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**EVALUATION AND DESIGN OF DOWNHOLE
HEAT EXCHANGERS FOR DIRECT APPLICATION**

Annual Report, May 1, 1976—July 1, 1977

By
G. G. Culver
G. M. Reistad

August 1977

Work Performed Under Contract No. EY-76-S-06-2429

Oregon Institute of Technology
Klamath Falls, Oregon

MASTER

U. S. DEPARTMENT OF ENERGY
Geothermal Energy

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EVALUATION AND DESIGN OF DOWNHOLE HEAT EXCHANGERS
FOR DIRECT APPLICATION

Annual Report

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May 1, 1976 - July 1, 1977

Date Published - July 1977

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PREPARED FOR THE
U.S. ENERGY RESEARCH AND DEVELOPMENT ADMINISTRATION
UNDER CONTRACT NO. EY-76-S-06-2429

ABSTRACT

Oregon Institute of Technology and Oregon State University are jointly involved in an experimental and analytical research program to evaluate and improve the design of downhole heat exchangers (DHEs) for direct applications of geothermal energy.

During the past year, significant progress has been achieved in: (i) characterizing flows in wells with and without perforated casings and DHEs installed, (ii) determination of energy extraction rates for conventional DHEs in both cased and uncased wells, and (iii) utilization of information obtained in (i) to develop and substantiate models of heat transfer within the well.

Flow characterization was obtained by experimentally measuring temperature and velocity profiles.

In uncased wells, there was a relatively large vertical temperature gradient ($\sim 0.5^{\circ}\text{C}/\text{m}$) and insignificant measured vertical flow, while for the same wells when cased the temperature gradient was essentially zero ($\sim 0.03^{\circ}\text{C}/\text{m}$) and a large vertically upward flow was measured (9 to 14 cm/s as measured with a hot-film anemometer). These show that a significant connection cell is established after installation of the casing.

The energy extraction rates for a conventional DHE both before and after casing of a 60 m deep well was experimentally measured. In the uncased well up to 300 kW was obtained while the cased well produced over 500 kW.

Analytical heat transfer models have been developed for the cased well both with and without DHEs. For the cased well without a DHE, the model predicts flow rates that agree with the measured value within about 30 percent, which is considered to be very good in light of the number of unknown parameters. Further, the energy extraction rates predicted by the models agree even better with the measured values.

These models allow the energy extraction rates of design parameter variations to be evaluated. At this stage, only trends can be shown but with additional substantiation they should produce optimum energy extraction designs.

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I. Introduction

In 1976, the Division of Geothermal Energy of the U.S. Energy Research and Development Administration awarded a contract to Oregon Institute of Technology (OIT) to carry out a joint research program with Oregon State University (OSU) entitled "Evaluation and Design of Downhole Heat Exchangers for Direct Applications." Mr. Gene Culver of OIT is the Program Director and co-principal investigator. Dr. Gordon M. Reistad is co-principal investigator and manager of the OSU portion of this work.

The objective of the program is to examine the applicability and design of DHE systems.

This progress report concerns the principal results which have been obtained during the period May, 1976 to July, 1977.

In addition to the co-principal investigators, the personnel associated with the project during the first year are:

OIT: Mr. Doug Ford, Mechanical Engineering Technology, Student

Mr. Sherman D. Anderson, Electronics Engineering Technology Department, Associate Professor

Mr. Walter R. Richartz, Electronics Engineering Technology Department, Professor

OSU: Mr. David J. Churchill, Mechanical Engineering Department, Research Assistant

Mr. David B. Kreitlow, Mechanical Engineering Department, Research Assistant

II. Background

Figure 1 shows a typical installation of a DHE. It consists of (1) the wellbore, generally 15 to 36 cm in diameter and drilled with a cable rig, (2) a casing sealed to the wellbore at the upper end and perforated with slot-like perforations below this at two or more levels, and (3) an unfinned U-shaped heat exchanger made from bare steel pipe. The depths at which the perforations are to be located are determined when the wells are being drilled. The lower location coincides with depths where temperatures and the absence of fine dirt particles (in the bailing operation during drilling) indicates hot-water flows are present.

This type of system has operated satisfactorily in Klamath Falls for space and water heating of individual residences, businesses, and small industries for a long time. However, they have not been technically designed and evaluated. Furthermore, the reasons that the downhole heat exchanger is so readily used in Klamath Falls and almost never used elsewhere have not been identified.

It is the purpose of the work reported on here to examine the design of DHE systems in detail, and subsequently estimate the applicability of such systems relative to pumped or free-flowing systems.

III. Scope of Work

The two main objectives of the research project are:

- A. To characterize the circumstances under which the use of a DHE would and would not be economically desirable.

FIGURE 1. Typical downhole heat exchanger installation.

- B. To develop the procedure and critical inputs for designing a DHE in situations where its use appears desirable.

During the first year, the work has concentrated on measurement and modeling of flows and heat transfer within the well. Specifically, significant results have occurred in the areas of: 1) flow measurement, 2) DHE testing, and 3) heat transfer modeling. These results are described in the following sections.

IV. Flow Characterization and Measurement

At the beginning of this work it was felt that there were four basic possibilities for these flow patterns. These are shown in Figure 2.

1. Figure 2(a) shows flow in through perforations at one level from the aquifer at that level and out through perforations at another level to the aquifer at that level.
2. Flows may pass from the aquifer, through the perforations on one side of the casing, out the perforations on the other side of the casing, and back into the same aquifer they entered the casing from. This is shown in Figure 2(b).
3. Figure 2(c) depicts convective flow inside the casing in the absence of a heat exchanger. Hot water enters the casing through perforations in the bottom and rises, cooling slightly as it does so. At the top perforations the flow exits and heads down in the space between the well wall and the outside of the casing. The flow is cooled more because of the relatively cool well walls.

Figure 2. Possible flow patterns in well.

As it nears the bottom, however, the well wall temperature increases to the point where the water is heated again. The water reaches the bottom perforations, enters the casing, and the circuit is complete. This action is termed "thermosyphoning."

4. A case similar to the previous one, except for the addition of a DHE, is shown in Figure 2(d). Since energy is being taken from the well water by the DHE, well water cooling will be much greater now; the water will be cooled fast enough so that it flows down inside the casing and out through the bottom perforations. The water then rises up the outside of the casing, being heated at the same time. Reentry into the casing occurs at the top perforations, and the cycle is completed. These flows may occur by themselves or in combination.

It was necessary then to determine both the flow pattern and magnitude of the flow in order to model the heat transfer process.

To obtain flow pattern and general velocity magnitude, information at an early state in the project, preliminary experimental work involved: a) Temperature Measurements at various depths, and b) "Spinner" Measurements of vertical flow. Additional work on flow measurement was directed at a more exhaustive flow measuring program and consisted of: c) a Literature Search of flow measuring methods, d) Hot-Film Anemometer Measurements, e) Power Transistor

utilization for flow measurement, and f) Thermistor development for flow measurement. Each of these areas are discussed in the following paragraphs.

A. Temperature Measurements

Vertical temperature profiles were obtained in several wells. Measurements were made with a 300 k Ω Yellow Springs Electronics No. 44014 thermistor potted in epoxy. Data were taken for both cased and uncased wells. The most useful information was produced when a well was tested both before and after casing installation although this was not possible in all situations. Figure 3 shows water temperature as a function of well depth for a 58 m deep, 25.4 cm diameter wellbore before and after the installation of a 21.9 cm diameter casing. The uncased temperature profile is probably somewhat indicative of the ground's temperature profile. Notice the large difference in temperatures (in excess of 50°C) at the static water level (SWL) before and after casing installation. Well logs show that the region of relatively constant temperature at the bottom of the well corresponds to a layer of pumice with hot-water flow between 54.6 m and 58 m depth. The small top-to-bottom temperature difference after casing installation, compared to the large difference before installation, indicates the existence of large, vertical water movements in the cased well which tend to keep well temperatures nearly constant with depth. Similar results were obtained in another well 61 m deep, as shown in Figure 4.

Figure 3. Temperature as a function of well depth for well at 122 Hillside, Klamath Falls. SWL = 9.5 m.

Figure 4. Temperature as a function of well depth for well at 2130 Herbert, Klamath Falls. SWL = 10.4 m.

B. Spinner Measurements

To make preliminary measurements of vertical flow within the wells, a common instrument called a "spinner" was used. The spinner is a long, aluminum tube about 1 m in length with a 7.6 cm inside diameter. A four-bladed brass turbine is located inside the top end of the tube. The spinner is suspended inside the well from a steel cable connected to a winch with a depth-indicating meter. As water flows up or down the wellbore, some will flow through the tube and cause the turbine to turn.

Inside a small interchangeable canister connected to the top end of the turbine shaft there is either a magnetic or "cat's whisker" switch (both were available and used) which sends one electrical signal to the surface through a wire inside the steel cable, for each revolution of the turbine.

Calibration. The spinner was calibrated by suspending it in a 20 cm diameter, 3 m high vertical pipe. Water was passed through the pipe in the upward direction at various known flow rates. The maximum average velocity inside the standpipe which could be produced from the water supply was about 2.9 cm/s, which was insufficient to consistently rotate the spinner. To increase flow through the spinner tube, rubber foam insulation was wrapped around the outside of the tube to form a shroud that, together with the spinner, fit tightly in the calibration standpipe. Now nearly all of the water flowed through the spinner, rather than around it. This allowed the spinner to be calibrated from the point at which it would just barely turn consistently to velocities of 20.6 cm/s in the spinner tube.

Calibration curves of average velocity inside the spinner tube versus spinner revolutions per minute were obtained and are displayed in Figure 5. Different curves were obtained for the "cat's whisker" and magnetic spinners because of different blade pitches and also differing loads placed on the turbine by each of these switch types.

Figure 5. Calibration curves for Cat's Whisker and magnetic spinners.

The spinner was calibrated at water temperatures of about 13°C. Well temperatures, on the other hand, are much higher. Errors in flow measurements in the hot-water wells at about 100°C when using spinner calibration curves obtained at 13°C should be less than 5 percent, since the spinner is a volumetric flowmeter and 5 percent is the approximate change in water density between 13° and 100°C. Some calibration runs at low flow rates and water temperatures up to 90°C were made, but no significant difference between these results and those at 13°C were observed. Larger flow rates could not be tested because of limitations on the heat exchanger used for heating the water.

Well Tests. Realizing that the same difficulties would be experienced in well measurements as in calibration (i.e., insufficient flow through the spinner to cause it to rotate), a shroud was constructed to block water flow around the spinner and force it through the spinner. An aluminum collar with welded-on flange was slid tightly up against each side of a thick sheet rubber disk 20 to 30 cm in diameter, depending on the well diameter.

The results of the first well to be carefully measured with the spinner are shown in Figure 6. Average velocities in the 30.5 cm diameter well casing of up to 1.14 cm/s were recorded. Note that velocities are very low outside the zone between the top and bottom group of perforations, suggesting that water is flowing in and out

of the perforations. There was no spinner reading between the SWL (14 m) and the region near the shallowest set of perforations.

The reason for the increase in indicated flow with depth is unknown.

Figure 6. Spinner measurements from Assembly of God well,
Henry and William Streets, Klamath Falls.
SWL = 13.1 m.

Perhaps the shroud makes a tighter seal as the spinner descends into the well. It is also possible that well temperatures could affect the bearings in the turbine shaft, but this effect was never observed in the laboratory.

Because the exact reason for the increase in spinner reading with depth could not be determined, flow rates found from the calibration

curves may not be accurate.* Therefore, results are presented in terms of spinner revolutions per minute as well as average estimated velocity.

Figure 7 shows the results of a well both before and after it was cased. Before the well was cased, velocity was very low, nearly unmeasurable. In fact, this was the only uncased well measured that showed any evidence of vertical flow. After the casing was installed, flows over ten times larger than in the uncased well were measured. As in Figure 6, the cased well had no measurable flow until the region near or between the two perforation zones was reached.

With one exception, the absence of measured vertical flow in uncased wells suggests that there are only small, if any, flows from one aquifer to another inside the wellbore.

Limitations of Spinner. The results obtained using the spinner are useful because they indicate little or no vertical water movement in uncased wells but show that there must be sizable vertical

*Calculations based on observation of the tightness of the shroud seal to the inside of the casing show that up to about 10 percent of the flow could bypass the spinner tube by flowing around the shroud. Such flow would not be registered by the spinner, leading to an underestimation of the flow. In an uncased well up to 20 percent of the flow could leak around the spinner. These calculations are based only on cross-sectional areas available for flow and do not take into account any effects which may be caused by pressure drop across the spinner.

Figure 7. Spinner measurements from well at 2130 Herbert, Klamath Falls. SWL = 10.4 m.

flows present in cased wells. However, the shroud blocks the well, thus possibly altering natural flow patterns inside the well. Also, the spinner cannot detect horizontal flows.

C. Literature Review

To find a flow measuring device which would overcome the two main spinner limitations--blockage of the wellbore and inability to detect horizontal flows across the bore--a literature search was undertaken to find a more appropriate fluid flow or velocity measuring method. Pressure-related devices such as pitot tubes, electromagnetic flowmeters, inferential meters, tracers, and thermal methods such as hot-film and hot-wire anemometers were considered. A summary of the results of the literature review is presented in Appendix A.

The literature review indicated that a hot-film anemometer would be one of the better potential measuring methods for this application. It can detect both horizontal and vertical flows, and should be sufficiently rugged to withstand well conditions. Reorientation of the probe is not necessary for velocity measurements, except for rotation. It is small enough so that flow disturbance will be kept to a minimum. Separate calibration for each well diameter is not required. Hot-films have been widely used, as evidenced by the vast amount of literature on the subject. Furthermore, they are readily available.

D. Hot-Film Anemometer Measurements

The hot-film probe selected for detailed well measurements has two platinum, quartz-coated, hot-film sensors mounted on a stainless steel body, as shown in Figure 8. Figure 9 shows a photograph of the profile. Channel 1 will detect vertical flow, and channel 2 measures horizontal flow. The large size of the sensor should provide good fouling resistance.

Electronics supplied with the probe include a power supply, constant-temperature anemometer circuit, temperature-switching circuit, and a variable decade resistor. The temperature switching circuit permits fluid temperature measurement when the anemometer is in the well. The decade resistor sets the constant temperature (overheat) at which the sensor is to operate.

The anemometer cable is three-pair, twisted, No. 18 AWG stranded copper with copper braided shield. Two wire pairs lead to the two sensors. The other pair is shorted inside the probe body and is used for measuring the cable resistance. For ruggedness and high-temperature duty, polyurethane was selected for the cable jacket.

Calibration of Probe. Temperatures in the wells in Klamath Falls approach 100°C. Common probe overheats (ratio of sensor operating resistance to sensor resistance at fluid temperature) used in literature have been 1.05 to 1.10. The lower limit corresponds to a sensor temperature of 126°C (for channel 1) at a fluid temperature

Figure 8. Sketch of two-sensor hot-film probe. All dimensions are in centimeters. Drawing courtesy of Thermo-Systems, Inc.

Figure 9. Photograph of anemometer probe. Scale is in centimeters.

of 93°C, a common well temperature. Clearly there will be local boiling at the sensor surface, upsetting heat transfer from the probe, if the fluid is not placed under a pressure higher than the saturation pressure of water at 126°C. Gas bubble problems will also be alleviated by placing the fluid under pressure. In the well, water will be under pressure due to depth below the static water level. This pressure requirement resulted in the design of a pressurized calibration system.

The calibration system is shown schematically in Figure 10. Starting at the lower left-hand corner, water enters the square channel and flows through a 5 cm long honeycombed aluminum flow straightener mounted inside the channel. The water then flows along inside the channel, encountering the probe and also a copper-constantan thermocouple used to measure fluid temperature. After leaving the channel the water flows up through a 2.54 cm size Potter turbine-meter which measures calibration system fluid flow rate. The purpose of the pressurization and surge tank is to accommodate thermal expansion of the water and also any waterhammer effects which may occur if flow in the system is suddenly stopped. An air pocket about 20 cm high is left in the surge tank to allow sufficient room for expansion. The top of the tank is connected to a compressed air tank and regulator to permit system pressurization to about 350 kPa. (This gives a boiling point of 139°C, 13°C higher than the highest temperature at which the hot-film surface will be operated.) The 3.18 cm gate valve ("flow control valve") is the only means for flow control in the system since the pump operates

Figure 10. Anemometer calibration system. All dimensions are in centimeters.

at constant speed. The pump is capable of circulating up to 0.60 l/s in the system, corresponding to a maximum average velocity of about 15 cm/s in the square channel. After leaving the pump, water enters the tube side of a shell-and-tube heat exchanger, where sufficient thermal energy is added to the water to compensate for energy lost to the surroundings as it flows around the system. Steam flow through the heat exchanger can be varied with two valves. A pitot tube used for backup flow measurement is encountered after the water leaves the heat exchanger.

Photographs of the calibration system are shown in Figures 11 and 12. Figure 11 was taken looking along the channel in the flow direction. The location of probe entry, the short vertical turbine meter section, the pressurization and surge tank, and the pump are clearly visible. A portion of the heat exchanger can be seen just to the left of the pump. The protrusion from the turbine meter section is a gate valve and hose connection used to help drain water from the system. Figure 12 shows the upstream end of the channel. A thermometer for backup temperature measurement is visible at the left-hand edge of the picture, with the pitot tube just to the right of this. The city water source used to fill the system is located slightly to the left of the heat exchanger.

Channel 1 is calibrated as a function of temperature and average velocity in the flow channel with the probe mounted in the channel

Figure 11. Calibration system, looking along the channel in the flow direction.

Figure 12. Calibration system, looking at the upstream end of the channel.

so that the probe body's longitudinal axis is parallel to the flow channel axis. Because vertical flow in the well may be in either direction, this channel can be calibrated for both directions of flow parallel to the probe body axis. Channel 2 is calibrated as a function of average velocity by inserting the probe in a vertical position, sensor end first, through the top of the flow channel cover plate.

The probe is calibrated in terms of power dissipated in the sensor at a given velocity, sensor overheat, and fluid temperature. Refer to the calibration instrumentation in Figure 13 and the anemometer equivalent circuit in Figure 14. The bridge output voltage E of the anemometer circuit is measured instantaneously by the digital multimeter and averaged over a 10 second time interval by the voltage-to-frequency converter and electronic counter arrangement. The cable resistance R_c is determined from

$$R_c = R'_{ci} R'_{cs} / R'_{cs}, \quad (1)$$

where R'_{ci} and R'_{cs} are the cable resistances for channel "i" and the shorted cable pair, respectively, at 21°C, and R_{cs} is the resistance of the shorted pair measured at calibration time by the resistance bridge and d.c. null detector. The probe resistance R_p is found from the condition necessary for balance of the anemometer bridge circuit:

$$R_p = (R_1 R_3 / R_2) - R_c - R_e \quad (2)$$

Figure 13. Calibration instrumentation.

Figure 14. Anemometer equivalent circuit.

where R_e is the resistance between the anemometer bridge and the beginning of the probe cable, and R_1 , R_2 , and R_3 are anemometer bridge circuit resistors. Then, from circuit analysis, the power P_a dissipated in the sensor is

$$P = E^2 R_p / (R_c + R_p + R_3 + R_e)^2 \quad (3)$$

As stated above, it is desired to calibrate the probe for a given fluid temperature and anemometer overheat. As velocity is varied during a calibration run, however, it is difficult to control fluid temperature closer than $\pm 1^\circ\text{C}$ of its desired value. Also, variation in R_c causes variation in R_p . Both of these changes result in deviations in the temperature difference between the fluid temperature T_f and the sensor temperature T_s from desired values, changing the sensor-to-fluid heat transfer. Correction was made by observing that, all other variables held constant, heat transfer from the sensor (power dissipated in sensor) to the fluid is proportional to $T_p - T_f$. Let the subscripts "d" and "a" denote "desired" and "actual," respectively; the power P_d dissipated at the desired fluid and probe temperatures is then given by

$$P_d = P_a (T_{p,d} - T_{f,d}) / (T_{p,a} - T_{f,a}) \quad (4)$$

Since $T_{p,d}$ and $T_{p,a}$ will be within 2° to 3°C of each other, fluid property variations should be insignificant in the application of equation (4). This also applied to $T_{f,d}$ and $T_{f,a}$. It is also

assumed here that heat transfer from the sensor by natural convection is much smaller than that due to forced convection.

Nagib, Pluister, and Tan-atichat [23] showed that most of the effect of water temperature on heat transfer from the sensor can be removed by calibration at constant $R_p - R_{p,c}$, where $R_{p,c}$ is the sensor resistance at the fluid temperature. Good results at velocities up to 8 cm/s over a temperature range of 11°C were achieved. This method was used in our work.

Calibration results (Figure 15) show a plot of flow channel average velocity versus sensor power dissipation for water temperatures of 65.56° (150°F) and 71.11°C (160°F). The curves for the two different temperatures nearly coincide, as would be expected since the calibration method of Nagib et al. [24] was used. The data were taken by starting in the middle of the 65.56°C calibration curve and proceeding to lower velocities, and then returning to midrange and taking data at higher velocities. The 71.11°C curve was obtained by beginning in the middle, calibrating large velocities, and then returning and going towards smaller velocities. The time taken for calibrating half of the velocity range at each temperature was about 45 minutes, with the two complete runs being made in 3 hours. The 65.56°C calibration curve shows evidence of drift between 8 and 10 cm/s (data points taken 45 min apart). There is no evidence of drift in the 71.11°C curve.

Figure 15. Average calibration channel velocity versus sensor power dissipation for channel 1 (old probe).

After a number of calibration runs with this probe, carried out between November, 1976, and January, 1977, the probe's channel 1 resistance suddenly began exhibiting signs of instability. The sensor's resistance would vary several ohms in only a few minutes. Such changes clearly rendered channel 1 useless since calibration is dependent on accurate control of the sensor resistance.

The anemometer probe was returned to the manufacturer (Thermal Systems Inc., TSI) for examination and replacement.* TSI could not determine the reason for failure; no manufacturer- or user-caused damage could be found, although they confirmed the report of the unstable channel 1 resistance. A new probe was built, with calibration resuming again in April, 1977, and extending through May.

Due to time limitations after the replacement probe was received, and a concern about the probe life, both channels were calibrated only for water temperatures of 87.8°, 93.3°, and 98.9°C.

Figures 16 and 17 show plots of average velocity in the calibration channel versus sensor power dissipation. Necessary corrections to the sensor power have already been made.

*It should be noted that if a sensor on our probe was damaged, both sensors and the probe body had to be completely rebuilt.

Figure 16. Average calibration channel velocity versus sensor power dissipation for channel 1.

Figure 17. Average calibration channel velocity versus sensor power dissipation for channel 2.

There are two features in these results not present in those shown in Figure 15, or not present to as great an extent: the scatter of the data is larger, but still clearly shows the necessary grouping trend. For each channel there were some measurements taken at the same temperature on a different day (refer to Figure 16, channel 1). Two data points for 93.3°C at higher velocities have noticeably lower power dissipation than data forming a complete run on another day. In Figure 17 the same occurrence is seen for data at 98.9°C. Because the deviant results were obtained after the probe had been in the water for a period of about two hours, drift was suspected.

To test this hypothesis, the anemometer was tested by holding fluid velocity and temperature constant, leaving it in the RUN position (sensor heating current ON), and recording calibration data every few minutes until the probe had been in the water for about two hours. Channel 1 was tested at 87.8°C and channel 2 at 98.9°C, with the fluid velocity for each being about 4 cm/s. Results are shown in Figures 18 and 19 in terms of percent velocity deviation from the first data point in the drift test versus time since immersion for channels 1 and 2. Because it took between 45 and 60 min from the time of probe immersion in the water until data could be taken (due to calibration system preparation), these deviations are not based on a clean probe. It appears that the rate of drift (change in percent deviation with time) is higher nearer the probe immersion time. Because of the lack of data for the time immediately

Figure 18. Percent velocity deviation from first data point in drift test for channel 1. $T_{f,d} = 87.78^{\circ}\text{C}$, velocity = 4 cm/s.

Figure 19. Percent velocity deviation from first data point in drift test for channel 2. $T_{f,d} = 98.89^{\circ}\text{C}$, velocity = 4 cm/s.

after probe insertion in the channel and because drift depends on the particular contaminants in the fluid being used, it was difficult to draw conclusions about error caused by drift before well measurements were taken. The only thing that could be done would be to repeat measurements at the same well depth over a period of one or two hours and compare them.

Field Measurements. Much less instrumentation was used in the field at Klamath Falls than in the calibration laboratory. The authors felt that the operational amplifier-VFC-counter system for voltage averaging would not be necessary except under widely varying flow conditions. Also, it was felt that the methods of handling the anemometer in the well should be perfected before adding any auxiliary complicated equipment. The analog meter on the anemometer electronics was considered sufficient for anemometer output voltage measurement; a digital voltmeter was not used. A photograph of the instruments in the back of the van appears in Figure 20.

A 30 m length of anemometer cable connected the instruments at one end to a male octal connector which plugged into the winch. The female connector on the winch was connected to the cable length wound on the winch drum. This arrangement eliminated the need for a set of complicated slip rings.

The anemometer probe was placed in a steel cage so that it would not be damaged by contact with the casing wall or other

Figure 20. Instruments for field measurements.

surfaces. A photograph of the cage is shown in Figure 21. The cage was designed so that the anemometer would have free access to water flow in the well. A 15 cm long additional layer of heat shrink tubing was placed on the anemometer cable at the location of cage attachment to prevent damage to the cable. A steel winch was used to raise and lower the cable.

Temperature was measured at the anemometer locations by a thermistor on a separate cable attached at the top of the anemometer cage with electrical tape.

Figure 22 shows the equipment at the well. The thermistor winch is at the extreme right, with the anemometer winch slightly to the left. A depth-measuring indicator is located on the top of the well for the anemometer cable, and on the thermistor winch for the thermistor cable. To date, only a single cased well has been tested with the anemometer. The characteristics of the tested well are listed in Table 1.

Table 1. Characteristics of Tested Well

Location:	2130 Herbert, Klamath Falls
Depth:	61 m
Wellbore Diameter:	26.7 cm
Casing Gage:	250
Casing Diameter:	20.3 cm
Casing installed	from surface to bottom of well
SWL:	10.4 m
Perforations:	
Size:	1.3 cm by 15.2 cm
16 perforations	from 10.7 m to 13.7 m
28 perforations	from 55.8 m to 60.7 m

Figure 21. Photograph of anemometer cage.

Figure 22. Equipment at well.

Channel 1 Results. Data taken in the field together with the reduced results are presented in Table 2 for channel 1, the sensor used to measure vertical flows. Velocities were obtained by looking up the velocity pertaining to a given P_d on the calibration curve for 98.89°C (210°F) in Figure 16. The curve for this temperature was used because 98.89°C is close to the well temperatures. Power dissipations which indicated velocities less than 1 cm/s are marked as such, since 1 cm/s was the lowest velocity which could be used in our calibration system.

Because the pressures at depths where data were taken had to be greater than the saturation pressure at the probe operating temperature $T_{p,a}$ to prevent boiling of water at the sensor surface, no data could be taken at depths less than 34 m.

Velocities in the unperforated region seem to fall between 9 and 14 cm/s, with the average being about 12 cm/s. We can use this to estimate the mass flow rate in the well because the velocity profile in the well is probably quite flat, due to turbulent flow (velocity for $Re_D = 2000$ is 0.3 cm/s). A flat velocity profile of 12 cm/s corresponds to a mass flow rate of 3.9 kg/s. The analytical model carried out earlier (Section VI of this report) for a well with nearly identical characteristics, yielded a mass flow of 3 kg/s. The spinner measurements for the same well indicated 1.2 kg/s mass flow.

The authors suggested earlier that the spinner may serve as a significant flow blockage in the cased well because of the use of the shroud. This appears to be the case.

Some drift was observed in the channel 1 output. This was tested by making a measurement at 46.6 m depth after the probe had been in the water for 3.3 hours. The probe was then pulled out, cleaned, and lowered back to 46.6 m. The indicated velocity before cleaning was 9 cm/s and after reimmersion it was 14 cm/s. Two data points taken shortly after the initial immersion indicated 11 cm/s velocity at this same depth. These four data points are marked in Table 2 by an "a."

Estimates of vertical flow* in the perforation region (so noted in Table 2) are much more difficult to interpret. The readings connected with "+" in Table 2 indicate fluctuations between the voltage readings connected. It is important to note that these fluctuations were not encountered until the perforation region was reached. In the zone above the perforations, readings at any one point were relatively steady. This indicates the presence of considerable turbulence from water flowing into the casing through the perforations.

If power dissipation had increased markedly in the perforation zone, we would have interpreted this to mean the existence

*The entry under "velocity" in Table 2 is not, however, the vertical velocity in the perforation region. It is the purely vertical velocity that would give the equivalent power dissipation from channel 1 that the flow in the well is giving.

Table 2. Channel 1 field measurements.

Depth (m)	R_T (Ω)	R_1 (Ω)	E_A (V)	$T_{a,f}$ ($^{\circ}\text{C}$)	$R_{p,a}$ (Ω)	P_a (W)	P_d (W)	Velocity (cm/s)
36.6	6.0947	12.36	9.25	97.8	6.451	0.201	0.202	11
46.6 ^a	6.2072	12.48	9.35	98.4	6.459	0.205	0.203	11
46.6 ^a	6.2062	12.48	9.35	98.4	6.460	0.205	0.203	11
46.6 ^a	6.1853	12.48	9.1	100.7	6.481	0.195	0.192	9
46.6 ^a	6.1701	12.47	9.65	101.1	6.486	0.219	0.215	14
48.8	6.1685	12.45	9.3	99.7	6.468	0.203	0.204	12
48.8	6.2796	12.57	9.7	100.9	6.477	0.220	0.222	14
50.3	6.1737	12.47	9.45	101.2	6.482	0.210	0.210	13
51.8	6.1785	12.47	9.35	101.1	6.478	0.206	0.209	13
53.4	6.1905	12.49	9.3	101.1	6.486	0.203	0.200	10.5
53.4	6.2736	12.56	9.53	100.2	6.473	0.213	0.213	14
53.4	6.2776	12.57	9.7	100.4	6.479	0.220	0.217	14
54.9	6.202	12.50	9.4	101.3	6.484	0.208	0.207	12.5
55.5	6.2001	12.50	9.1	101.3	6.486	0.195	0.192	9
↓	↓	↓	9.6	↓	↓	0.217	0.214	14
56.1	6.2026	12.50	8.8	101.2	6.484	0.182	0.181	6.5
↓	↓	↓	9.4	↓	↓	0.208	0.206	12.5
56.7	6.2045	12.50	7.8	101.1	6.482	0.143	0.143	<1
↓	↓	↓	9.5	↓	↓	0.212	0.212	13.5
57.3	6.2065	12.50	8.6	101.1	6.480	0.174	0.174	5.5
↓	↓	↓	9.5	↓	↓	0.212	0.213	13.5

(Continued on next page)

Perforation zone

Table 2. (Continued)

Depth (m)	R _T (Ω)	R _I (Ω)	E _A (V)	T _{a, f} (°C)	R _{p, a} (Ω)	P _a (W)	P _d (W)	Velocity (cm/s)
57.9	6.2026	12.50	8.1	101.1	6.484	0.154	0.153	1
↓	↓	↓	9.0	↓	↓	0.190	0.189	9
58.5	6.2055	12.50	7.6	101.1	6.481	0.136	0.136	<1
↓	↓	↓	9.5	↓	↓	0.212	0.213	14
59.1	6.2076	12.50	7.7	101.2	6.479	0.139	0.141	<1
↓	↓	↓	8.5	↓	↓	0.170	0.172	4.5
59.8	6.2108	12.51	7.8	101.2	6.486	0.143	0.141	<1
↓	↓	↓	8.6	↓	↓	0.174	0.171	4.5
60.4	6.2120	12.51	7.7	101.0	6.484	0.139	0.137	<1
↓	↓	↓	8.2	↓	↓	0.158	0.155	1
61.1	6.2161	12.51	7.55	101.2	6.480	0.134	0.135	<1
61.4	6.2187	12.52	7.9	101.5	6.4877	0.147	0.145	<1

^aIndicates data points used for investigating sensor drift in the well.

of a strong crossflow, since vertical flow was expected to decrease as the probe was lowered further into the perforation zone. This decrease in flow with depth is easily seen in Table 2.

Direction of vertical flow in the unperforated region was found by lowering the probe slowly, recording the anemometer output voltage, and then raising the probe at about the same speed and recording the voltage again. The voltage was significantly higher when the probe was being lowered, indicating an upward flow. Since temperature profiles in cased wells with no DHE show increasing temperature with depth, this flow direction would be expected.

When the results of the error analysis were considered, it was found that the average vertical velocity falls between about 5 and 19 cm/s. However, we believe the actual value to be much nearer the average value indicated above because of the small scatter of the readings and because the power dissipations at the bottom of the well were what would be expected for near zero velocity.

Channel 2 Results. Data taken in the field together with the reduced results are presented in Table 3 for channel 2, the sensor used to measure horizontal flows. Velocities shown*

*The entry under "velocity" in Table 3 is not, however, the horizontal velocity. It is the purely horizontal velocity that would give the equivalent power dissipation from channel 2 that the flow in the well is giving.

Table 2. Channel 2 field measurements.

Depth (m)	R _T (Ω)	R ₁ (Ω)	E _A (V)	T _{a, f} (°C)	R _{p, a} (Ω)	P _a (W)	P _d (W)	Velocity (cm/s)
<u>First run</u>								
53.4 ^a	6.2765	13.24	7.40	100.6	7.051	0.136	0.133	2
53.4 ^a	6.2793	13.24	7.15	100.6	7.048	0.127	0.126	1.5
55.2	6.3014	13.27	6.2	101.2	7.056	0.096	0.094	<1
↓	↓	↓	8	↓	↓	0.159	0.156	4
average	↓	↓	7	↓	↓	0.122	0.120	1
55.2	6.3014	13.27	6	101.2	7.056	0.089	0.089	<1
↓	↓	↓	8.5	↓	↓	0.180	0.178	7
average	↓	↓	6.2	↓	↓	0.096	0.095	<1
Rotate								
55.2	6.2895	13.25	6.2	101.1	7.048	0.095	0.096	<1
↓	↓	↓	6.6	↓	↓	0.108	0.109	<1
average	↓	↓	6.4	↓	↓	0.102	0.103	<1
Rotate								
55.2	6.2895	13.25	6	101.1	7.048	0.089	0.090	<1
↓	↓	↓	6.4	↓	↓	0.102	0.103	<1
average	↓	↓	6.2	↓	↓	0.095	0.096	<1
Rotate								
55.2	6.2895	13.25	6	101.1	7.048	0.089	0.090	<1
↓	↓	↓	6.6	↓	↓	0.108	0.109	<1
average	↓	↓	6.3	↓	↓	0.099	0.100	<1
Rotate								
56.7	6.2870	13.25	6.4	101.0	7.050	0.102	0.101	<1
↓	↓	↓	8	↓	↓	0.159	0.158	5
57.0	6.3013	13.27	6.0	101.6	7.056	0.089	0.089	<1
↓	↓	↓	7.0	↓	↓	0.122	0.121	<1
Rotate								
57.0	6.2997	13.27	6.0	101.6	7.057	0.089	0.088	<1
↓	↓	↓	7.0	↓	↓	0.122	0.120	1
61.4	6.2131	13.18	5.8	101.2	7.055	0.084	0.083	<1

(Continued on next page)

Table 3. (Continued)

Depth (m)	R _T (Ω)	R ₁ (Ω)	E _A (V)	T _{a,f} (°C)	R _{p,a} (Ω)	P _a (W)	P _d (W)	Velocity (cm/s)
Second run								
50.3 ^a	6.0621	13.02	5.6	101.2	7.047	0.079	0.080	<1
↓	↓	↓	5.8	↓	↓	0.084	0.086	<1
54.9 ^a	6.1191	13.08	5.5	101.3	7.050	0.076	0.076	<1
↓	↓	↓	5.8	↓	↓	0.084	0.085	<1
55.5	6.1276	13.09	5.6	101.3	7.051	0.078	0.079	<1
↓	↓	↓	6.1	↓	↓	0.093	0.093	<1
56.1	6.1352	13.10	5.6	101.2	7.054	0.078	0.078	<1
↓	↓	↓	6.3	↓	↓	0.099	0.098	<1
56.7	6.1310	13.09	5.6	101.1	7.048	0.078	0.079	<1
↓	↓	↓	6.2	↓	↓	0.096	0.097	<1
57.3	6.1540	13.12	5.8	101.3	7.055	0.084	0.083	<1
↓	↓	↓	6.2	↓	↓	0.096	0.095	<1
57.9	6.1656	13.13	5.7	101.1	7.053	0.081	0.080	<1
↓	↓	↓	6.1	↓	↓	0.093	0.092	<1
58.5	6.1754	13.14	5.5	101.2	7.053	0.076	0.075	<1
↓	↓	↓	5.7	↓	↓	0.081	0.080	<1
59.1	6.1837	13.15	5.7	101.1	7.055	0.081	0.080	<1
↓	↓	↓	5.8	↓	↓	0.084	0.082	<1
59.8	6.1996	13.16	5.6	101.0	7.049	0.078	0.078	<1
↓	↓	↓	5.9	↓	↓	0.087	0.087	<1

^aIndicates data points taken outside the perforation zone.

were obtained by looking up the velocity pertaining to a given P_d on the calibration curve for 98.89°C in Figure 17 (as was done for channel 1). Power dissipations indicating velocities less than 1 cm/s are so marked, again as in Table 2.

The data from run 1 indicate a general increase in activity (turbulence) when the probe is moved into the perforation zone from the unperforated zone. (Perforations are located between 55.8 and 60.7 m.) There does seem to be an absence of high horizontal velocities.

The velocities shown represent minimum velocities (not considering error*) flowing past the sensor. The hot-film surface has a shape matching part of the circumferential surface of a cylinder. At the low Reynolds numbers of our flow (Re_D based on a hot-film cylinder diameter of 1.5 mm and fluid velocity of 7.2 cm/s is about 340), the most heat transfer will take place when flow impinges on the film perpendicular to the surface [3]. (The flow here is being modeled as flow past the leading edge of a cylinder in crossflow.)

The large variation in indicated velocities in the perforation zone could be explained by flow reversal or probe rotation.

*The error at 0.18 W power dissipation is about 0.014 W, corresponding to ± 5 cm/s.

Nusselt numbers (at $Re_D = 300$, based on hot-film cylinder diameter) averaged over the leading and trailing edges of a cylinder in crossflow have a ratio* of about 2.75 [25]. The observed heat transfer variations from the film have a ratio of 1.5 to 2.0. When one realizes that a significant portion of the power dissipated from the film is absorbed by the probe body and not by the fluid, the above explanation gains credence.

Data points marked with an "a" in Table 3 were not taken within the perforation zones. In run 1 these were steady readings (not fluctuating as in the perforation region) signifying upward flow with no turbulence. Data points marked "average" are the approximate average voltage readings that were visually observed. The word "rotate" indicates when the anemometer cable was rotated to try to detect crossflows.

Some of the readings from runs 1 and 2 resulted in sensor power dissipations that were off the lower end of the calibration scale. This could be due to the fact that the flow was much slower than in our calibration channel, since anemometer sensitivity (sensor power dissipation change per unit velocity change) increases as zero velocity is approached. The absence of high horizontal velocities during this run, however, would seem to be indicated.

*ratio is that of the maximum Nusselt number to the minimum Nusselt number. Reference [25] used $Pr = 0.7$, while $Pr = 1.9$ for the current fluid, water at $3^\circ C$. However, the influence of Prandtl number over this range is quite small [26].

Field measurement of the channel 2 resistance showed it to be 0.1Ω lower than expected at 100°C . When returned to OSU, the resistance was slightly more than 0.5Ω lower than expected. This does not explain the low field readings of run 2 because low probe resistance would lead one to set the overheat too high, resulting in large power dissipations and not small ones. But the failure of channel 2 to maintain constant resistance at a given temperature under both laboratory and field conditions leads to the conclusion that the results obtained using channel 2 are not entirely reliable. Overall, however, they do indicate flow coming in through the perforations because of the turbulence, and the absence of high horizontal velocities in the lower perforation zone.

Summary of Anemometer Field Measurements to Date. Vertical flow measurements yielded an average mass flow rate of 3.9 kg/s , in good agreement with analytical work, but three times larger than the spinner measurements. Channel 1 readings were fairly steady above the lower perforation zone, but showed significant fluctuations when the perforation zone was entered. It is believed that this signified turbulence due to flow entering through the perforations. Flow direction was established as being upward.

Horizontal flow measurements were less successful. Like channel 1, channel 2 showed evidence of turbulence in the

perforation region. We did not find evidence of high horizontal velocities. It was difficult to separate horizontal flow from vertical flow for two reasons: the long, narrow perforation slits allow only thin streams of water to enter, probably giving rise to erratic readings. Also, each anemometer sensor responds to both horizontal and vertical flows.

E. New Anemometer Systems

Since one of the main thrusts of this work is to provide the means to readily determine the economics and design information required to select the best DHE, we felt that instrumentation to obtain this information should also be readily available and simple in operation, as opposed to the hot-film anemometer which is expensive. Although not specifically included in the proposal, work was initiated early in the program to develop this instrumentation and has continued on a low-priority basis as time was available.

Based on the literature search for the selection of the most appropriate method presently available (which led to the selection of the hot-film anemometer), power transistors and thermistors were singled out for further investigation, since they are economical, readily available and require relatively simple electronic equipment for operation. These devices are also small and rugged; features that were considered desirable for this application.

Power Transistor Flow Meter

Power transistors were selected for initial investigation since the literature search indicated that although thermistors had been used successfully to measure small flow rates, they have not gained wide acceptance.

The bipolar junction transistor (i.e., the common transistor) displays a number of well-known and generally undesirable responses

in electronic circuitry to changes in temperature. The transistor is also claimed to be "the most repeatable nonlinear device known to physics." Selected transistor pairs from the same manufacturing lot will have nearly identical characteristics. This makes it possible, in anemometry, to use one transistor, shielded from flow, to compensate the other transistor of the pair for ambient temperature effects. Finally, the thermal resistance from the transistor collector junction to the exterior of the transistor case is quite low. This allows the thermal effect sensing element (the transistor die) to be tightly coupled to the medium (boundary layer).

A three-ampere medium-power, plastic-encapsulated, silica npn transistor, the 2N4922, was chosen for this application. This Motorola transistor comes in Motorola's 77-03 case which has a flat brass plate, electrically connected to the collector, for a heat transfer surface.

The flow meter was intended to be free-field type. The thermally insulated surface of the transistors was epoxied to each side of a 10 cm long, 1.6 cm wide brass strip.

Thermoplastic insulated lead wires were connected to the transistor lead wires. The connection region was vacuum-impregnated with epoxy. The collectors were electrically connected to the brass strip.

The brass strip was inserted in a 10 cm long, 1.6 cm ID plastic pipe. This divided the pipe into two volumes. One of these volumes was closed to flow, the transistor sensing only ambient temperature. The other is open to flow, that transistor responding to both flow and ambient temperature.

Power Transistor Anemometer Probe

Figure 23

The transistors were electrically connected as shown below.

Power Transistor Circuit

Figure 24

The collector currents were maintained at 100 mA. V_{ce_1} and V_{ce_2} were adjusted by V_{cc} to control power dissipation.

The device was placed in a 15 cm diameter vertical pipe, in which varying flows were created. The flow rates ranged from 0 to 5 cm/min. The voltage, ΔV , showed responses to changes in flow of the order of millivolts, indicating extremely high sensitivity. The dominant response to changes in flow occurred within 1 or 2 minutes; however, a second very slow response occurred over a time of about one hour. This effect was so complex that in order to investigate it the flow-meter was tested in vessels of still water by operating it at various power dissipation levels. The very long response times to changes in power input persisted. No experimental technique was devised to determine the reason for this response. One of the experimentors revealed that some users of plastic encapsulated transistors had experienced difficulty with these transistors in high humidity environments. For want of a better reason, we accepted the notion that the plastic encapsulated transistors were unsuitable to this water immersion application. Since hermetically sealed, metal encapsulated transistors did not offer the convenient geometry of the plastic encapsulated transistors, and work was terminated.

Thermistor Flow Meter

Since the flowmeter utilizing power transistors did not prove satisfactory, work was shifted to utilizing thermistors in a flowmeter.

A flowmeter transducer for use in deep hot-water wells must have the following characteristics.

1. Sensitivity--capable of providing reliable measurements at temperatures to 120°C at minimum flow rates of 50 mm per second.
2. Size--a diameter of 5 mm to 7mm maximum; a length of approximately 150 mm maximum.
3. Cost--the flow transducer must be considered expendable as it will often become entangled in well plumbing at depths of several hundred meters and not be retrievable. Replacement cost must therefore be consistent with the value of the data.

Consideration of these three major factors suggests implementation of the convection heat transfer technique. This is a well-known technique and is widely utilized in varying forms where thermal conductivity from a heat source is a function of the character of the surrounding medium.

$$T_{th} = T_a + \theta P_{th} \quad (5)$$

Where:

T_{th} = the temperature of the heat source

T_a = the temperature of medium surrounding the heat source

θ = the thermal resistance (°C/watt) of the thermal path of the heat source and the medium

P_{th} - the power (watts; joules/sec) delivered to the heat source

$$T_{th} - T_a \triangleq \Delta T \quad (6)$$

$$\theta = \frac{\Delta T}{P_{th}} \quad (7)$$

Since flow of the medium surrounding the heat source will manifest itself as a modification of θ , flow can be monitored by observing changes in θ produced by the flowing medium.

A thermistor was selected as the heat source for the convection heat transfer flow-measuring device. The thermistor satisfies the size and cost requirements with other satisfactory performance yet to be verified.

Since the thermistor resistance as a function of temperature can be accurately determined, is repeatable and predictable, it should provide suitable accuracy. Also, observation of two thermistor circuit parameters when in a "self-heat" mode will provide heat source temperature. When T_a is known (determined by either the same or another thermistor), ΔT is known and θ is readily determined for conditions of flow of the medium.

Fabrication of thermistor flow-sensing probes that are suitably responsive, electrically insulated, small in size and operable in a

"self-heat" mode with appropriate ΔT for accurate θ measurement requires application of as much art as science.

An experimental thermistor thermal resistance measuring test setup (Figure 25) is in operation for nominal 20°C water. It utilizes a thermistor resistance that approximates the same resistance range of the glass-encapsulated Fenwal UUT 45J1, which will be in the probe for θ measurements in geothermal water of 70°-100°C.

Figure 25
Block Diagram of θ Measuring Circuit

$$\theta = \frac{\Delta T}{P_{th}}$$

Preliminary tests show θ essentially constant at 190°C/W for the probe style in use over a ΔT range of 45° to 75°C in 20°C water.

Since test data to date indicate the experimental instrument may satisfactorily measure flow, thermistor probes suitable for use in 70° to 120°C geothermal water will be fabricated. No-flow tests will be

made to acquire thermal resistance and sensitivity data. If the probe and instrument tests continue to be satisfactory, preliminary flow tests will be carried out. Should the preliminary flow data warrant, the flow instrument will be calibrated using the thin film anemometer calibration facility at Oregon State University.

V. DHE Tests--Cased and Uncased Well

Although it has been known for many years that the use of a perforated casing smaller than the wellbore increased the output of a DHE, there was no information available about how much the output could be increased or what the mechanism was that provided the increase, except that it was theorized that the perforations allowed convection.

Temperature measurements made by plumbers and well drillers in cased wells indicated that temperature increased with depth (although the increase was much less in cased than in uncased wells), and it was assumed that hot water flowed up through the casing, was cooled and flowed down in the annulus between the casing and the wellbore. The few measurements made in wells with DHEs installed tended to confirm this theory. There were no known attempts to establish a maximum rate of energy extraction from any wells with DHEs installed.

In order to obtain information about the convection cell relating to energy balances within the well, tests were run on a well both before and after casing with a DHE installed.

The test well was privately owned and its use was donated for the one-week period of the test. Drilling with a cable tool rig was started August 2, 1976, and completed on August 5, 1976. The completed well depth was 65.5 m and .3 m diameter. A small flow of water was encountered at 17.4 m and the temperature estimated at about 90°C.

Water rose to a static level of 14.2 m. Drilling continued to 59.4 m, where water at 102.8°C was encountered. Static water level remained the same, indicating the upper and lower flows were interconnected in the surrounding strata. From 59.4 to 64 m drilling was through a "live water strata" where only large cuttings were removed with the bailer. This flow was considered large enough for the intended use of heating several small houses and an additional 1.5 m was drilled to provide a solid casing seat and allow room for sloughed material.

Temperature and flow velocity profiles were run immediately after removing the tools from the well (Figures 26 and 27). The well was allowed to stabilize until August 8, when temperature and flows were measured again with no change from the previous profiles. Flow was in the upward direction.

A. DHE Test Uncased Well

On August 8, a DHE consisting of 20 lengths of 5-cm black iron pipe (128 m) was installed and output testing began.

The test consisted simply of establishing a flow of city water at 21.1°C into the DHE while monitoring the DHE output temperature and flow rate, and the well water temperatures at the 15.2 m, 30.5 m and 61 m depths. A flow rate was continued until the DHE output temperature rate of change appeared to be approaching zero, at which time the

Temperature Profile

Gibbs and
Old Fort Road

Figure 26.

Figure 27.

flow was increased and the process repeated. Temperatures vs. elapsed time from initially starting the test were recorded. DHE flow rates were .76 l/sec, 1.30 l/sec, and 1.89 l/sec. Water temperature vs. elapsed time can be obtained from Figures 28, 29, and 30.

B. DHE Test Cased Well

After the testing in the uncased well was completed, the DHE was removed and a 203 cm full-depth casing was set in the well on August 8. The casing had been perforated from the 56.7 m depth to the bottom with 40 slots 1.3 cm wide by 15.2 cm long, and from the 15.5 m to 17.7 m depth with 16 slots the same size as the lower perforations. This perforation combination was recommended by the well driller and based on his experience. Temperature and flow measurements were taken immediately and had not changed 14 hours later (Figures 26 and 27).

Notice that after the casing was set, relatively large flows* were induced in the well (upward inside the casing) and the temperature was nearly constant for the total depth.

The DHE was installed exactly as before and the DHE output tests run again (Figures 31, 32, and 33). DHE flow rates for the second test were .74 l/sec, 1.4 l/sec and 2.6 l/sec. Well water temperatures were measured as before. (Note that the test was interrupted for 90 minutes. The temperature measuring probe was fouled at approximately 60 m depth and the DHE was lifted about 5 m to free it.)

*These are "spinner" flow measurements. Hot-film anemometer flow measurements, believed to be more accurate, indicate flows approximately three times as high.

Figure 28.

Figure 29.

*Temperature vs. Time Elapsed During DHE Test
Gibbs and Old Fort Road Not Cased Well*

Figure 30.

*Temperature vs. Elapsed Time During DHE Test
Gibbs and Old Fort Road Cased Well*

Figure 31.

*Temperature vs. Elapsed Time During DHE Test
Gibbs and Old Fort Road Cased Well*

Figure 32.

*Temperature vs. Elapsed Time During DHE Test
Gibbs and Old Fort Road Cased Well*

Figure 33.

Results of DHE Tests

As expected, the DHE in the cased well was able to produce a significantly higher output than in the uncased well (Figure 34). At the highest DHE flow rate production of the cased well was 175 percent of the uncased well, about 500 kW. However, it must be remembered that energy production would not be that high in the normal mode of operation since the ΔT across the DHE was 63°C , much more than is normal.

One of the more interesting results is that at elapsed time, 12 minutes, the direction of flow in the convection cell reversed. Note that prior to time 12 temperatures in the deeper part of the well were higher than those in the upper part, indicating upward flow, and after time 12 the reverse situation, indicating downward flow.

If the curves are studied carefully, certain perturbations in well water temperature can be noted, particularly shortly after changes in DHE flow rates, and these perturbations can be followed down the well with increasing time. There is a temptation to use these perturbations to estimate flow rates inside the casing; however, there is no assurance that the temperature probe is in the same position relative to the DHE at all depths, although it was probably in the same position for each of the measurements. Estimates made using

Figure 34.

the perturbations are so inconsistent they seem to be nearly useless, although there is some indication flow rates increase with increasing DHE output, a situation that is expected before the estimates are made.

The most valuable result of the DHE test is the agreement between the test results and the predictions based on the analytical model to be discussed in the next section of this report.

VI. Heat Transfer Modeling

After the spinner measurements discussed above were made, work was started on modeling the heat transfer within the well. The several models developed are all based on thermosyphoning as the major mechanism of fluid motion within the wells. The models allow the effects of several design parameter variations to be estimated. It must be noted that these models are preliminary at this time and should not be construed as completed.

Three thermosyphoning models are developed. The first model is for thermosyphoning in the cased well with no DHE. This thermosyphoning is due to conduction through the wall. Conservation of energy results in two simultaneous ordinary differential equations and momentum considerations lead to an algebraic equation. These equations lead to a single equation which is solved for the mass flow rate. Once this is known, all other quantities of interest can be calculated.

The second model includes a DHE. An energy balance leads to a two-point boundary value problem with four simultaneous ordinary differential equations. Parameters of the differential equation are dependent on the unknown mass flow rate through the casing. Momentum considerations lead to an equation for the mass flow rate in terms of the integral of the unknown temperature difference between the inside and outside of the casing. This problem was solved numerically using a computer program.

The differential equation model gives reasonably good results but is rather cumbersome. Since several questions dealing with the limitations of DHEs require coupling the DHE to other systems, an effort was made to develop a much simpler model suitable for this purpose as well as to investigate the effects of mixing at the bottom of the well, which is ignored in the other models.

The result of this effort is a set of simple models which model the fluid flow and heat conduction as a network of paths and nodes, the paths for fluid flow being modeled as flow-through conduits. The result of this formulation is a set of simultaneous algebraic equations. It appears that this approach can be readily extended to include the effects of the aquifer flow and thermosyphoning in the DHE, though that is yet to be done.

A. Thermosyphoning in the Well Without DHE

The preliminary measurements discussed above show substantial vertical flows in the cased wells (1 to 5 cm/s) while vertical flows in the uncased wells are generally insufficient to activate the vane anemometer used for measurements, which has a sensitivity of approximately .2 cm/s. These results lead to the question of whether thermosyphoning between the inside and outside of the casing due to the temperature gradient of the wall is sufficient to account for the large observed flow rates.

A mathematical model was constructed to answer this question and to determine the effects of various parameters on the mass flow rate.

The driving force for this thermosyphoning comes from the temperature gradient of the well wall. The casing apparently serves to isolate the upward and downward flows. This results in greatly increased flow rates and a nearly uniform temperature profile inside the casing after the well is cased, as illustrated above.

The simplified physical model consists of two concentric vertical impermeable cylinders representing the well and casing. The bottoms of the two cylinders are assumed to open into a relatively open reservoir. Near the top of the cylinders, which corresponds to the water level in the well, the inner cylinder has perforations that allow fluid to flow either to or from the outer cylinder.

The model was analyzed by assuming fluid properties and velocity to vary only in the vertical direction for both the inner pipe and for the annulus. Application of mass continuity, energy, and momentum laws then leads to the two simultaneous ordinary differential equations below, which can be solved in terms of the unknown mass flow rate, assuming that the wall temperature is known.

$$\frac{dT_o}{dx} = \frac{U_w \pi D_w}{C_p \dot{m}} (T_o - T_w) + \frac{U_i \pi D_i}{\dot{m} C_p} (T_o - T_i) \quad (8)$$

$$\frac{dT_i}{dx} = \frac{U_i \pi D_i}{\dot{m} C_p} (T_o - T_i) \quad (9)$$

Where the subscripts i, o, and w₁ refer to inside, outside and wall respectively. The boundary conditions are:

$$(i) \quad T_i(0) = T_b \quad (10)$$

$$(ii) \quad T_o(L) = T_i(L) \quad (11)$$

The heat transfer coefficients are evaluated by the Prandtl analogy. The mass flow rate is determined by equating the head resulting from the density differences between the outside and the inside of the casing to the losses due to friction. This results in an algebraic equation which is solved iteratively for the mass flow rate.

$$\dot{m} - \frac{\rho^2 g \beta}{2C_1 L \left(\frac{f_i}{A_i^2 De_i} + \frac{f_o}{A_o^2 De_o} \right)} [B_1 (1 - e^{\lambda_1 L}) + B_2 (1 - e^{\lambda_2 L}) + A_w L] = 0 \quad (12)$$

where

$$C_1 = \frac{U_i \pi D_i}{C_p}$$

$$C_2 = \frac{U_w \pi D_w}{C_p}$$

$$\lambda_1 = \frac{C_2 + (C_2^2 + 4C_1 C_2)^{1/2}}{2\dot{m}}$$

$$\lambda_2 = \frac{C_2 - (C_2^2 + 4C_1C_2)^{1/2}}{2\dot{m}}$$

and

$$B_1 = \frac{-A_w[1 + \frac{\dot{m}}{C_1}\lambda_2 e^{\lambda_2 L}]}{\lambda_2 e^{\lambda_2 L} - \lambda_1 e^{\lambda_1 L}}$$

$$B_2 = \frac{A_w[1 + \frac{\dot{m}}{C_1}\lambda_1 e^{\lambda_1 L}]}{\lambda_2 e^{\lambda_2 L} - \lambda_1 e^{\lambda_1 L}}$$

The temperature profiles can then be calculated from:

$$T_1(x) = B_1 e^{\lambda_1 x} + B_2 e^{\lambda_2 x} + \frac{A_w}{C_1} \dot{m} + T_b - A_w x \quad (13)$$

$$T_o(x) = T_1(x) + \dot{m}/C_1 (B_1 \lambda_1 e^{\lambda_1 x} + B_2 \lambda_2 e^{\lambda_2 x} - A_w) \quad (14)$$

The wall temperature profile for this analysis is taken to be approximately that of the water temperature profile of the uncased well (assumed linear). The validity of this assumption depends on the location of aquifers, variations in thermal conductivity, and time. It is felt that the approximation accurately indicates trends and gives reasonable estimates of temperatures and flow rates. The thermal conductivity of the surrounding rock is considered by introducing an annular ring outside the well through which the energy must diffuse, and calculating an overall heat transfer coefficient based on this and the film coefficient.

The model ignores any mixing of the hot upgoing center fluid and the colder fluid moving downward in the annulus outside the casing that may take place at the bottom of the well. A model to examine mixing effects is discussed later.

The model results for vertical mass flowrate through the well of varying: (i) D/e ($1/e$ the relative roughness) for the annulus; (ii) inside the diameter; (iii) wall temperature gradient; and (iv) conduction distance, the thickness of the annual ring outside the well at which point the wall temperature gradient is assumed to exist (and through which the energy entering or leaving the well at the walls must be conducted) are illustrated in Figure 35. The results are presented for parameter variations about two standard values. The standard parameter sets, I and II are tabulated in Table 4 and denoted by solid and broken lines respectively in Figure 35. These curves show that the flow rate is quite dependent on the inside diameter and assumed values of D/e , conduction of distance, and the wall temperature gradient. The conduction distance has the greatest influence of these assumed values.

The conduction distance effectively shows the effect of time. As the conduction distance increases, conditions approach steady state. The one-dimensional conduction model is no longer valid for a conduction distance of more than a few meters, since the vertical temperature gradient is then of the same magnitude as the horizontal temperature gradient.

TABLE 4. Test well characteristics and standard parameters sets.

GENERAL CHARACTERISTICS

$$D_w = 0.254 \text{ m}$$

$$D_n = 0.0508 \text{ m}$$

Static Water Level - 14 m

Perforation Levels: $\begin{bmatrix} 17 \text{ to } 22 \text{ m and} \\ 57 \text{ to } 65 \text{ m} \end{bmatrix}$

Temperature at bottom of well = 100°C

STANDARD PARAMETER SET
WITHOUT DHE

$$D_i =$$

$$D/e \text{ for outer annulus} =$$

$$D/e \text{ inside casing} =$$

$$\text{Conduction Distance} =$$

$$\text{Wall Temperature Gradient} =$$

$$L \text{ for cased well} =$$

<u>I</u>	<u>II</u>
0.203 m	0.203 m
50	50
3000	3000
0	0.1 m
0.525°/m	0.525°C/m
40 m	40 m

STANDARD PARAMETER SET
WITH DHE

As above except for:

$$\text{Conduction Distance} = 1 \text{ m}$$

$$\text{Inlet Temperature to DHE} = 50^\circ\text{C}$$

FIGURE 35. Model results, no DHE. Mass flow rate for various parameter variations about standard set I, —; and standard set II, ---; (see Table 2.1) — — — Experimental value for actual well with physical characteristics of well specified in Table 2.1.

The two standard sets of parameters give results on both sides of the observed mass flow rate which supports the validity of the model. The vertical temperature profile on the inside of the casing is nearly constant as experimentally observed (Figure 36).

B. Thermosyphoning with DHE

Early wells at Klamath Falls were not cased except for a short length near the ground surface. Later it was discovered that casing the well appreciably extends its life. Some wells that were not originally cased have been cleaned out and a casing installed. This resulted in significantly increased heat exchanger output, but it was not clear how much of this was due to the casing and how much was due to the cleaning (Culver, et al., [4]). The preliminary measurements shown in Figure 34 indicate that the output of a well can be nearly doubled by the installation of a slotted casing. Thus the slotted casing plays a major role in efforts to increase the effectiveness of downhole heat exchangers for these applications.

The model for the cased well was extended to include a downhole heat exchanger. This results in four differential equations rather than two equations as in (8) and (9) above.

$$\frac{dT_o}{dx} = \pm \left[\frac{U_w \pi D_w}{C_p} (T_o - T_w) + \frac{U_i \pi D_i}{C_p} (T_o - T_i) \right] / \dot{m} \quad (15)$$

Figure 36. Temperature profile comparison, model and experimental results.

$$\frac{dT_i}{dx} = \pm \left[\frac{U_i \pi D_i}{C_p} (T_o - T_i) - \frac{U_h \pi D_h}{C_p} (T_1 + T_2 - 2T_i) \right] / \dot{m} \quad (16)$$

$$\frac{dT_1}{dx} = \frac{U_h \pi D_h}{C_p \dot{m}_h} (T_1 - T_i) \quad (17)$$

$$\frac{dT_2}{dx} = \frac{U_h \pi D_h}{C_p \dot{m}_h} (T_i - T_2) \quad (18)$$

where subscripts are as above and subscripts 1 and 2 refer to the entering and leaving legs of the heat exchanger, and subscript h refers to the heat exchanger.

The boundary conditions are

- (i) $T_o(0) = T_b$ or $T_i(0) = T_b$
- (ii) $T_o(L) = T_i(L)$
- (iii) $T_1(0) = T_2(0)$
- (iv) $T_1(L) = T_{in}$

These correspond to: (i) the temperature of the reservoir (assumed known); (ii) temperatures of the water inside and outside the casing are equal at the top of the well; (iii) the temperatures in the two legs of the heat exchanger are equal at the bottom where they join; and (iv) the temperature of the water entering the downhole heat exchanger (assumed known). These equations were solved numerically, iterating on the mass flow rate.

The mass flow rate is again determined by setting the head due to the density differences between the outside and the inside of the casing equal to the frictional loss, which results in an integral equation. The heat transfer coefficients are determined by the Prandtl analogy.

An interesting result of the model is shown in Figure 37. This figure shows the results of varying the length between perforations in the casing. Note that the maximum output occurs for a length of approximately 45 meters when other parameters are as in the standard parameter set (Table 4). The output is small for short lengths, as expected because of the small heat transfer area. As the length increases, area increases, which increases output. When the length becomes sufficiently great, however, the friction effects on the fluid column become more important and the vertical mass flow rate through the casing decreases, decreasing the thermal output.

In a real well, this effect would be less pronounced, since a significant part of the DHE is in contact with water outside the region between slots, and this would tend to flatten the curve somewhat. The actual location of the maximum depends strongly on the assumed value of D/e for the outside annulus and the flow area of the outside annulus.

In Figure 38, the effect of the conduction distance is shown. As previously discussed, the conduction distance essentially shows the effect of time as the region surrounding the well comes into thermal equilibrium with the well water.

Figure 37. Effect of length on model results.

Figure 38. Effect of conduction distance on model results.

It can be seen from the diagram that the effect of the conduction distance is negligible beyond 0.1 meters. After an initial startup period, the effect of conduction through the wall is negligible.

The effect of the assumed value of D/e for the outer annulus is shown in Figure 39. The smoother the surface, the greater the mass flow rate through the casing and the greater the thermal output. This parameter is important because it has a large effect on model results and is difficult to estimate accurately, since it depends on the "roughness" of the sediments and rock that compose the well walls. The "standard" value of 50 for D/e is estimated from values for rough concrete. This value is expected to change somewhat with time as material sloughs off the well walls. Changes in this parameter do not appreciably influence the trends predicted by the model, however.

The casing diameter has considerable influence on DHE output and mass flow rate through the casing, as indicated in Figure 40. For maximum output it is important to provide sufficient cross-sectional area for the flow outside the casing. For the standard set of parameters, the output is relatively independent of the casing diameter when it is in the range of .14 to .18 meters, but it falls off quickly for a casing diameter greater than .19 meters. If caving of the well wall is not a serious problem, it is clearly advantageous to use a casing somewhat smaller than is usual.

Figure 39. Effect of roughness of the outer annulus

Figure 40. Effect of casing diameter

Figure 41 shows the effect of changing the mass flow rate through the heat exchanger. It can be seen that increasing the mass flow rate increases the output, but the output temperature decreases, as expected. Values of 1 or 2 kg/s through the DHE are typical for thermosyphoning DHEs (no forcing pump).

The inlet temperature naturally has an effect on the outlet temperature and mass flow rate through the casing as shown in Figure 42. The relationship is essentially linear in the region shown.

When results of the model are compared to the experimental test results shown in Figure 43, it can be seen that the model tends to give results that are somewhat low. Some discrepancy is to be expected because of the neglect of the portion of the DHE not between perforations. This could account for much of the difference. Overall, the agreement is good considering the number of estimated parameters.

C. Network Models

Experience gained with the differential equation model above shows that many nonlinearities can be neglected and this leads to the simple models of geothermal well systems discussed below.

These models are based on modeling the DHE-well-aquifer system as a network. There are two types of flow in the network: fluid flow (which also transports thermal energy), and heat flow (conduction)

Figure 41. Effect of mass flow rate through heat exchanger. Inlet temperature 50°C .

Figure 42. Effect of inlet temperature.

FIGURE 43. Comparison of model and test results.

alone. Nodes are connected by paths through which fluid flows. These paths may be connected by heat flow paths. See Figure 44 for notation.

The governing equations for these networks are derived by applying conservation of mass, momentum, and energy to nodes and paths. The definition of the overall heat transfer coefficient is also used for conductive paths. There are three possible equations of motion, depending on the type of flow path being modeled. The three types are: (i) a convective loop; (ii) pressure-forced conduit flow; and (iii) pressure-forced porous media flow. Only the convective loop equation is used in the models presented here; the others are included for completeness. The general form of the governing equations is shown in Figure 45.

The models discussed below all include the following assumptions: (i) conduction is negligible except through the casing and into the DHE; (ii) the source temperature is known; (iii) the source mass flow rate is unrestricted; (iv) fluid properties are constant except for density changes in the convective motion equation; (v) temperature differences are assumed small; and (vi) constant heat transfer coefficients. The last assumption restricts the model validity to small changes in mass flow rate through the casing.

The network model corresponding to the differential equation model for the cased well with DHE previously discussed is shown in Figure 46.

HEAT FLOW PATH
(Conduction)

FLUID FLOW PATH
(with flow resistance)

FIGURE 44. Graphical notation for network models.

FIGURE 45. Network model with DHE.

Conservation of Mass:

$$\dot{m}_{ij} = S_i$$

where $\dot{m}_{ij}$ = mass flow rate from node i to node j.

S_i = source strength at node i

Motion:

$$(w_1 + w_2) \dot{m}^2 = \Delta T_{avg} \quad \text{Convection}$$

$$\dot{m}_{ij}^2 = C_{ij} (P_i - P_j) \quad \text{Conduit Flow}$$

$$\dot{m}_{ij} = C_{ij} (P_i - P_j) \quad \text{Porous Media Flow}$$

where

$$w_1 = \frac{2f_{fi}}{\rho^2 g A_i^2 De_i \beta}$$

$$w_2 = \frac{2f_{fo}}{\rho^2 g A_o^2 De_o \beta}$$

C_{ij} = constants of proportionality

ΔT_{avg} = average temperature difference between sides of convection loop

Energy:

$$\sum_j \dot{m}_{ij} T_j = \sum_j \frac{HR_{ij}}{C_p} + S_i T_i$$

Figure 46. General Form of Governing Equations.

The thermal output of the DHE is Q , T_{out} is the outlet temperature and HR_{21} is the heat transfer through the casing. w_1 and w_2 relate the mass flow rate to the average temperature difference. Other symbols are as previously used. T_2 is assumed to be at T_b , the source temperature. The governing equations are:

$$\text{Convective motion: } 2(w_1 + w_2)\dot{m}^2 = T_2 - T_3 \quad (19)$$

$$\text{Energy: } T_2 - T_3 = Q/C_p \dot{m} \quad (20)$$

$$T_1 - T_3 = Q/C_p \dot{m} + HR_{21}/C_p \dot{m} \quad (21)$$

where

$$HR_{21} = \frac{U_i A_c}{2} (T_2 - T_3) \quad (22)$$

$$Q = \dot{m}_h C_p (T_{out} - T_{in}) \quad (23)$$

$$w_1 = \frac{2f_{fi}}{\rho^2 g A_i^2 De_i \beta} \quad (24)$$

$$w_2 = \frac{2f_{fo}}{\rho^2 g A_o^2 De_o \beta} \quad (25)$$

$$Q = \frac{U_h A}{2} (T_1 + T_3 - T_{in} - T_{out}) \quad (26)$$

Combining these equations to give a single equation for $\dot{m}$ gives:

$$\left[\frac{2C_p}{UA} + \frac{1}{\dot{m}_h} \right] \dot{m}^3 + \dot{m}^2 + \frac{U_i A_c}{2C_p} \dot{m} - \frac{T_b - T_{in}}{w_1 + w_2} = 0 \quad (27)$$

After $\dot{m}$ is obtained from this equation, the other unknowns are determined by:

$$Q = 2C_p(w_1 + w_2)\dot{m}^2$$

$$T_{out} = T_{in} + Q/\dot{m}C_p$$

$$T_3 = T_2 - Q/C_p\dot{m}$$

$$T_1 = T_2 - \frac{U_i A_c}{2\dot{m}^2 C_p}$$

These equations allow a relatively simple investigation of the effects of varying parameters. Shown in Figure 47 are the results of this model for various combinations of length and casing diameter. The conclusions reached with the differential equation model are confirmed by the network model. Although the quantities differ somewhat, the effect of changes in length and casing diameter is the same. The figure shows that proper sizing of the casing is even more important as the length becomes greater. Not shown on the diagram is that maximum output occurs with D_i approximately .16 m.

Until this point, the effect of mixing at the bottom of the well has been ignored. Perforations at the bottom of the well allow part of the fluid from inside the casing to mix with fluid outside the casing and be recirculated up the outside of the casing. No provision is

Figure 47. Effect of length and casing diameter on output.

$\dot{m} = 2.5 \text{ kg/s}$, $T_{in} = 50^\circ\text{C}$, $D_h = .0508\text{m}$

made for this occurrence in the previously discussed models. The following model provides a means of estimating the effect of this mixing on the mass flow rate through the casing and DHE output.

The mixing model considered is shown in Figure 48. In this model the temperature at node 2 (T_2) is no longer fixed at $T_2 = T_b$ but is determined by the mixing of flow from the source and recirculated fluid. The flow rate of the source is S and the fraction of flow through the casing that is recirculated is r . Other parameters are as previously indicated.

The governing equations then consist of previous equations (19)-(26) and the following:

$$\text{Mass conservation: } S + k\dot{m} = \dot{m}$$

$$\text{Energy: } T_2 = rT_3 + (1-r)T_b$$

If this system of equations is reduced to an equation for the single unknown $\dot{m}$, the following equation results:

$$\left[\frac{2C_p}{UA} + \frac{1}{\dot{m}_h} \right] \dot{m}^3 + \frac{1+r}{1-r} \dot{m}^2 + \frac{U_c A_c}{2C_p} \dot{m} - \frac{T_s - T_{in}}{w_1 + w_2} = 0 \quad (28)$$

where the terms are as previously defined. When equation (28) is solved for $\dot{m}$, the other unknowns can be determined from the following:

$$Q = 2C_p(w_1 + w_2)\dot{m}^3$$

$$T_{out} = T_{in} + Q/\dot{m}_h C_p$$

$$T_2 = T_s - (Q/\dot{m} C_p) \left(\frac{r}{1-r} \right)$$

FIGURE 48. Network Model with Recirculation.

FIGURE 49. Results of recirculation model.

$$T_3 = T_2 - \frac{Q}{\dot{m}} C_p$$

$$T_1 = T_2 - \frac{U_1 A_c}{2 \dot{m} C_p} (T_2 - T_3)$$

Some results of this calculation are shown in Figure 49. The graph shows that moderate recirculation and energy extraction rates do not have a large effect on energy output. When greater amounts of energy are withdrawn, the recirculated fluid becomes increasingly colder and its effect becomes somewhat greater.

D. Methods of Increasing DHE Output

Downhole heat exchanger output can be increased by three basic methods: (i) increased film coefficients; (ii) increased heat transfer area; and (iii) increased log mean temperature difference (LMTD).

The film coefficient can be increased by increasing the flow velocity and installing turbulence promoting devices. The flow velocity can be increased by the following methods: (i) proper sizing of the casing and wellbore to provide a low resistance path for the flow of liquid up the outside of the casing; (ii) slots of adequate size; and (iii) pumping fluid past the DHE. Turbulence promoting devices have the unfortunate effect of increasing friction which tends to decrease the velocity since the flow is due to thermosyphoning only and is therefore sensitive to flow resistance.

The importance of making the outer annulus sufficiently large when the DHE consists of a single loop is very evident from Figures 40 and 47, which show that the DHE output is very sensitive to the casing diameter. When multiple loops are present, the output becomes less sensitive to casing diameter and the optimum diameter is larger than when only a single loop is present. The optimum diameter for the casing becomes smaller as the distance between perforations increases.

Pumping fluid past the DHE appears to add considerable complication and expense, and was therefore not considered in this work.

Area can be increased by: (i) adding more tubing to the heat exchanger; (ii) using multiple small diameter tubes; (iii) adding fins; and (iv) providing a high conductivity path from the heat exchanger to the casing to use the casing as a fin. Fins are attractive if there is a significant difference in the film coefficients between the inside and outside of the heat exchanger tubes. Fins are added to the outside of the heat exchanger tubes, which is the side with the lower heat transfer coefficient. If the heat transfer coefficients are approximately equal, the gain from fins is slight and additional area of plain tubing is generally the most economical method of increasing the area. Multiple small tubes have a greater surface area than the same capacity using a large tube, but would be somewhat more expensive and difficult to install.

An interesting but rather impractical method of increasing the surface area is to have a good thermal connection between the casing and the DHE so that the casing functions as a large fin for the DHE. This could be very effective but would make installation and replacement of the DHE very difficult.

Figure 50 shows the effect of adding fins to the DHE. It is evident from Figure 50a that with a mass flow rate through the DHE of 2 kg/s, approximately the rate measured for thermosyphoning DHEs (Culver, et al., [4]), there is only a small increase in output possible, as the fluid in the DHE is already heated nearly to the maximum possible temperature.

With a higher flow rate through the well, Figure 50b shows a significant improvement for a finned DHE. An increase of about 40 percent is indicated by the network model. The model shows 24 fins about .0127 m by .0064 m to be optimal for the well dimensions indicated.

The use of multiple small diameter tubes can result in significantly higher output than a finned tube, as shown in Figure 51. For a mass flow rate of 4 kg/s, the network model with 12 loops of .0254 m pipe results in an increase in output of approximately 100 percent. Increasing the number of loops beyond 12 resulted in decreased output. While fins can significantly increase output, multiple loops are necessary for really large increases. These results are for flow in parallel

FIGURE 50a. Effect of Fins on model results.
 $\dot{m}_h = 2\text{kg/s}$ $T_{in} = 50^\circ\text{C}$ (Network model)

FIGURE 50b. Effect of fins on model results.
 $\dot{m}_h = 4\text{kg/s}$ $T_{in} = 50^\circ\text{C}$ $L=40\text{m}$
 (network model)

Figure 51. Effect of multiple tubes on model results. (Network model)

$L = 40\text{m}$

$D_c = .203\text{m}$

loops and with friction factors and heat transfer coefficients assumed constant. The effect of allowing the friction factors and heat transfer coefficients to vary results in a decrease in the optimum number of tubes and fins.

The log mean temperature difference (LMTD) depends on the direction of flow in the well. If flow is down inside the casing, the hottest point will be at the top. If there is free thermosyphoning with little mixing at the well bottom, the ideal arrangement is to have the fluid taken quickly to the well bottom. Then, after it has been warmed, give it a final heating at the top before leaving the well. This arrangement is not particularly attractive, however, since caving may occur. If appreciable caving does occur, thereby blocking the outer annulus, this method will be considerably inferior to having additional area at the well bottom where it would be more accessible to the hot fluid flowing in the aquifer.

VII. Future Work

Future work is basically a continuation of the work to date in the four major areas of:

1. Flow measurement
2. DHE testing
3. Heat transfer modeling
4. Economic analysis

Specific tasks are detailed below.

Flow Measurement

Horizontal flows in the aquifer should be measured in order to help determine the amount of mixing possible between the geothermal reservoir and the water in the wellbore. According to the heat transfer model as presently developed, the amount of mixing has a relatively large effect on output when energy extraction rates are high.

Vertical flows should be measured before the casing is installed, after the casing is installed, and with an operating DHE installed in order to determine if there is a correlation between flows in these stages of well construction and to corroborate the heat transfer model.

Correlate hot-film anemometer and spinner flow measurements. The hot-film anemometer because of its complexity and apparent short life does not appear to be the type of instrument that well drillers and heating contractors could routinely use. On the other hand, spinners may be,

but apparently give misleading results. Hence the anemometer and spinner readings should be compared for additional wells and a correlation between the two worked out.

Thermistor flow meter (or other relatively simple and rugged instruments) should be developed as an instrument to provide more accurate data than a spinner and yet be nearly as simple and rugged. Work will continue on this development.

DHE Tests

Conventional DHE tests should be run on at least two additional wells if possible in order to provide information to refine the heat transfer model and check model predictions based on measured well parameters.

A Multi-Tube DHE should be designed, installed, and tested since the heat transfer model predicts that this will be one of the best DHE configurations. Ideally, this should be done in a well in which a conventional DHE has been tested in order that a direct comparison can be made. Flow measurements should be made while testing the multi-tube design.

Heat Transfer Modeling

Using information derived from the work outlined above and recent hydrological information developed, the present model will be refined and further substantiated. Two aspects of this will be to include mixing and evaluate energy extraction limits.

Economic Analysis

After obtaining accurate information on feasible energy extraction rates for DHEs and economic data, economic comparisons will be made among DHE, surface heat exchange geothermal, and conventional systems.

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APPENDIX

LITERATURE REVIEW OF FLOW MEASURING METHODS

The literature review resulted in a ranking of potential flow measuring methods for this application as follows:

1. Hot-film anemometer.
2. Resistance heater enclosed in a tube.
3. Jet velocity meter.
4. Acoustic flowmeters.
5. Thermistors.
6. Vane anemometers
7. Vortex-shedding flowmeters.

Each of these and their applicability are discussed below:

Hot-Film Anemometers

First developed by S. C. Ling [1] at Iowa State University in 1955, hot-film anemometers are made by depositing a thin conducting film about 100 nm thick on a ceramic substrate. The film is electrically insulated from the fluid by a ceramic coating (usually quartz). Therefore hot-films are not subjected to electrolysis problems. Also, since the film can be backed by reinforced substrate, hot-films are much more rugged than hot-wires. Because of their larger surface area, hot-films are more sensitive than hot-wires to changes in mean velocity [2].

Calibration drift of hot-films is a function of the ratio of the particle size (of impurities in the water) [3] to the sensor radius. Therefore the larger the sensor, the less the drift. Each of the different shapes that hot-films are manufactured in have their own drift characteristics. For example, conical hot-film sensors experience less drift than wedge films [4]. (Two other principle shapes are cylindrical and side-patch.) Kline and Morrow [3] found clean, hard water (200 ppm CaCO_3) had no effect on nickel-wedge, platinum cylindrical, or platinum cone-type hot-films. The tested hot-films also had no drift in clean, deionized water. For particles of size less than $0.8 \mu\text{m}$, wedge and cylindrical hot-films had very little drift (less than 0.2 percent voltage change per hour).

Hot-film sensors have been calibrated and used in water over velocity ranges from zero to several meters per second. According to some researchers [5], these sensors can be used down to the velocity at which the forced convection component of heat transfer from the film is no longer much greater than the natural convection component, while others have gone much lower. Dring and Gebhart [6], for example, calibrated hot-wires from 0 to 0.508 cm/s, and found that neither natural nor forced convection was dominant. (Dominance of one mode of heat transfer or the other may be dependent on the type of probe used.)

Resistance Heater

Skibitzke [7] has developed a flowmeter using thermistors for vertical flow measurements in wells. Consisting of a tube 1.5 cm in diameter and 30 cm

long, it uses two thermistors, one on each side of a resistance heating element mounted inside the tube. Centered in the well, water flows up the tube and is heated by the resistance heater. (A flow-channeling device is used at low well velocities to force more water to flow through the tube.) The faster the flow, the less the water is heated as it passes the heating element. The degree of water heating is measured by the upstream and downstream thermistors. Good results down to 5 cm/s without and 1 cm/s with a flow-channeling device were obtained. The flow-channeling device would be necessary for measurements in Klamath Falls wells in order to obtain sufficient sensitivity. However, flow disturbance or blockage could result. Another serious disadvantage is that fluid velocity inside the tube is much less than the velocity outside the tube, so the device must be calibrated separately for each well diameter [8]. Calibration must also take into account the large variations in fluid temperature which lead to fluid property variations in the wells. This instrument cannot measure crossflow without reorientation unless a multiple-device array is built.

Jet Velocity Meter

Tanney [9] describes a velocity sensor which ejects a free jet normal to the main flow and has two receiving total head tubes opposite the free jet nozzle and oriented toward the jet (see figure on next page). The nozzle-to-receiver spacing is usually greater than 15 initial jet diameters, and the jet velocity must be at least one order of magnitude greater than the crossflow velocity of the measured fluid. Fluid movement causes

Tanney's jet velocity sensor from [9].

spreading of the jet and decreasing jet velocity as distance from the nozzle increases. The amount of this spreading depends on the fluid velocity and is measured by the differential pressure between the two receiving heads. This differential pressure is relatively large and hence easily measured. It ranges from 0 to 12.4 kPa over the working range of the instrument, depending on the pressure of the supply jet, and is proportional to the crossflow velocity between 1.52 and 30.5 cm/s. If higher velocities must be measured, a ruggedized version of the jet velocity sensor is used. Instrument calibration has little sensitivity to fluid temperature and density. Calibration is, however, affected by the pressure of the supply jet. Direction (up or down) of vertical flow in the wellbore could be determined with this sensor. A jet velocity sensor for two-dimensional flow has also been developed [9]. In all, five tubes are required for this, versus three tubes for the

one-dimensional flow sensor. The instrument can be made resistant to contamination and corrosion. For use in a well, the supply pressure must be precisely controlled to give constant jet flow rate, or the calibration will change. As with pitot tubes, there is a potential problem with air bubbles in the long lines coming from the receiving tubes.

Acoustic Flowmeters

Acoustic flowmeters use the difference in upstream and downstream travel times of sound waves to determine fluid velocity (see figure on next page). This is often accomplished by comparing the phase of the two signals; phase difference is proportional to flow rate. Lester [10] reports the design and successful testing of a unit able to measure velocities from about 0.15 cm/s to 15 cm/s with 1 percent accuracy. As is true of other acoustic flowmeters, this instrument is (1) nonfouling due to a lack of moving parts, (2) linear, and (3) capable of measuring velocity components. It can also be zeroed and calibrated in the field. The instrument is often used in pipe flow studies and underwater in oceanographic applications. One potential problem is vortex shedding from the transducers. This problem is overcome by setting the transducers, and hence the acoustic path, at a slight angle to the flow; then the vortices will not disturb the acoustic path. The transducers should be placed as close together as possible (less than 30 cm) so that problems from reflected and refracted acoustic signals are minimized. If the transducers can be

Acoustic flowmeter system from [10].

made smaller* and also withstand the 90° to 100°C temperatures of Klamath Falls wells, then the acoustic flowmeter would be a promising device for our application.

Thermistors

Thermistors are semiconductors with large negative temperature coefficients of resistance, minus 3 to 5%/°C as compared with 0.4%/°C for platinum hot-films (to be discussed later) [11]. The temperature coefficient of resistance is positive for most metals, further contrasting the thermistor's behavior. The high temperature coefficient produces larger signal changes for given velocity changes than a hot-film would. Also, because their sensor resistance is two or more orders of magnitude larger than that of a platinum film, long cable lengths which are necessary to lower the instrument into the well do not add significantly to the resistance of the circuit. For the same reason, thermistor data can also be corrected more accurately for fluid temperature variation. Because of their larger mass, thermistors usually have a lower frequency response (less than a few hundred Hertz) than hot-films, though it is felt that this would not be a problem in our application since high-frequency turbulence is not of interest.

Thermistors can be operated in either the constant-power or constant-current mode. Keller and Lane [12] found that thermistors can be used

*Lester [10] stated that one of the physical measurements of the transducers was 6 cm. No other size information was given.

to measure velocities of 0.25 to 15 cm/s in water, with the greatest sensitivity attained in constant-current operation. Lumley [13] measured turbulence using a thermistor in the constant-temperature mode. He found that the signal-to-noise ratio of thermistors in water was one order of magnitude better than that for platinum films, and states further that they require simpler electronics because of their larger resistance.

A thermistor bead used just by itself has no directional sensitivity; it has to be mounted on some type of special probe body to acquire such ability. Thermistors are subject to some of the same problems as other thermal methods.

Deflecting-Vane Anemometers

Deflecting-vane anemometers measure the velocity pressure, $v^2/2$, caused by the flow. The deflecting vane anemometer measures the velocity pressure by letting the fluid impinge on a vane which is mounted on jeweled bearings but is prevented from rotating freely by a delicate spring. Used in air flows, these vanes have been reported to have working ranges of 7.62 to 9144 cm/s [14]. Extrapolated to water, this range is 0.27 to 320 cm/s, clearly encompassing the limits of our work.* The larger the vane, the greater is its sensitivity.

*The velocity limits for air were extrapolated to those for water by noting that for a given velocity pressure, the water velocity is related to the corresponding air velocity by

$$v_{\text{water}} = \sqrt{\frac{\rho_{\text{air}}}{\rho_{\text{water}}}} v_{\text{air}}$$

Head and Surrey [15] described a vane anemometer consisting of a 0 to 0.5 mA moving-coil ammeter; the movement is connected to an arm extending from a 10.16 cm square vane, giving a total vane area of 103 cm^2 . A current is fed to the ammeter coil that produces the necessary torque to balance torque created by the velocity pressure of the fluid on the vane. They report a measurement range of 6.10 to 122 cm/s in air, which corresponds to 0.22 to 4.3 cm/s in water. The upper velocity limit is probably too low for use in our wells, but this can be adjusted by varying the area of the vane or the ammeter sensitivity. For well use, the transducer would have to be developed to enable an observer at the surface to tell when the ammeter arm was balanced. A feedback-transducer system could also be used.

Two vanes would be required for directional capability, each connected to a transducer that converts angular position to an electrical signal.

It appears that a deflecting-vane anemometer small enough to measure well water velocities could be developed. Special attention would have to be paid to the anemometer deflection transducer or feedback-transducer system so that well pressures and high temperatures would not be damaging.

Vortex-Shedding Flowmeters

Vortex-shedding flowmeters utilize the principle that the frequency of vortex shedding behind a bluff body depends on the free-stream velocity. If the flow is perpendicular to the axis of a circular cylinder, the

frequency at which vortices are shed is measured in terms of two parameters, the Strouhal number $S = nD/V$, and the Reynolds number $Re = DV/\nu$, where n is the frequency at which vortices are shed, D is the cylinder diameter, V is the free-stream velocity, and ν is the kinematic viscosity. Strouhal number is a well-known function of Reynolds number [16] (see figure below).

Strouhal number as a function of Reynolds number from [16].

The presence of vortices, and hence their frequency, can be detected by instruments such as hot-wire or hot-film anemometers. Bradshaw [17]

reports that the lowest Reynolds number for which the vortices are detectable is about $Re = 100$. Notice that above $Re = 600$ on the figure from the previous page, that the Strouhal number is constant, giving a 6:1 usable velocity ratio. Using information in Schlichting [16] the minimum measurable free-stream velocity for several cylinder diameters was obtained. The results are presented below in table form:

Minimum measurable velocity as a function
of circular cylinder diameter, using
vortex-shedding.

Cylinder diameter (cm)	Free-stream velocity (cm/s)
2.0	0.159
1.5	0.212
1.0	0.318
0.5	0.636
0.25	1.270

As cylinder diameter decreases, minimum measurable velocity increases. Cylinder diameter would have to be 0.25 cm before the resulting measurable velocity range, in this case 1.27 to 7.62 cm/s, would match our needs.

Vertical flows in wells could be measured with vortex-shedding flowmeters, but two cylinders and two anemometers would be required--one for upward flow and one for downward flow. Crossflow measurement requires the addition

of another cylinder and anemometer. There is one serious disadvantage, however. Vertical flow can be measured only in the absence of crossflow, and crossflow can be measured only when vertical flow is not present. If there is a component in one direction when the other component is being measured, the flow will not be perpendicular to the cylinder being used and calibration dependent on perpendicular flow will be lost. This problem could be overcome by enclosing the horizontal and vertical flow sensors inside separate small tubes and orienting them in the desired directions. This would, however, require that the instrument be calibrated for each well diameter.

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