DESIGN OF A GEOTHERMAL ENERGY DRYER FOR BEANS AND GRAINS DRYING IN KAMOJANG GEOTHERMAL FIELD, INDONESIA

Untung Sumotarto, Agency for the Assessment and Application of Technology-BPPT, Indonesia

ABSTRACT

Indonesia is a country rich in geothermal energy. Of approximately 20,000 MWe energy potential, only about 850 MW has been utilized for electricity purposes. There are not many direct utilization activities for various purposes that have been implemented. This paper discusses a design of a geothermal dryer for beans and grains drying that will be implemented in Kamojang geothermal field, West Java, Indonesia. Geothermal fluid waste from a Kamojang well of approximately 160°C will be used to supply the equipment. The heat will be extracted to produce a room drying temperature, for which coffee bean will be used as an experimental grain to be dried. A tube-bank heat exchanger has been designed, consisting of 1-meter-staggered pipes of 2-inches (50.1-mm) outer diameter. An air blower from one side produces air flow of varying velocities to flow heated air into a drying room on the other side. With a geothermal fluid flow containing a heat transfer rate of 1000 W and various air flow velocities of 4 to 9 m/s, the HE design could produce an output drying temperature of 45.48 to 40.64°C and drying energy (heat) produced in the drying room of about 41.0 to 68.0 kW/m length of the heat exchanger. Depending on the bean's humidity, the drying time has to be set accordingly.

INTRODUCTION

Indonesia is a country having many volcanoes and rich in geothermal energy. There are at least 177 volcanic centers that are spread over volcanic belts of 7000 km along the Indonesian islands. From that many centers, there is at least 20,000 MWe-equivalent from geothermal energy resources contained in the volcanic areas. Islands in which geothermal energy can be found are Sumatera, Java, Nusa Tenggara, Sulawesi, and Maluku.

By now, only about 850 MW of that much energy potency has been utilized for electricity power generation. The rest has not been utilized optimally. Among the fields that have been developed are Kamojang, Darajat, Gunung Salak, Patuha, Wayang Windu, Dieng, Lahendong, and Sibayak. Electricity has been produced from those major fields. Unfortunately, there is only limited direct utilization of geothermal energy in those fields as well as in other undeveloped ones.

Meanwhile, in the geothermal fields that have been developed and utilized such as Kamojang, Dieng, Darajat, Gunung Salak, etc, there have been production geothermal wells that already have depleted pressure, temperature, and production rate thus unable to supply the existing electric power plants. Such wells have been modified to re-injection or monitoring wells. There are also waste geothermal fluids from electric power plants that are usually re-injected into the reservoir to maintain the life of the geothermal reservoir. The heat contained in the fluids can still be extracted to supply equipment or engines for producing fresh water steam, to supply heating or drying equipment for sterilization of growing media, drying agricultural and husbandry products and other direct utilizations.

The existence of geothermal energy resources that is commonly found in mountainous and inland regions has its own benefit. In Indonesian mountainous and inland regions there are found agricultural, plantation, and forestry areas in which their products require processes such as drying, preservation, heating, sterilization, pasteurization, etc. The agricultural and plantation product processing requiring heat are for examples: rice, coffee, and tea drying, potato seeding, mushroom cultivation, milk pasteurization, etc.

To initiate direct utilization of geothermal energy, the Agency for the Assessment and Applied Technology (BPPT), has been doing research in that field since 1999-2000. The first effort was a research in geothermal energy utilization for sterilization of mushroom growing media in Kamojang geothermal field. The research was a success and is now planned to continue to commercial scale. The research presently continues to design a geothermal dryer for beans and grains. This paper discusses such design to see the technical feasibility if the equipment will be built (Sumotarto et al., 2000)(Sumotarto, 2001).

METHODOLOGY & DESIGN OF EQUIPMENT

Traditionally, grains and beans drying in Indonesia have been done by heating the products under sunshine (solar drying). The products will be influenced by seasonal and weather changes, thus making the drying process un-continuous. This will result in cracking, fracturing, and imperfect drying products.

To improve the process, drying has to be done continuously, requiring continuous heat supply. This can be reached by using continuous energy resource supply such as geothermal fluid flows.

EQUIPMENT DESIGN

The dryer used in this research will be made of a fluid-air heat exchanger to produce hot air that will be blown into a drying room filled with trays of grains or beans. Figures 1 – 4 show the design of the equipment. The waste geothermal fluid is flowed into a bank of steel pipes, and air is blown outside the pipes to extract heat from geothermal fluids inside the tubes for the drying process.

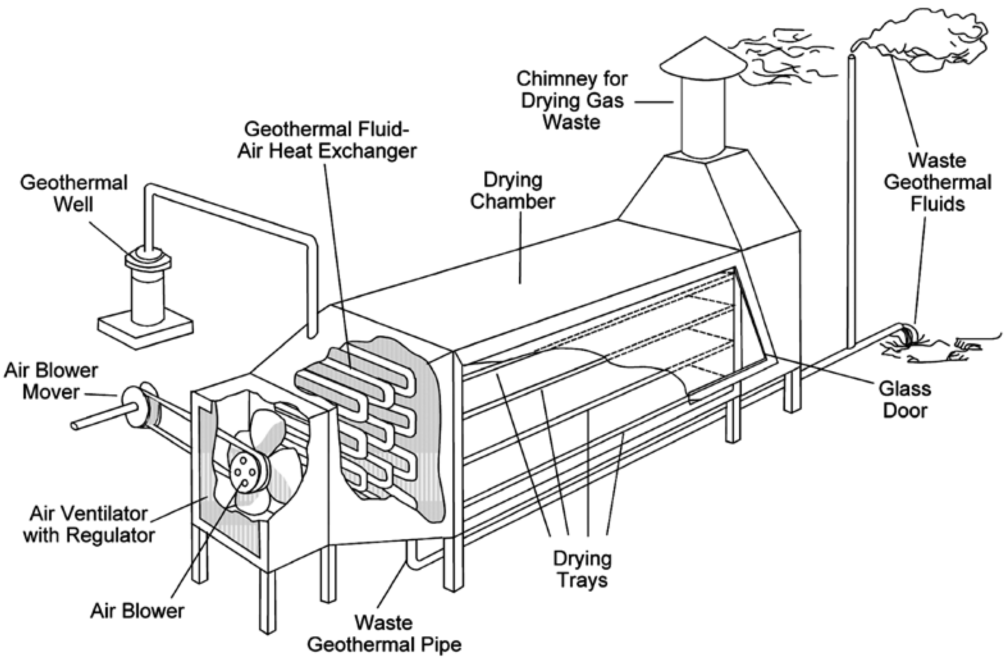

Figure 1. 3D view of the geothermal dryer design for drying grains and beans.

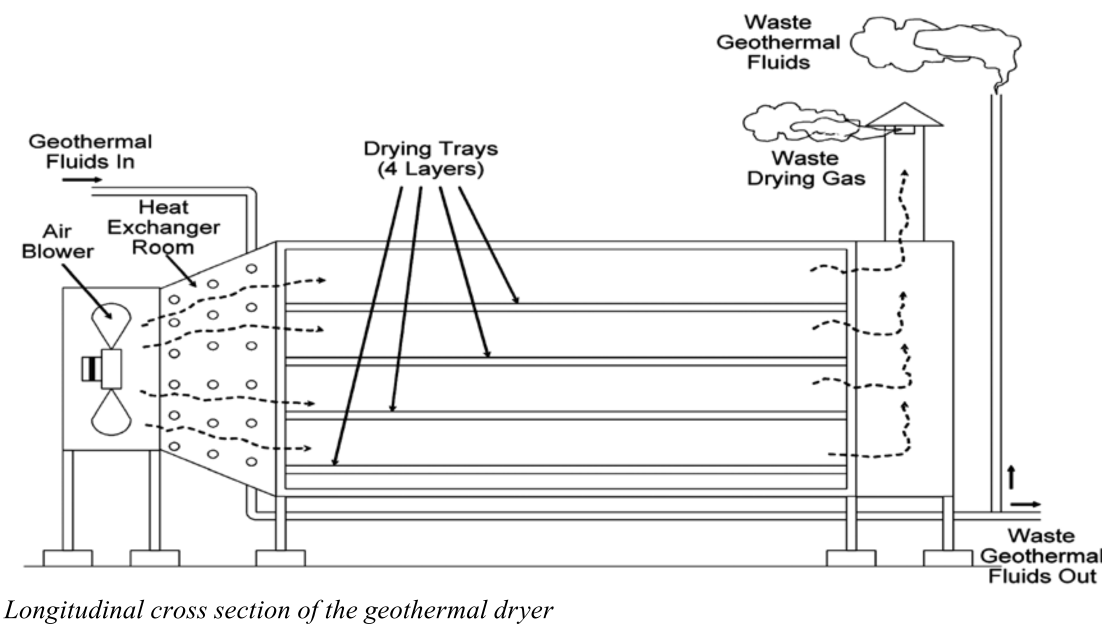

Figure 2. Longitudinal cross section of the geothermal dryer

The equipment does not use a drying belt to save energy for moving the belt. Instead, the beans and grains are placed on trays in the drying room. The only moving part is an air blower that can be designed to move by geothermal energy (pressure), while its heat content is used for the heat exchanger. The air blower is placed on one side of the heat exchanger while the drying room is on the other side.

The drying duration depends on the original humidity of the products. By doing several drying experiments, an ideal drying time can be found for which the product is perfectly dried. The dryer is designed as simple to assist the technical feasibility of geothermal energy direct utilization. If the drying is proven to be feasible, then the technology and design can be improved while the scale can be increased to meet a commercial project.

HEAT & ENERGY EQUATIONS

The energy balance equations governing the heat exchanger can be modeled in two parts. The first part is heat transfer from the geothermal fluids inside the tubes to the outer side of the tubes, and the second part govern the heat transfer process outside the tubes into the drying room. Figure 3 shows the first part of energy (heat) transfer and, figure 4 shows the second part.

Calculation of energy transfer in this part is performed when the equipment is in operation with steady state energy

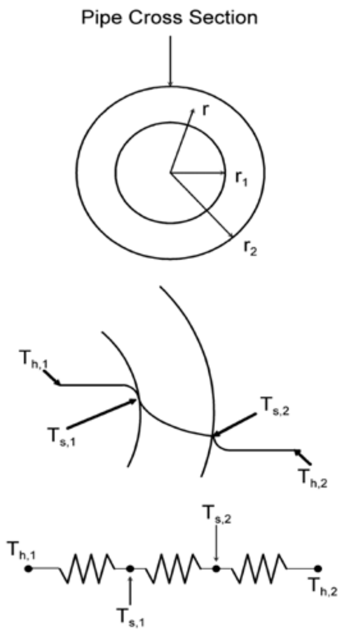

Figure 3. Heat transfer across the heat exchanger pipes.

transfer (heat flow). The calculation is based on phenomena where energy (heat) flows across a cylindrical pipe (Figure 3). Outside the heat exchanger the air is assumed to flow convectional into the drying room. There is assumed no other mode of heat flows i.e. radiation, because of the high speed of convection air current.

PART I:

For simple calculation, this part can be modeled as one-dimensional radial heat flow. If there is no energy generation in the equipment, heat transfer equation governing the system is

$$\frac{1}{r}\frac{d}{dr}\left(kr\frac{dT}{dr}\right)=0 \qquad (1)$$

According to Fourier's Law, energy (heat) flow rate by conduction through solid cylindrical surface can be expressed as

$$q_r=-kA\frac{dT}{dr}=-k(2\pi rL)\frac{dT}{dr} \qquad (2)$$

where A=2πrL= area of the surface normal to the direction of heat transfer. The heat transfer rate q_r , not heat transfer flux q_r", is a constant value in radial direction.

Equation (1) can be integrated twice to find a general solution

$$T(r)=C_1\ln r+C_2 \qquad (3)$$

With boundary conditions: T(r1) = Ts,1 and T(r2) = Ts,2 (Figure 4), C1 and C2 can be found as

$$C_1=\frac{T_{s,1}-T_{s,2}}{\ln(r_1/r_2)} \quad \text{and} \quad C_2=T_{s,2}-\left(\frac{T_{s,1}-T_{s,2}}{\ln(r_1/r_2)}\right)\ln r_2$$

Substituting C1 and C2 back to Equation (1) results in a general equation of temperature for the system as follows

$$T(r)=\frac{T_{s,1}-T_{s,2}}{\ln(r_1/r_2)}\ln\left(\frac{r}{r_2}\right)+T_{s,2} \qquad (4)$$

Note: Temperature distribution for a system associated with radial conduction heat flow through a cylindrical surface is in the form of logarithmic, not linear such as on a flat wall of similar condition.

Further, substitution of Equation (4) into Equation (2) results in a general equation of heat rate as follows

$$q_r=\frac{2\pi Lk(T_{s,1}-T_{s,2})}{\ln(r_2/r_1)} \qquad (5)$$

where

$R_{t(cond)}=\dfrac{\ln(r_2/r_1)}{2\pi Lk}$ is also called *Thermal Resistance.*

The heat flow from the center of the tube to the inside wall of the tube and from the outer side of the tube into the open air is a convection flow. For the whole system from $T_{\infty,1}$ -> $T_{s,1}$ -> $T_{s,2}$ -> $T_{\infty,2}$, the heat flow rate equation (5) can be formulated as

$$q_r=\frac{T_{\infty,1}-T_{\infty,2}}{\dfrac{1}{2\pi r_1Lh_1}+\dfrac{\ln(r_2/r_1)}{2\pi kL}+\dfrac{1}{2\pi r_2Lh_2}} \qquad (6)$$

which can also be expressed in heat rate equations for each portion of the entire flow as

$$q_{r,1}=2\pi r_1Lh_1(T_{\infty,1}-T_{s,1}) \qquad (6.a)$$

$$q_{r,2}=\frac{T_{s,1}-T_{s,2}}{\ln(r_2/r_1)}2\pi Lk \qquad (6.b)$$

and

$$q_{r,3}=2\pi r_2Lh_2(T_{s,2}-T_{\infty,2}) \qquad (6.c)$$

where
q = heat transfer rate [W]
r = pipe radius [m]
L = pipe length [m]
T = temperature [K] or [C]
k = thermal conductivity [W/m.K]
h = convection heat transfer coefficient [W/m^2.K]

Those three equations (6.a, 6.b, and 6.c) can be used to calculate $T_{s,1}$ and $T_{s,2}$ because $q_r = q_{r,1} = q_{r,2} = q_{r,3}$.

PART II:

As heat leaves the outer side of the pipes the governing equations can be modeled as an air convection flow through a bank of tubes (Figure 4). The tube rows in this heat exchanger are staggered in the direction of *fluid velocity (V)*. The configuration is characterized by the *tube diameter (D)* and by the *transverse pitch (S_T)* and *longitudinal pitch (S_L)* measured between tube centers. Flow conditions within the bank are dominated by boundary layer separation effects and by wake interactions, which in turn influence convection heat transfer. Incropera et.al (1985), describes such phenomena with the following equations.

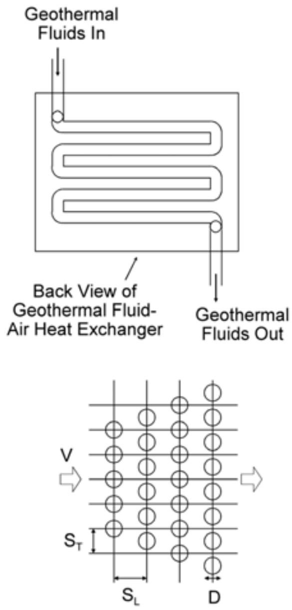

Figure 4. Heat exchanger pipes lay-out.

The amount of heat transfer can be calculated by, first calculating the air-side Nusselt number as:

$$\overline{Nu}_D = C\,\mathrm{Re}_{D,\max}^{m}\,\mathrm{Pr}^{0.36}\left(\frac{\mathrm{Pr}_\infty}{\mathrm{Pr}_s}\right)^{1/4} \qquad (7)$$

where

$$\mathrm{Re}_{D,\max} = \frac{V_{\max} D}{\nu} \qquad (8)$$

ν = kinematics viscosity of the air [m^2/s].
Pr = Prandtl number
C and m are constants that depend on tube configuration (tabulated in table 1 (Howell and Buckius, 1987)).

The *Reynolds number ($Re_{D,max}$)* for the foregoing correlations is based on the *maximum fluid velocity (V_{max})* occurring within the tube bank. For the staggered configuration, the maximum velocity may occur at either the transverse plane or the diagonal plane. If the rows are placed such that

$$S_D = \left[S_L^2 + (S_T/2)^2\right]^{1/2} > (S_T + D)/2 \text{ , then}$$

$$V_{\max} = \frac{S_T}{S_T - D} V \qquad (9.a)$$

Otherwise,

$$V_{\max} = \frac{S_T}{2(S_D - D)} V \qquad (9.b)$$

The average convection *heat transfer coefficient (h)* can be calculated using the following equation

$$\bar{h} = \overline{Nu}_D \frac{k}{D} \qquad (10)$$

where

k = gas (air) thermal conductivity [W/m.K].

The heated air temperature produced from the heat exchanger can be calculated using a log-mean temperature difference

$$\Delta T_{lm} = \frac{(T_s - T_i) - (T_s - T_o)}{\ln\left(\dfrac{T_s - T_i}{T_s - T_o}\right)} \qquad (11)$$

where T_i and T_o are temperatures of the fluid as it enters (T_i) and leaves (T_o) the bank, respectively and T_s is the temperature of the tube outside surface. The outlet temperature T_o, which is needed to determine ΔT_{lm} may be estimated from

$$\frac{T_s - T_o}{T_s - T_i} = \exp\left(-\frac{\pi D N \bar{h}}{\rho V N_T S_T c_p}\right) \qquad (12)$$

where

ρ = air mass density [kg/m^3]
c_p = gas (air) specific heat at constant pressure[J/kg.K]
N = total number of tubes in the bank, and
N_T = number of tubes in the transverse plane

Finally, the heat transfer rate per unit length of the tubes may be computed from

$$q' = N\left(\bar{h}\pi D \Delta T_{lm}\right) \qquad (13)$$

where

q' = heat transfer rate per unit length [kW/m]

SIMULATION AND RESULTS

Using the equations described in the above section, it can be determined the relation between the *air flow velocity (V)* produced by the air blower and the *drying temperature (T_o)*. Depending on the drying temperatures that are specific to each product, the *air flow velocity (V)* can be adjusted accordingly. The calculation and simulation are conducted as follows:

1. Write a computer program for the calculation according to the equations described in previous sections.
2. Prepare parameters and constants needed in the calculation such as geothermal fluid temperature inside the tube, drying temperature required, heat transfer coefficients, etc. Table 1 shows parameter and constants needed for the calculation.

3. Using the proper data and calculation procedure, it can be calculated the parameter needed for the drying process.
4. Tables and figures showing various relations can be made according to the calculation results.

Table 1. Parameters used in the simulation and calculation of the geothermal drying equipment for grains and beans.

Parameters		Unit
Geothermal Fluid heat convection coefficient (h_1)	5000	$W/m^2.K$
Heat Exchanger pipe thermal conductivity (k)	10	W/m.K
Air heat convection coefficient (h_2)	15	$W/m^2.K$
Geothermal Fluid temperature	160	^0C
HE Input Temperature (T_i)	20	^0C
HE transverse pitch (S_T)and longitudinal pitch (S_L)	0.08, 0.11	m
Air kinematics viscosity (ν)	15.2678 $*10^{-6}$	m^2/s
Gas (air) thermal conductivity (k_{air})	0.0263	W/m.K
Air mass density (ρ)	1.1614	kg/m^3
Constants in Nusselt number calculation (C&m)	0.330430629 , 0.6	-
Gas (air) specific heat at constant pressure (c_p)	1.007	J/kg.K
Total number of tubes in the HE bank (N)	26	-
HE number of tubes in the transverse plane (N_T)	6	-
HE pipe diameter and total length	0.0501, 29.3	m

Using the above simulation procedure it is found that at a geothermal fluid temperature of 160˚C, there can be calculated various heat transfer rate and drying temperature at various air flow velocity produced from the air blower. Tables 2, 3 and 4 show the results of the calculations and the relations between air flow velocity to heat transfer rate and drying temperature.

Table 2 shows that at various geothermal fluid flows with *heat transfer rate* (q_r) of between 1000 to 6000W, the outside surface temperature of the heat exchanger pipes would reach as high as 159.93 to 159.60˚C. Further, Table 3 shows that at the above various heat transfer rate, the output *temperature* (T_o) of the HE would reach 45.48 to 45.44˚C, from which we can have enough temperature for drying purposes. It can be seen here that varying geothermal heat transfer rate does not result in significant range of outside temperature of the HE pipes and output temperature of the HE in the drying room. Therefore, it is enough to pick one value of the geothermal heat transfer rate to be used for sensitivity analysis of the other governing parameters i.e. *air flow rate (V)* from the blower.

Further, by picking a fixed heat transfer rate of 1000W, which results in outside surface temperature of the HE pipes of 159.93˚C, it can be seen in Table 4 that for various air flow

Table 2. Relationship between geothermal heat flow rate (q_r[W]) inside the HE tubes and inside and outside pipe surface temperature (T_{s1},T_{s2}[C]) and air temperature ($T_{\infty 2}$[C]) for a constant geothermal fluid temperature ($T_{\infty 1}$[C]) of 160 [C].

q_r	T_{s1}	T_{s2}	$T_{\infty 2}$
1000	159.95	159.93	145.48
2000	159.91	159.87	130.96
3000	159.86	159.80	116.45
4000	159.82	159.73	101.93
5000	159.77	159.66	87.41
6000	159.73	159.60	72.89

velocities from the air blower of between 4 to 9 m/s, the drying temperature would vary between 45.48 to 40.64˚C, which will produce a drying heat transfer rate of 41.0 to 68.0 kW per meter length of the HE.

Table 3. Output (drying) temperature (T_o[C]) and heat transfer rate per length of HE (q_{rate}[kW/m]) at various outside surface temperature of the HE (T_{s2}[C]).

T_{s2}	T_o	q_{rate}
159.93	45.48	41.50
159.87	45.47	41.48
159.8	45.46	41.46
159.73	45.45	41.44
159.66	45.44	41.42
159.6	45.44	41.40

Table 4. Output (drying) temperature (T_o[C]) and heat transfer rate per length of the HE (q_{rate}[kW/m]) at various air flow rate (V[m/s]) from the air blower for a constant geothermal heat flow rate (q_r[W]) of 1000 [W] and a constant surface temperature of the HE pipes (T_s[C]) of 159.93 [C].

V	T_o	q_{rate}
4	45.48	41.50
5	44.01	47.74
6	42.89	53.52
7	41.99	58.92
8	41.26	64.03
9	40.64	68.90

CONCLUSIONS

From the design of the geothermal drying equipment and equations model needed for the design there can be calculated various parameters needed for the drying process. From the calculation and simulation performed in this research it can be concluded the following:

- The drying equipment designed in this research uses a heat exchanger that can be modeled as staggered pipes bank in which fresh air from the atmosphere is flowed through the heat exchanger using an air blower into a drying room filled with trays of products to be dried.
- The equations governing the heat exchanger design can be modeled into two parts. The first part is heat transfer from geothermal fluids inside the pipes to the outer side of the pipe where heat is transferred through convection and conduction modes. The second part is heat transfer from the outer side of the pipe into the drying room where heat is mainly transferred by (forced) convection mode.
- Depending on the product being dried, the drying temperature can be set to find a proper air flow velocity from the air blower. In this experiment, calculation is performed to compute various heat transfer rate and air flow velocity for varying drying temperature using a fixed geothermal fluids temperature of 160˚C.
- Calculation of the experiment using a fixed geothermal fluid flow with a heat transfer rate of 1000W with various air flow velocities of 4 to 9 m/s results in an output drying temperature of 45.48 to 40.64˚C, a temperature range enough for drying purposes, with drying energy (heat) produced in the drying room of about 41.50 to 68.90 kW/m length of the heat exchanger.
- From the simulations performed in the experiments shows that the most important parameters to govern the drying temperature are, among others, geothermal fluid temperature, geothermal fluid flow rate which determines geothermal heat transfer rate, and air flow velocity from the blower.
- This research has to be followed up with more detail experiments and calculations in order to find a complete and thorough design of the equipment that can work optimally.

REFERENCES

Howell, J.R. & Buckius, R.O.: Fundamentals of Engineering Thermodynamics, McGraw-Hill Book Co., 1987, 697 p.

Incropera, F.P. & De Witt, D.P.: Fundamentals of Heat and Mass Transfer, 2nd Ed., John Wiley & Sons, 1985, 802 p.

Sumotarto, U., Surana, T., and Lasman, F., (2000) Utilization of Geothermal Energy for Mushroom Growing, Presented at the 4th INAGA Annual Conference, Jakarta, 25-26 January 2000.

Sumotarto, U.: Problems in Direct Utilization of Geothermal Energy in Kamojang Geothermal Field, Indonesia. Presented at the INAGA Annual Conference, Yogyakarta, 2001.

GEO-HEAT CENTER
Oregon Institute of Technology
Klamath Falls, OR 97601-8801