GEOTHERMAL HEAT TRANSFER - PLANT WORK AND FUTURE CHALLENGES

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ABSTRACT

Heat transfer from geothermal fluids involves problems of condensing steam containing non-condensible gas, scaling due to silica deposition and two-phase flow systems. A current chemical engineering project involves studying these problems in relation to advancing geothermal heat exchanger design and operating procedures.

Work implemented includes the performance testing of two production geothermal heat exchangers at Kawerau and Rotorua and experimental measurements on a pilot plant at Broadlands(this work is described in the second paper). Modern computer programs employing the latest shell-and-tube heat exchanger technology have assisted the evaluation of the results.

The future challenges for geothermal engineers in heat exchanger plant design are examined for four applications; process steam generation, hot water heating, hot air heating and total flow (two-phase) units.

INTRODUCTION

All new industrial geothermal projects will involve the design, fabrication, operation and maintenance of heat exchanger plants. In direct end-use heating applications the primary geothermal heat exchanger will be one of the important items of equipment in process steam generation, hot water heating or air heating. Reliable operation of an efficient geothermal heat exchanger leads to sound economics, overdesign is wasteful in both capital and operating expenditure.

Use of geothermal fluids in surface heat exchangers poses a number of problems, solutions to must be incorporated into design and operating procedures:

* Separated steam - The presence of non-condensible gas requires use of partial condenser design methods and provision must be made for safe venting.

* Separated water - The silica supersaturation is responsible for hard scaling of the heat exchanger surfaces which must be cleaned.

* Total flow - The two-phase flow system uses more complex design methods. Currently these are imprecise.

In 1980 DSIR (Department of Scientific and Industrial Research]& MWD(Ministry of Works and Development) engineers recognised the need to develop a base of performance data on geothermal heat exchangers in order to improve the design of new plants. Figure 1 shows the 'state-of-the-art' of the heat exchanger design capability in New Zealand prior to commencing the geothermal heat transfer project when the process design of any large heat exchanger was normally done overseas. No design fouling factors were available nor had effective cleaning procedures been demonstrated. "Can we use standard shell-and-tube heat exchangers for geothermal duties or do we have to develop novel exchangers similar to the liquid fluidised bed and direct-contact types being field tested overseas?" Questions like this could not be answered.

A DSIR/MWD engineering project team has been working in three main areas:

1) Plant work - The performance of two geothermal heat exchangers. in service has been measured.

2) Field tests - A heat exchanger test rig has been operational at Broadlands.

3) Computer programs- DSIR membership in Heat Transfer Research, Inc (HTRI) has given access to modern methods for the performance analysis of shell-and-tube heat exchangers.

This paper reviews the main implications of the plant work with use of the computer programs and examines the challenges in four future geothermal heat exchanger applications. The field tests are described in the second paper.

PLANT WORK

A. KAWERAU - Kettle Steam Generator

Plant Description

Two kettle boilers (Nos. 3 and 4 geothermal heat exchangers) were installed at Tasman Pulp and Paper Co., Limited in the early seventies to generate clean process steam at 4.7 bar, 149°C (50 psig) from geothermal steam at 7.8 bar, 169°C (100 psig) containing 3% w/w (1.2% mole fraction) non-condensible gas, predominantly CO_2. Each boiler was sized for a heat toad of 17.3 MW and a process steam production rate of 27.3 t/h. Geothermal steam condenses in the tubes. The tube bundle consists of 1700, 19.05 mm od carbon steel tubes on a 25.4 mm, 90° pitch with two tube-passes. The tubes are 6.1 m long and the heat transfer surface area is 619 m^2 (outside). These kettle boilers have given a satisfactory performance once the precipitation fouling on the process steam side had been resolved by correct water treatment. Scale is removed from outside the tubes by high pressure water blasting at the annual maintenance period but there is no cleaning of the inside of the tubes where corrosion products have slowly accumulated.

Performance Testing

The standard practice for performance testing of any production heat exchanger is to measure the terminal temperatures, the flowrates and the pressure drops. Data are normally considered reliable if heat balances better than a ± 10% error can be achieved.

Attempts to achieve this target have not been consistently successful due to the limitations in plant instrumentation and a fluctuating load. However performance data taken at start-up after a plant maintenance period are available as an indication of the observed performance of these heat exchangers.

Results

The process conditions were as shown in Figure 2. The HTRI condenser rating program has been used to check the performance of this partial condenser. The geothermal steam flowrate was regarded as being the more reliable so the treated water flowrate was relaxed to complete the heat balance.

Heat load	-	22.5 MW
Process steam	-	35.6 t/h
Effective ΔT	-	18.8°C
Observed U	-	2000 W/m^2°C
Calculated clean U	=	4000 W/m^2°C

Thermal resistances from overall observed performance:

Boiling in Shell	Tube-wall	Condensing in Tubes	Total Fouling
22%	8%	20%	50%

Design U = 1640 W/m^2°C when total fouling resistance is 60%.

Temperature and condensing coefficient profiles are shown on Figures 3 and 4.

Calculated tubeside ΔP = 8 kPa.

Results Analysis

Temperature profile

The bulk temperature profile of the geothermal steam in the tubes is as expected for a partial condenser (a condensing vapour/non-condensible gas mixture) showing the decline in saturation temperature as the non-condensible gas composition increases and the pressure decreases. With a boiling coolant, the ΔT driving force will also decrease through the heat exchanger and in this case the ΔT goes from 19°C to 12°C.

Condensing coefficient profile

The condensing heat transfer coefficient is also expected to decline (this corresponds to an increasing resistance to heat transfer) as the gas composition is increased and the vapour/gas velocity is reduced so changing the controlling regime from shear to gravity. Condensation takes place in the gravity controlled regime towards the exit of the second tubepass and operation in this regime should be minimised where possible.

Vented steam with the non-condensible gas

All the gas must be vented continuously or else the performance would rapidly fall due to the accumulation of non-condensible gas. In this case 94% of the steam fraction has been condensed so 6% of the entering steam is exhausted with the gas stream.

Heat duty

The observed performance is better than design which of course should be the case after a shutdown for cleaning.

Thermal resistances

Half of the total thermal resistance is due to fouling. Therefore this heat exchanger can be said to be fouling controlled. The tubeside thermal resistance due to fouling from the accumulation of corrosion products now contributes a significant thermal resistance (the outside of the tubes was cleaned by water blasting at the shutdown). Visual analysis of the iron sulphide scale removed from the tubes at the condensate exit shows a layered effect corresponding to a buildup following each start-up. The mechanism appears to be due to the changed environment at shutdown when the scale could dry and crack and so has to reform a protective film on contact again with the geothermal Steam.

Tubeside pressure drop

The geothermal side pressure drop is satisfactory with 23% loss in the nozzles.

B. ROTORUA - Tubular Hot Water Heaters

Plant Description

Banks of three shell-and-tube heat exchangers in series operate at the Forest Research Institute to raise a water heating circuit temperature by 10°C from 70°C to 80°C by using a design total geothermal supply at 167°C, 7.4 bar (90 psig). The geothermal two-phase supply is essentially water but contains up to 1% w/w gas which flows through the tubes. A bank is sized for a heat load 0.6 MW and 50 t/h hot water circulation on the shellside.

The tube bundle consists of 26, 31.75 mm od carbon steel tubes on a 46.0 mm, 60° pitch with two tube-passes. The fixed bundle is 2.5 m long with a total 19 m^2 (outside) surface area for the three units. A longitudinal baffle is used on the shellside with no segmental baffles.

The units are easy to fabricate, robust and reliable. At annual shutdown in the summer, any loose pumice is taken out by a wire brush pushed through the tubes. The thin hard silica/corrosion product scale inside the tubes is not removed.

Performance Testing

Difficulties have again been experienced in achieving the ±10% heat balance due to limitations in plant instrumentation and problems in measuring inlet geothermal enthalpy and gas content.

Typical summer performance data are available from after a shutdown and these are given as indicative of the actual performance of these heat exchangers.

Results

The process conditions shown in Figure 5 have been used to check the performance on the HTRI computer program for single phase heat transfer.

Heat load	=	0.21 MW
Hot water	=	14.4 t/h
Effective ΔT	=	18.1°C
Observed U	=	630 $W/m^2{}^\circ C$

1) Calculated U based on water only in the tubes = 500 $W/m^2{}^\circ C$ (clean U = 610 $W/m^2{}^\circ C$)

2) Calculated U based on water and gas in the tubes = 630 $W/m^2{}^\circ C$ (clean U = 770 $W/m^2{}^\circ C$)

Thermal resistances for case (2)

Water in Shell	Tube-wall	Water/gas in Tubes	Total Fouling
45%	5%	30%	20%

Results Analysis

Heat load

During the summer, the design heat duty is not required and the heat exchanger is operated in a turn-down situation.

Enhancement due to gas flow

Comparison of the actual overall heat transfer coefficient with the coefficient based on water only in the tubes indicates a conservative design. (Calculated clean U < Observed.) The flow of gas with the geothermal water will improve the heat transfer coefficient by raising the turbulence in the bulk and reducing the liquid film. Enhancement of the single phase coefficient by 1.5 to 2 times by the gas flow is supported by the field test work and a correlation in Shah. An increase in the tubeside film heat transfer coefficient by 1.7 times from 1600 to 2700 $W/m^2{}^\circ C$ gives a more accurate comparison of the observed and calculated overall heat transfer coefficients. (Calculated clean U > Observed.)

Thermal resistances

The main resistance is on the shellside due to the low velocity, longitudinal flow. Use of vertical segmental baffles will increase the velocity and promote cross flow which is the desired flow pattern on the shellside. The shellside coefficient at 3000 $W/m^2{}^\circ C$ (ΔP = 200 kPa) could be increased to 8500 $W/m^2{}^\circ C$ (ΔP = 100 Kpa) using segmental baffles in place of the single longitudinal baffle for the design flowrate, with also a saving in pressure drop.

FUTURE CHALLENGES IN GEOTHERMAL HEAT EXCHANGER PLANT DESIGN

1) Process Steam Generation

One challenge is to design for a lower ΔT driving force from the current 20°C. Maybe an effective ΔT of 10°C should be the target even though large heat exchangers will be required.

Vertical units offer the prospect of more efficient operation at a lower ΔT and better overall heat transfer coefficients (reduced fouling at higher velocities) but these units would be more expensive than kettle boilers.

Preheating the water feed to a boiler is essential for economy reasons. Use of the rejected total flow (steam/gas/condensate) in a preheater should lead to 99% of the geothermal steam fraction being condensed. New plant planned for the No. 5 geothermal heat exchanger unit at Tasman will confirm this design philosophy.

Carbon steel is the primary material of construction but there is probably scope to improve shutdown procedures inorder to minimise the longterm build-up of corrosion products.

2) Hot Water Heating

All the field tests and plant performance data (Wairakei, Broadlands and Rotorua) confirm that a surface heat exchanger using the separated water directly (no ponding or atmospheric contact), does not cause a significant fouling problem. The geothermal side fouling factor due to silica deposition is likely to be lower than the process-side fouling. The Broadlands results indicate that silica deposition would contribute only 15% to the design surface area of a water/water heat exchanger which is quite acceptable. The Broadlandspilot plant cleaning trials demonstrated that the silica/corrosion product scale could probably be removed using cold strong acid treatment.

Another challenge is to use the separated geothermal water which is currently wasted, in a surface heat exchanger application. Plans are being prepared to use separated water from Wairakei flash plant 6 to heat a closed hot water system for the Wairakei MWD buildings. The design heat load will be 1.5 MW.

Both carbon steel shell-and-tube or stainless steel plate heat exchangers can be used for this geothermal/clean water application. Similar sized units would have comparable capital costs. Any benefits would be in operating costs and until some data are available on cleaning effectiveness and frequency for the different types it is impossible to say which is the better unit for this duty. The direct-contact type of heat exchanger should not be neglected for clean hot water generation as large units may have cost advantages. Six 10 MW units are currently operated in Iceland.

3. Air Heating

A well designed bank of finned tubes for air heating must present good heat transfer at low air-side pressure drop to minimise fan running costs. Cheap tension-wound carbon steel finned tubing is available. However it has many disadvantages including low fin thermal conductivity, a fin-to-tube bond resistance and often a limited life. The temperature approach (difference between the geothermal inlet temperature and hot air exit temperature) is high. An existing geothermal air heater uses a temperature approach of 60°C in heating air to 120°C.

Bimetallic finned tubes will have advantages. An aluminium on carbon steel tube could improve the temperature approach to 30°C for the same pressure drop. Will geothermal process engineers accept the challenge and demonstrate heating air to aver 150°C using geothermal fluids?

4. Total Flow Exchangers

At some locations,--use of the total flow may have cost advantages over the conventipnal separated flow plant. Field tests and plant performance data have demonstrated high rates of heat transfer. Fouling rate data is not available but fouling is not expected to be higher than for the single phases. The total flow concept appears to be technically viable. However design methods for the thermal and hydraulic performance of a heat exchanger are still uncertain. There may be problems caused by flow instabilities which have not been studied properly and a design procedure with a minimum AT (pinch condition) needs more attention.

DISCUSSION

This preliminary study has contributed to advancing the subject of geothermal heat transfer but there are still problems to resolve. A major step forward came with DSIR's membership in HTRI which enables the performance of shell-and-tube exchangers and banks of finned tubing to be analysed using computer programs for geothermal duties in New Zealand. There are many benefits, one of the most important is the direct interaction possible now between the process engineer, geothermal engineer, heat exchanger design specialist, and services engineer which should lead to well-designed heat exchangers being fabricated, installed and efficiently operated. Figure 6 summarises the current availability of design information for geothermal heat exchangers.

Future projects should continue to examine the fouling, cleaning and total flow aspects of geothermal heat transfer where only basic work has so far been completed. Even though the magnitude of the geothermal side fouling is likely to be less than the process side, there is still the need to learn more about the possible accumulation of fouling deposits, the effectiveness of cleaning methods and the optimisation of shutdown procedures. There is also the need to monitor production heat exchangers through commissioning/operation/shutdown/operation cycles especially for geothermal water in case the silica deposition rate changes appreciably after a particular cleaning method or shutdown. Field testing a total flow exchanger would help to verify design procedures in order to reduce the risk of excessive overdesign in a commercially sized unit.

When new plant is installed it should allow easier, more accurate and reliable performance testing of the geothermal heat exchanger. Only basic instrumentation and fittings are required for temperature, pressure and flowrate measurement. With today's technology it is possible to log the performance regularly and cheaply. So with some thought at the design stage and an understanding of the need for reliable performance measurements, production geothermal heat exchangers could provide us with invaluable operating data.

The challenges for the geothermal engineer are also there - lower ΔT's, lower temperature approaches, new materials, larger units, new cleaning methods, better shutdowns etc. Guesswork could be eliminated from design procedures by the use of computer programs and a sound performance data base. If development work proceeds, the new generation of geothermal plants will have heat exchanger systems operating at higher thermal efficiencies.

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Nomenclature

U	Overall heat transfer coefficient, W/m²°C
ΔT	Temperature difference driving force $(T_{hot}-T_{cold})$, °C
ΔP	Pressure drop, kPa.

Figure 2.

KETTLE STEAM GENERATOR

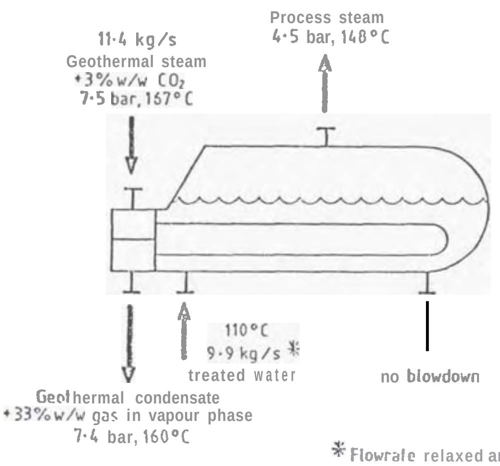


Figure 1 : STATE-OF-THE-ART OF DESIGN METHODS FOR GEOTHERMAL HEAT EXCHANGERS IN NEW ZEALAND IN 1980

Geothermal Flow	Film Heat Transfer Coefficient	Fouling Factor	Pressure Drop	Cleaning Procedure
Separated Steam	Time-consuming, unreliable manual methods. Tubeside only	No data for reliable design	Manual Methods	No test data
Separated Water	Reliable Manual methods, eg ESDU information etc Tubeside only	Limited field test data only	Accurate manual methods	No test data
Total Flow.	Very approximate manual methods only	No data	No data	No test data

Figure 3.

STEAM GENERATOR TEMPERATURE PROFILE

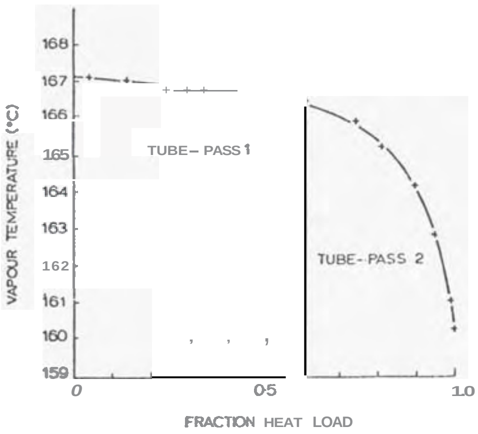
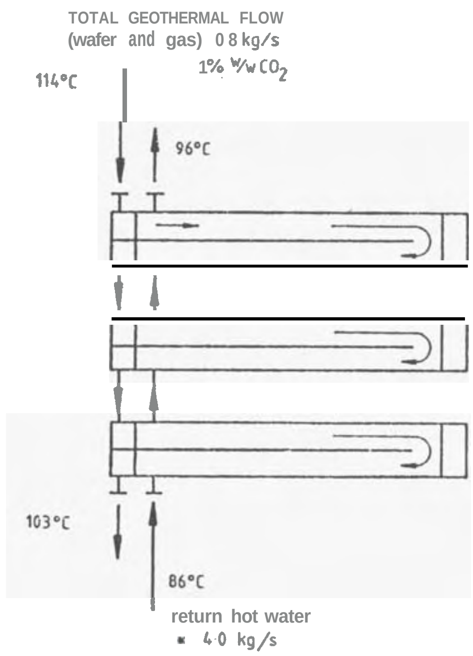


Figure 5.

TUBULAR HOT WATER HEATER.



* flowrate relaxed and calculated to complete heat balance.

Figure 4.

STEAM GENERATOR CONDENSING HEAT TRANSFER COEFFICIENT

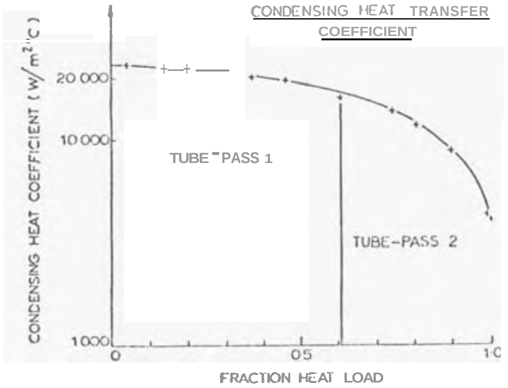


Figure 6 : CURRENT STATUS OF GEOTHERMAL HEAT EXCHANGER DESIGN METHODS IN NEW ZEALAND

Geothermal Fluid	Film Heat Transfer Coefficient	Fouling Factor	Pressure Drop	Cleaning Procedure
Separated Steam	Computer program for both tubeside and shellside performance. Supporting plant and field test data.	Some data. Planned field tests to examine effect	Computer program	Planned plant and field test trials
Separated Water	Computer program for both tubeside and shellside performance	Some data from plant and field tests	Computer program for clean performance	Some field test data
Total Flow	Basic plant and field data. Preliminary computer methods	Planned field tests, end 1982	Basic plant and field data. Preliminary computer methods	Planned field tests, early 1983