advantage over a dry steam reservoir at 236 C, the simplicity and low overall cost of the latter together with its avoidance of massive water injection (with largely unknown side-effects) makes the exploitation of dry steam reservoirs a much more attractive proposition.

CONCLUSIONS

Because of the greatly increased power potential of hotter water at deeper horizons, there is considerable incentive to ignore shallow drilling. However, high temperature water usually contains increased concentrations of dissolved minerals which can lead to scaling of reinjection water lines, reinjection wells, and the surrounding reservoir, with serious consequences for the "life" of a project.

As reinjection is an inherent part of most future power developments and the problem of scaling not yet solved technically and economically, it is recommended that a sustained effort be made into locating (and subsequently exploiting) the dry steam believed to exist at close to the criticus temperature of 236 C, James (1986). Such steam horizons would be situated above hot water reservoirs and ideally located at a depth of about 350 m. Even if hot water is found at this level, it will most probably have a relatively low level of dissolved minerals and will permit two-stage flash exploitation, with reinjection free from scaling problems; a not unattractive scenario. And continued production of water will lead (it is hoped) to eventual changeover to steam flows with development of a spreading vapour-filled zone.

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HEAT LOSS AND PRESSURE-DROP BALANCE FOR GEOTHERMAL STEAM TRANSMISSION

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ABSTRACT

Heat loss through long, large, insulated pipelines transmitting dry saturated geothermal steam, results in the steady formation of condensate. For low-pressure steam - defined as below the criticus temperature of 236 C and associated pressure of 31 bar - pressure-drop along the pipeline at constant enthalpy, evaporates condensate.

These contradictory effects determine the mineral concentration (in the pipe water) which is derived from carryover brine escaping into the steamline from wellhead separators. This liquid has to be removed from long pipelines by extraction pots, and from short pipelines by mist extractors.

A study of these opposing wet-dry tendencies is necessary to determine the efficiency of extraction devices and to estimate whether minerals will attain and possibly scale the turbine.

Steam flow can, in fact, be increased in an existing pipeline up to a point at which 'dry-out' occurs, water extractors cease to function, and minerals precipitate to coat the pipe walls as well as the turbine.

For hot water geothermal fields, separated steam is dry saturated and to evaluate the true wetness present in steam pipelines, the required parameters are pipe diameter, insulation thickness, steam flow and pressure. Results on a Wairakei low-pressure steam pipeline of 1.22 m diameter (48 inch) agree closely with calculated predictions.

INTRODUCTION

Dry saturated steam, when transmitted through insulated pipelines, loses heat and a fraction of the vapour condenses into water. In the case of geothermal steam derived from hot water reservoirs, some additional liquid, in the form of brine, usually escapes into the steam line from the water separators installed at wellheads. As the steam moves down the pipeline, dilution of the mineral constituents of this brine takes place with the steady generation of condensate so that one would expect a diminishing concentration of chemicals along the line. Even if the water present was pure, it would have to be removed bevore reaching the power house, as it can cause erosion of turbine blades. However, pure water presents problems in the form of severe corrosion of the lower half of transmission steam pipelines, James (1980), particularly if some carbon dioxide is present in the vapour and this gas is a normal constituent of geothermal steam. Due to carryover brine in the line, the chemicals present inhibit corrosion but most of these dissolved constituents have to be removed before the end of the pipeline, otherwise they can cause deposition of solids on blades and within turbine nozzles, James (1986). A residual amount is retained as it has been found that a few parts per million of silica is sufficient to significantly retard corrosion. It is obviously of importance to determine how much liquid and minerals are moving along the pipeline with the steam and how efficiently these quantities are removed by extraction devices spaced at intervals along the line. The subject is complicated by the fact that the mass of carryover brine is not accurately known. More subtle is the phenomenon of low-pressure steam drying condensate as it flows with frictional pressure-drop to the pipe end. This takes place when dry saturated steam at a pressure less than the criticus ($\sim$31 bar) passes along pipelines at constant enthalpy; this is normal for geothermal steam where steam line pressures are usually in the range of 2 to 20 bar. Steam velocities are commonly less than 100 m/s so that the kinetic energy effect on enthalpy is negligible; this consequently does not deviate significantly from the dry saturated steam condition for a given pressure.

As scale deposition, together with erosion and corrosion of metals are all involved in this subject with economic ramifications for geothermal power, this study deals with the opposing forces of wetness and dryness which influence flows of minerals and liquids towards the turbine house.

As an example of the problems faced, we may cite a steam line in which the frictional pressure-drop is increased to a point in which drying just matches formation of condensate. In this case, chemical concentration in the liquid would remain constant and extraction devices might be rates as virtually useless even though they may actually be efficiently reducing the mass flow of both minerals and water from the line.

Another example would be of dry saturated steam passing from a vessel through a long insulated pipeline to the atmosphere. In this case, the initially dry steam would become progressively wetter with heat loss followed by gradual drying out as pressure falls along the line. Finally near the pipe outlet, velocity and kinetic energy would increase to such an extent that constant enthalpy would no longer prevail and the steam would become wetter again, reaching a maximum at the outlet plane with sonic conditions at steam velocities of about 500 m/s, James (1962). Unless such characteristics are appreciated, water extraction devices being tested in such a line could give confusing results unless the internal conditions are precisely known.

Heat Loss from Insulated Pipes

The usual means of calculating heat loss from insulated pipes, Potter (1959) with logarithmic functions of the ratio of outer to inner radii of insulation is avoided here as not conducive to ease of handling mathematically with other equations. Accordingly, the approach undertaken was to plot specific heat loss from pipes containing dry saturated steam at different insulation thicknesses, Lyle (1947), from which the following equation was derived:-

$$H = \frac{2.019\ P^{0.321}}{t^{0.737}} \qquad (1)$$

Where P = Pressure of dry saturated steam, bar.

t = Insulation thickness, mm.

H = Specific thermal heat loss, kWt/m^2

Ideally, this equation should apply to flat surfaces or large pipes (greater than say 0.25 m) and this is suitable for geothermal projects where even branch steam lines exceed 0.3 m diameter, and main transmission lines can attain 1.22 m diameter, at Wairakei, for instance. As factory interiors were used in the development of the equation, with ambient conditions of 20 C and no wind, pipeline tests in the geothermal field are undertaken on warm, windless days to reduce error. To calculate the condensate accumulated in a pipeline of 1 000 m length, as a percentage of steam flow, we have:-

$$\% \text{ Wetness/km} = \frac{H\pi D}{h_{fg}} \frac{1000}{W} 100 \tag{2}$$

D = Pipeline diameter, m.

W = Steam flow, kg/s

h_{fg} = Latent heat of steam at pipeline average pressure, kJ/kg.

A plot of the latent heat of steam versus the pressure of dry saturated steam gives the following equation:-

$$h_{fg} = \frac{2309.17}{P^{0.06724}} \quad \text{for } 2 < P < 20 \tag{3}$$

Therefore substituting equations (1) and (3) in (2), we have:-

$$\% \text{ Wetness/km} = 274.62 \frac{P^{0.3882} D}{t^{0.737} W} \tag{4}$$

However, the true pipeline wetness will be less than this due to pressure-drop.

Friction Press-Drop of Dry Saturated Steam

Most pipeline pressure-drop calculations depend on an iterative procedure, but for the specific case where the transmitted fluid is dry saturated steam, factors such as steam density and viscosity are related to steam pressure, and friction factor dependent on Reynold's number, Perry (1963), which in turn, is also dependent on flowrate, pipe diameter, density and viscosity. The following formula takes into account varying friction factor and is successfully employed at Wairakei. See appendix for derivation.

$$\Delta p/\text{km} = \frac{3.3 \quad W^{1.85}}{(10)^{4}\ D^{4.85}\ P^{0.93}} \tag{5}$$

$\Delta p/\text{km}$ = Pressure-drop per kilometre, bar/km

Drying Effect of Pressure-Drop

Plotting from Steam Tables, Keenan et al. (1969), we obtain the following relationship between dry steam enthalpy and saturated vapour pressure:-

$$h_g = 2675.42\ P^{0.016646} \quad \text{for} \quad 2 < P < 20 \tag{6}$$

Expansion of such dry saturated steam at constant enthalpy, results in superheating, which can evaporate a proportion of water present. Differentiating h_g with respect to P, we have:-

$$\Delta h_g = \frac{44.535}{P^{0.983354}}\ \Delta P \tag{7}$$

This increment of enthalpy Δh_g can evaporate $\frac{\Delta h_g}{h_{fg}}$ kg of condensate for every kilogram of steam flowing. Or in percentage terms,

$$\text{Wetness dried by pressure-drop} = \frac{\Delta h_g}{h_{fg}}\ (100) \tag{8}$$

Substituting (5) in (7) followed by (3) and (7) in (8), we obtain:-

$$\%\ \text{Wetness dried by pressure-drop/km} = \frac{6.3672 \quad W^{1.85}}{(10)^{4}\ P^{1.8421}\ D^{4.85}} \tag{9}$$

True wetness in pipeline requires deducting equation (9) from (4),

$$w_t = \text{True \% Wetness/km} = \frac{274.62\ P^{0.3882} D}{t^{0.737}\ W} - \frac{6.3672\ W^{1.85}}{(10)^{4}\ P^{1.8421}\ D^{4.85}} \tag{10}$$

Equation (10) can be used to determine the true wetness over a length of

1 km which is a convenient distance for geothermal overland pipelines. For other length of pipelines, wetness is directly proportional to length and can be factored accordingly.

Example 1

The 'G' line at Wairakei transmits 65.54 kg/s of dry saturated steam at 2.4 bar; pipe diameter = 1.2192 m and insulation thickness 38.1 mm. Determine condensate flowing in the pipeline between extraction pots which are spaced 137 m apart, and the pressure-drop per kilometre. From the first part of equation (10),

$$\% \text{ Wetness/km} = \frac{274.62(2.4)^{0.3882}}{(38.1)^{0.737}} \frac{(1.2192)}{(65.54)} = 0.491\%$$

From second part of equation (10),

% Wetness dried by pressure-drop/km

$$= \frac{6.3672\ (65.54)^{1.85}}{(10)^{4}\ (2.4)^{1.8421}\ (1.2192)^{4.85}} = 0.111\%$$

Therefore true % wetness/km = 0.491 - 0.111 = 0.38%

Amount of water condensing between pots/hour

$$= \left(\frac{0.38}{100}\right) \left(\frac{137}{1000}\right)\ 65.54\ \ (3600) = 122.83 \text{ kg/hr}$$

When this quantity is discharged to the atmospheric pressure of about 1 bar, 5% of steam is flashed off resulting in 116.7 kg/hr of extracted water at near 100 C or 121.4 litre/hr. Actual field measurements taken gave a collection rate of 120 litres/hour which is close to the above calculated value.

$$\text{From equation (5)}: \quad \Delta p/\text{km} = \frac{3.3 \quad (65.54)^{1.85}}{(10)^{4}\ (1.2192)^{4.85}\ (2.4)^{0.93}} = 0.129 \text{ bar/km}$$

To completely dry water condensing, the two parts of equation (1) must equate to give:-

$$\frac{274.62\ P^{0.3882}}{t^{0.737}}\ \frac{D}{W} = \frac{6.3672}{(10)^4}\ \frac{W^{1.85}}{P^{1.8421}\ D^{4.85}} \tag{11}$$

Hence:-

$$W = 94.87\ \frac{P^{0.783}\ D^{2.053}}{t^{0.259}} \tag{12}$$

Equation (12) gives the flowrate which should just keep the steam in the dry saturated condition, i.e. neither wet nor superheated and is a condition which is best avoided if carryover brine has entered the steam pipeline, as complete drying of this fluid might eventuate which would ensure deposition of minerals.

Example 2

What flowrate in the above example would keep the steam dry but not superheated, for the identical line pressure of 2.4 bar.

From equation (12):

$$W = \frac{94.87\ (2.4)^{0.783}\ (1.2192)^{2.053}}{(38.1)^{0.259}} = 110.176 \text{ kg/s } (=396.6 \text{ t/h})$$

The pressure-drop will now have increased as determined from equation (5).

$$\Delta p/\text{km} = \frac{3.3\ (110.176)^{1.85}}{(10)^4\ (1.2192)^{4.85}\ (2.4)^{0.93}} = 0.337 \text{ bar/km}$$

Example 3

What line pressure in Example 1 would keep the steam dry but not superheated, for identical flowrate of 65.54 kg/s

From equation (12):-

$$65.54 = 94.87\ \frac{P^{0.783}\ 1.2192^{2.053}}{(38.1)^{0.259}}$$

Therefore $P = 1.2363$ bar

The pressure-drop will now be, from equation (5):-

$$\Delta p/km = \frac{3.3 \quad (65.54)^{1.85}}{(10)^4 \; (1.2192)^{4.85} \; (1.2363)^{0.93}} = 0.2379 \text{ bar/km}$$

It is clear from Example 3, that for a long pipeline, of fixed diameter, flowrate and insulation thickness, the pressure downstream may fall to a value that keeps the water content constant by balancing any further condensation with the drying effect of the increased frictional pressure-drop. Continued fall in line pressure would result in progressive drying-out of the water present.

Superheated Steam Pipelines

Typical of this condition are the pipelines of dry steam fields such as The Geysers, Larderello, Kamojang and Matsukawa in which the steam at the wellheads is already superheated by about 5 to 15°C at wellhead pressures 5 to 10 bar.

The amount of superheat which is necessary to just dry out the true % wetness, w_t, in long insulated pipelines can be calculated as follows, where the specific heat of superheated steam in the range 2 to 20 bar is close to 0.5 calories/g. The superheat available (above the dry saturated condition) is ΔT degree celcius, and is considered to fall to zero with dry-out of condensate, over a distance of 1 kilometre.

Hence,

$$0.5 \; \Delta T \; J = \frac{w_t}{100} \; h_{fg} \tag{13}$$

Substituting for h_{fg} from equation (3), and for Joule's equivalent $J = 4.1868$ kJ/kg

w_t = True % wetness as determined by equation (10).

$$\Delta T = \frac{11.03 \; w_t}{p^{0.06724}} \tag{14}$$

For overland geothermal steam pipelines which are within the pressure range of 2 to 20 bar, $P^{0.06724}$ is 1.048 and 1.200 respectively, accordingly the above equation simplifies to approximately:-

$$\Delta T = 10 \, w_t \tag{15}$$

Example 4

How many degrees of superheat entering the pipeline of Example 1, would be required to keep the steam dry over a line length of 1 kilometre?

The true wetness was evaluated in the example as 0.38 % for 1 km, hence superheat required $\Delta T = 10\ (0.38) = 3.8^{\circ}C$. Geothermal steam pipelines range up to about 3 km in length and for the latter figure, a superheat at the wellheads would have to be 3 times $3.8^{\circ}C$, namely $11.4^{\circ}C$ to avoid condensate appearing in the line.

Example 5

Dry saturated steam enters a pipeline of 1 km length, diameter 0.762 m, insulation thickness 38.1 mm, flowrate 58.6 kg/s and average pressure of 7.931 bar. Determine the wetness at the end of the line. Drying-out of this condensate could be accomplished if the steam was initially superheated. Alternatively increasing the flow to a certain value would result in frictional drying to retain the steam in the dry saturated condition. Evaluate these alternatives.

From equation (10), true % wetness/km

$$= \frac{274.62\ (7.931)^{0.3882}}{38.1^{0.737}} \frac{0.762}{58.6} - \frac{6.367}{(10)^4} \frac{(58.6)^{1.85}}{(7.931)^{1.842}(0.762)^{4.85}}$$

$$= 0.545 - 0.098$$

$$= \underline{0.477\ \%\ \text{wetness/km}}$$

From equation (15), initial superheat to eliminate this wetness,

$$\Delta T = 10\ (0.477) = 4.77^{\circ}C$$

Alternatively, dryness is ensured by increasing the flow to a value determined by equation (12), as follows:-

$$W = 94.87\ \frac{(7.931)^{0.783}\ (o.762)^{2.053}}{(38.1)^{0.259}} = 107\ kg/s$$

For this high flow, it might be advisable to estimate the pressure-drop over the 1 km length. From equation (5), we have:-

$$\Delta p/km = \frac{3.3}{(10)^4}\ \frac{(107)^{1.85}}{(0.762)^{4.85}\ (7.931)^{0.93}} = 1.03\ bar/km$$

Hence pressure at end of line = 7.931 -1.03 = 6.9 bar of dry steam.

CONCLUSIONS

The purpose of this paper is to derive working field equations free from the iterative procedures common to the genre. Geothermal steam is either dry saturated or slightly superheated, and heat loss from pipes wets the former and reduces the latter, while frictional pressure-drop dries the condensate. These conflicting factors have to be evaluated in order to determine what is happening within the line; most importantly, whether the steam is dry or wet and if wet, how wet? This is because pure condensate (with some dissolved carbon dioxide) is corrosive to pipes and turbines; James (1980), while superheated or dry steam contains silica dissolved in the vapour, or existing as fine particles (the so-called Geyser dust), James (1986). Consequently whether corrosion or deposition occurs depends largely on the steam condition and upon this, in turn, depends whether liquid extraction devices are needed for wet steam or whether spray demisters should be involved for dry or superheated steam. Corrosion and scaling are common and expensive problems to geothermal power developments whether based on hot water or dry steam reservoirs. It is hoped that this work,which is specific to steam transmission pipelines, will help to reduce the costburden.

NOTATION

D	Pipe diameter, m.
f	Fanning friction factor
g	Gravitational constant, 9.81 m/s^2
G	Mass velocity, t/m^2 s.
h_{fg}	Latent heat of steam, kJ/kg
h_g	Enthalpy of dry saturated steam, kJ/kg
H	Pipeline thermal heat loss, kWt/m^2
J	Joule's equivalent, kJ/kcal.
L	Pipe length, m.
P	Pressure of dry saturated steam, bar.
R_n	Reynolds' Number of flow = $\frac{GD(10)^6}{\mu}$
t	Insulation thickness, mm
V_g	Specific volume of dry saturated steam, m^3/t
W	Steam flow, kg/s
w_t	True percentage wetness of steam
Δp	Pressure-drop, bar
ΔT	Superheat, degree celcius
μ	Viscosity of dry saturated steam, centipoise

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APPENDIX

Derivation of Pressure-drop Formula Specific for Dry Saturated Steam

The basic equation for the flow of any fluid through pipes, Perry (1963), is as follows:-

$$\text{Pressure-drop in } t/m^2 = 4 \; f \left(\frac{L}{D}\right) \frac{G^2 V}{2\,g} \tag{16}$$

To obtain this pressure-drop in the more convenient bar units we have to multiply both sides of the above equation by $\frac{g}{100}$, hence:-

$$\Delta p = \frac{f}{50} \left(\frac{L}{D}\right) G^2 \; V \tag{17}$$

For the flow of dry saturated steam, V is replaced by the specific volume of such steam, which is related to pressure when plotted from steam tables as:-

$$V_g = \frac{1675}{P^{0.945}} \quad \text{for} \quad 0.3 < P < 40 \tag{18}$$

We may also substitute in equation (17) the following equation, to obtain the more convenient units of W kg/s.

$$G = \frac{W}{1\,000 \ \frac{\Pi}{4} \ D^2} \tag{19}$$

The Fanning friction factor f, is related to the Reynolds' Number, Rn of the flow by an approximate equation derived from line 'D' of Figure 5-20 of Perry (1963) for commercial steel pipe, and is as follows:-

$$f = \frac{0.034}{R_n^{\ 0.1505}} \tag{20}$$

where $$R_n = \frac{G\ D}{\mu} \ (10)^6 \tag{21}$$

A plot of viscosity μ centipoise against dry saturated steam pressure, gives:-

$$\mu = \frac{P^{0.1}}{76.536} \quad \text{for} \quad 2 < P < 20 \tag{22}$$

Employing equations (19), (21) and (22), we obtain from equation (20),

$$f = 0.006035 \ \frac{P^{0.015} \ D^{0.15}}{W^{0.15}} \tag{23}$$

These various values may now be employed in equation (17), as follows:-

$$\Delta p = \frac{0.006035 \ P^{0.015} \ D^{0.15}}{50 \qquad W^{0.15}} \left(\frac{L}{D}\right) \left(\frac{W}{1000 \ \frac{\pi}{4} \ D^2}\right)^2 \frac{1675}{P^{0.945}}$$

Simplifying,

$$\Delta p = \frac{3.3 \ \ L \ \ W^{1.85}}{(10)^7 \ P^{0.93} \ D^{4.85}} \tag{24}$$

Taking a length L = 1 000 m, we obtain the pressure-drop/km as:-

$$\Delta p/km = \frac{3.3 \; W^{1.85}}{(10)^4 \; D^{4.85} \; P^{0.93}} \tag{25}$$

This equation (25) is useful for geothermal steam transmission pipes where the flowing steam is usually either at or close to the dry saturated condition.

TRANSMISSION OF HOT WATER AT THE BOILING POINT

by

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ABSTRACT

Future geothermal power projects will almost certainly involve the overland piping of hot water from the wellhead separator outlets to distant reinjection wells. This water, although under pressure is just at the boiling point and hence any reduction in pressure will result in the generation of steam within the fluid, leading to instabilities of flow and impossible design problems. By comparison, the design of pipelines transmitting steam-water mixtures is much simpler due to the buffering action of the omnipresent relatively large steam volume.

However, as a hot water pipeline would not be insulated, the inevitable heat loss can be nicely brought into balance with the frictional (and other) pressure-drops to ensure that at no point along the line does boiling actually occur. Precise control of the water velocity is required to suppress this likelihood and is surprisingly found to be independant of pipe diameter. A convenient relationship for horizontal pipeline is given by:

$$u_w \text{ ft/s} = P_s^{0.4}$$

where P_s is the saturated vapour pressure associated with the water temperature, in psia. So long as this velocity is not exceeded, the water can be transmitted without employing expensive - and trouble prone-pumps.

INTRODUCTION

Most geothermal projects throughout the world appear to be roughly similar to Wairakei in that fluid flowing from wells is a steam-water mixture which has to be separated at the surface in order for steam alone to be transmitted to turbines to generate electric power. Cyclone separators used for this duty are extremely efficient and only about 0.03% of carryover bore water enters the steam pipelines. However, even this small quantity of water which contains dissolved minerals, has to be carefully removed by a series of extraction pots along the lines, which in turn, create a potential problem of steam condensate corrosion. This has taken place in some of the so-called High Pressure lines at Wairakei which operate at 125 psia; both extraction pot design and corrosion control has been described and solutions proposed, James (1975, 1979). At present, therefore, we may cautiously state that the design of steam transmission of the separated hot water is a seemingly intractable problem.

Visitors to Wairakei will observe that this difficulty has been completely avoided by the simple technique of discharging the fluid direct to the atmosphere from the separator water outlet. Of course, "flashing" of the hot water into a steam-water mixture takes place and a twin-tower atmospheric separator is also installed in order to control these fluids, so that the steam is vented to the atmosphere and the water - now at close to 100°C - is disposed of by means of open concrete channels overland to the Waikato river, a distance of several kilometers, Haldane and Armstead (1962).

These open-air channels gradually choke with deposits of silica and other associated minerals and require a tedious and expensive cleaning programme but otherwise function well.

Nowadays however, it is considered untenable to reject the separated well water to a river, and throughout the world the emphasis is mainly on reinjection of this fluid back into the subterranean reservoir, or at least, somewhere underground in the periphery of the geothermal field. Arguments are still raging into exactly where and at what depth etc, but acceptance is fairly widespread into its general inevitability.

The great difficulty on transmitting the hot water leaving the separators is that it is precisely at the boiling point for its pressure and hence any fall in pressure would result in a quantity of steam being generated which would increase the volume of fluid flowing and render extremely hazardous the whole design of the system. For example, at 85 psia, saturated hot water (water just at the boiling point for pressure), would produce 1% by weight of steam if its pressure falls to 75 psia and this would result in a volumetric expansion of more than three times its original volume of all-water. In other words - and more accurately - the steam would now consist of about 77% by volume of the fluid flowing. Pressure drop, is of course, inherent to the flow of fluids, and in the case of long pipelines would be almost wholly due to friction, as care would be taken to avoid intense restrictions such as orifice plates or chokes, or even sharp bends and loops.

The term intractable was used specifically for the transmission of water which is at the boiling point. It is possible to expensively overcome the difficulty by pressurising the fluid by the use of pumps coupled with header tanks, and for added safety, the injection into the hot water line of slightly cooler water - the attemperation approach - in order to inhibit the chance of any boiling occurring. This method is the one which was used at Wairakei for transmission of separated hot water along the 17 inch diameter "H" line over a distance of about 2 km from a part of the borefield to the power house, as described by Smith (1958).

In the case of saturated hot water being piped overland for reinjection purposes, however, there is a difference which proves significant in that the pipeline would not be insulated, and hence the water would decline in temperature <u>en route</u>. The pressure at which the water boils will therefore also decline along the pipeline. Frictional pressure-drop (an inevitable concomitant of flow) will also produce a fall in pressure along the line which if it does not exceed that due to heat loss will produce a condition in which no boiling will take place. If these two conditions are brought into a state of quasi-equality, the hot water can indeed be transmitted at very close to its boiling point and the pipeline designed for the condition of all-water.

SATURATED HOT WATER TRANSMISSION

The approach here is to first estimate the heat loss to the atmosphere along the pipeline, then to convert this to the temperature decline of the all-water flow. Then convert this temperature-drop to an equivalent reduction in the saturated vapour pressure of the water. This in turn is equated to the frictional pressure-drop along the line to determine conditions at which these factors are in balance and in which incipient ebullition is imminent.

Over the likely range at which boiling water will be transmitted for reinjection or other purposes, which is from 20 to 200 psia, we obtain from Lyle (1947) the heat loss for bare pipe:

$$H_L = 102.75 \quad P_s^{0.437} \qquad \text{Btu/ft}^2\text{h}$$

where P_s is the saturated vapour pressure, psia. The specific heat of boiling water over the pressure range above is roughly given by:

$$S = \frac{P_s^{0.031}}{1.107} \qquad \text{Btu/}^\circ\text{F}$$

The ratio of pressure change to temperature change at the boiling points is given by:

$$R = \left(\frac{P_s}{66.6}\right)^{0.7784} \qquad \text{psi/}^\circ\text{F}$$

If we take the pipe diameter as d inches and the water flow-rate as W lb/h, then the pressure-drop over 1 ft length of pipeline is calculated as follows:

$$\Delta P_t = \frac{R\ H_L \left(\frac{\pi\ d}{12}\right) 1{,}0}{W\ S} \qquad \text{psi/ft.}$$

$$W = G\ \left(\frac{\pi}{4}\right)\left(\frac{d}{12}\right)^2\ 3600 = \left(\frac{u_w}{V_w}\right)\ \left(\frac{\pi}{4}\right)\ \left(\frac{d}{12}\right)^2\ 3600 \quad \text{lb/h}$$

where

G = flow in lb/ft^{2}s

u_w = water velocity in ft/s

V_w = water specific folume in ft^3/lb = 0.018 ft^3/lb

Substituting these various factors in the pressure-drop equation:

$$\Delta P_t = \frac{P_s^{1.1844}}{960.57 \quad d \qquad u_w} \qquad \text{(i)}$$

The frictional pressure-drop ΔP_f psi/ft is calculated as follows:

$$\Delta P_f = \frac{u_w^2 \qquad f}{V_w \quad (13.92)^2 \quad d} \quad \text{psi/ft.}$$

where f is the Fanning Friction factor. For commercial steel pipe, f is calculated from the Reynolds Number R_e as follows:

$$f = \frac{0.0344}{R_e^{0.1505}}$$

$$\text{where } R_e = \frac{124 \; G \; d}{\mu_w} = \frac{124 \; u_w \; d}{\mu_w \quad V_w}$$

We assume an average value of the water viscosity u_w = 0.17 c'poise as valid over the range of pressures 20 - 200 psia and water velocity as 6 ft/s. Reinjection pipelines will be about 12 inches diameter.

$$R_e = \frac{124 \quad 6 \quad 12}{0.17 \quad 0.018} = 2.92\ (10)^6$$

$$f = \frac{0.0344}{[2.92\ (10)^6]\ 0.1505} = 0.00366$$

$$\Delta P_f = \frac{U_w^2 \quad 0.00366}{0.018\ (13.92)^2\ d}$$

$$\Delta P_f = \frac{u_w^2}{952.95 \quad d} \qquad \text{(ii)}$$

Equating (i) and (ii), $\Delta P_t = \Delta P_f$

$$\frac{P_s^{1.1844}}{960.57\ d\ u_w} = \frac{U_w^2}{952.95\ d}$$

$$u_w = \frac{P_s^{0.3948}}{1.00266}$$

Taking into account the various slight inaccuracies inherent to this approach, we can take the maximum velocity acceptable for design purposes as:

$$U_w = P_s^{0.4} \quad \text{ft/s} \qquad \text{(iii)}$$

The velocities calculated from this equation are not very different from "normal" velocity of cold water flow in pepelines which is often taken as about 6 ft/s for moderate pressure-drop and pump power requirements. From the above equation (iii), boiling water velocities at say 65 psia and 165 psia are 5.3 and 7.7 ft/s respectively and should not be exceeded for horizontal pipes.

Mention should be made that the heat loss equation is based on factory conditions where the ambient temperature is 70°F (18.5°C) with no wind. For overland pipelines where wind and lower temperatures - as well as occasional rain - prevail, higher water velocities should be permissible before boiling can occur, hence for horizontal pipes, the velocity derived from equation (iii) should not be reduced to be on the safe side, as an inherent safety margin is already contained in it.

As completely flat ground is unlikely to be found in practice the best arrangement would be to select the sites for separators as somewhat uphill from that of injection wells. This will insure that further safety is built-in and even higher hot water velocities could be allowed.

CONCLUSIONS

As all equations used in this study are empirically based, there is little reason to doubt the general correctness of the approach. Equation (iii) can be used with confidence to calculate the boiling water velocity in overland pipelines because the heat loss in practice will exceed that used in this study which was based on in-door conditions. Also horizontal ground is unlikely to be found in a geothermal field and hence separators will be located up-hill from injection wells, which would permit higher water velocities to be used with impunity.

This rather deceptively simple approach will provide considerable economic and maintenance advantages in that pumps and various ancillary equipment will not be necessary. Because of the importance of not discharging bore water into a river or on the land surface, electrically-operated centrifugal pumps would no doubt also require Diesel-driven standbye pumps in case of electrical failure, thus adding to the cost.

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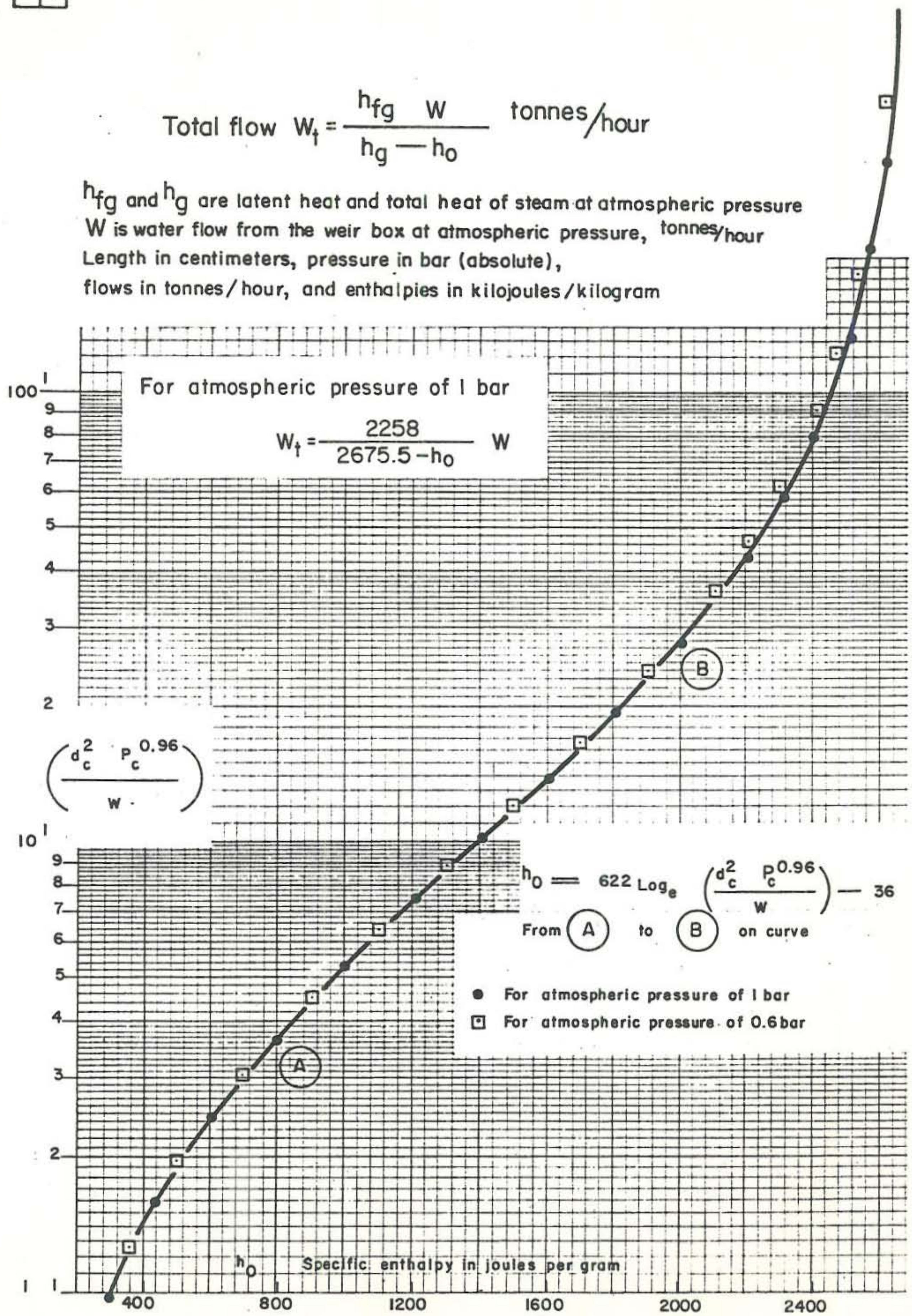

Flow and enthalpy of discharging well determined using Lip pressure pipe and Weirwater flow. Russell James (1986)