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HEAT PUMP ASSISTED GEOTHERMAL HEATING SYSTEM FOR FELIX SPA, ROMANIA

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ABSTRACT

The paper presents a pre-feasibility type study of a proposed heat pump assisted geothermal heating system for an average hotel in Felix Spa, Romania. After a brief presentation of the geothermal reservoir, the paper gives the methodology and the results of the technical and economical calculations. The technical and economical viability of the proposed system is discussed in detail in the final part of the paper.

1. THE FELIX SPA GEOTHERMAL RESERVOIR

Oradea is situated in the western part of Romania, in the largest geothermal area of the country, which is part of the Pannonian Basin system. The reservoirs identified in the area are very different from others located in sedimentary basins. Three geothermal reservoirs have been identified in the Oradea geothermal area. The main reservoir is situated almost entirely within the city limits of Oradea, the second one in Felix Spa Resort, 10 km SE from Oradea, and the third near the village Bors, 6 km NW from Oradea.

The Oradea reservoir is located in fractured Triassic limestones and dolomites 2,200 to 3,200 m deep, and the Felix reservoir is located in fractured Cretacic limestones 50 to 450 m deep. The extraction history shows that both are open and hydrodynamically connected reservoirs. The interference test of 1979 showed a natural recharge of 300 l/s. In the Oradea aquifer, the temperature decreases from NW towards SE and continues to decrease into the Felix Spa aquifer. The geothermal bore holes and natural hot springs in Felix Spa have surface temperatures of 35÷55°C.

The chemical composition of the geothermal fluid in the Oradea and Felix reservoirs is the same. The concentration of total dissolved solids (TDS) is up to 1,300 ppm, mostly of calcium-sulfate-bicarbonate type. There are also small quantities of dissolved non-condensable gases (up to 200ppm), mainly CH_4, and CO_2 (Cohut and Tomescu, 1993). A very small content of ^{222}Rn (23÷70 pCi/l) makes the geothermal water unacceptable for human consumption..

Calculations based on the chemical composition of the geothermal fluid (confirmed by practice) show a very low scaling potential, and only at temperatures below 20°C (Rosca, 1993). The geothermal water from the Oradea reservoir is neutral (pH 6 at 20°C). Corrosion problems caused by the geothermal fluid have not been reported up to present. As the Felix reservoir is located in fractured limestones, no sand has been reported to exist in the geothermal water.

At Felix Spa, the geothermal water is used for health and recreational bathing, it's therapeutic properties being known for a long time. About 7,000 beds are available in hotels, almost all having treatment facilities and highly qualified medical staff. Another tourist attraction is the natural reservation of *Nymphea Lotus*, variety *Thermalis*, a flower that is naturally growing in the open in geothermal water ponds. This is quite an uncommon occurrence at this latitude (~45°N). It is well known that geothermal health bathing combined with international tourism is a very lucrative business. Further development of the geothermal field for this purpose is possible, should the tourism market demand increases in the future. It is to be expected that some hotel owning companies will consider the possibility of using the geothermal energy for space and water heating. A pre-feasibility study of a geothermal heating system for a hotel is therefore presented in this paper.

2. THE PROPOSED HEATING SYSTEM

At present, all the hotels in the Felix Spa Resort are connected to a central heating system. The thermal energy is supplied by a coal fired co-generation power plant situated just outside the city limits of the town of Oradea, at a distance of 6 km from the Felix Spa Resort. Until 7 years ago, when the power plant was set on line, every hotel had its own heating system, usually powered by a heavy fuel fired boiler

The subject of this study is a geothermal heating system for a typical average hotel in the Felix Spa resort, as defined below. The study generally follows the guidelines developed by Harrison et al., (1990) and Piatti et al., (1992) for the Commission of the European Communities. For a pre-feasibility study the system is as simplified as possible to enable fast calculations, without access to highly specialized information. This type of a study yields a rough evaluation of the technical and economical viability of a system for a given set of conditions. The results' reliability may be improved by spending more money for a design project and a feasibility study.

For the purpose of this study, it is assumed that geothermal water at 50°C is available at the total required flow rate. This is believed to be a reasonable assumption considering that the reservoir is located right below the Resort, so that any temperature drop in the main pipeline is insignificant. The latest reservoir simulation predicts that an increase in the production rate is possible without significant effects on the reservoir temperature and pressure, provided all the extracted water is reinjected.

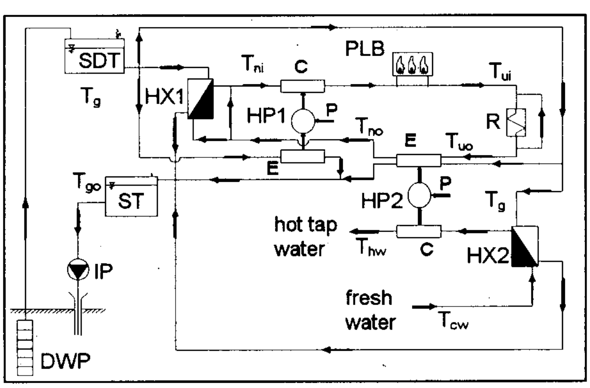

Fig. 1: Hotel heating system layout

The following geothermal heating system has been selected considering the available information and the experience gained in other countries (such as Iceland, France, etc.). For the available well head temperature and maximum inlet and outlet heating water temperatures, systems employing heat pumps were found to be more economic. The basic arrangement of the proposed system is presented in Fig. 1.

The system is basically a heat pump assisted with direct evaporator type. The flow rate of an existing well will be increased by using a deep well pump (DWP). The spent geothermal water will be injected by the injection pump (IP). The DWP can be driven by a variable speed drive, or a storage and degassing tank (STD) for the fresh and a storage tank (ST) for the spent geothermal water can be used to level off the daily variation in demand.

At low partial loads, as long as T_{ui} is below 45°C, the heat demand is supplied through direct heat exchange by the primary heat exchanger (HX1). The condenser of the heat pump (HP1) is by-passed in this case. As the required T_{ui} increases above 45°C, the primary heat exchanger can no longer supply the total heat demand and HP1 is turned on. T_{uo} is increasing at the same time, causing an increase in the geothermal water outlet temperature from the primary heat exchanger. The latter is therefore passed through the evaporator of the HP1, in order to lower the temperature of the waste geothermal water as much as possible. When the network return temperature (T_{no}) reaches 40°C, the heat exchange through HX1 is no longer efficient and it is consequently by-passed. The evaporator of HP1 is fed with geothermal water at well head temperature (T_g). During the periods of time when HP1 is working at partial loads, it is not desirable to regulate its speed continuously in order to ensure all the time the required T_{ui}. It was considered more energy efficient to mix a part of the outlet radiator water with the inlet radiator water. In this way, T_{ui} can be regulated continuously by regulating the mass flow rates of the two streams, while running the heat pump at constant speed. When the required T_{ui} increases above the maximum outlet temperature from the condenser of HP1, the heat supply is supplemented by the peak load boiler (PLB).

The fresh water is heated by direct heat exchange up to the intermediate temperature T_{iw} in the heat exchanger HX2. Subsequently, it is heated up to the standard temperature $T_{hw} = 55$°C in the condenser of the second heat pump (HP2). This insures a decrease of T_{uo}, improving the heat exchange in HX1. During the time the space heating system is turned off, geothermal water at the well head temperature can be fed to the evaporator of HP2.

3. TECHNICAL CALCULATIONS

The usual room heaters in Romania are standard cast iron radiators. The number of elements for each room is determined as a function of the room volume. The standard indoor design temperature is 18°C. The incidental heat gains from external sources, such as solar radiation and human activities (cooking, washing, body heat), increase the indoor temperature usually to about 20°C. The thermal power demand for a constant indoor temperature is then a function of the outdoor air temperature and wind velocity. The design outdoor air temperature for the Oradea area is -7°C. Lower temperatures are occasionally encountered but, as Karlsson (1984) demonstrated, it is neither economic nor necessary to design the heating system for the minimum measured outdoor temperature, because the heat stored in walls, floor, ceiling, furniture etc. tends to level off the indoor temperature variation for short periods of time (up to three days). The temperature demand intensity (T_d) is defined as the difference between the indoor and outdoor temperatures. For the above conditions, the maximum temperature demand intensity is therefore 25°C. In Romania, the thermal power supply is regulated by modifying the inflow temperature of the heating fluid into the radiators while keeping the mass flow rate constant. For the temperature range the radiators are working in, both the inflow and outflow water temperatures can be approximated as linear functions of the temperature demand intensity, as shown in Figure 2.

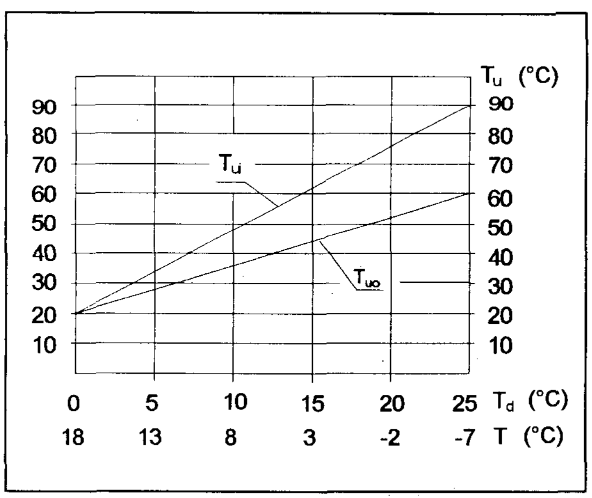

Fig. 2: Temperature characteristics of linearly regulated radiators: T_d - temperature demand intensity; T - outdoor air temperature; T_{ui} - radiator water inlet temperature; T_{uo} - radiator water outlet temperature.

For $T_{d\,max.} = 25°C$, corresponding to $T_{min.} = -7°C$, the maximum temperatures of the radiator water are respectively $T_{ui\,max.} = 90°C$ and $T_{uo\,max.} = 60°C$. When T reaches 18°C, no energy for heating is needed any longer, so $T_d = 0°C$. The indoor temperature is in this case about 20°C, and also the inlet and outlet radiator water temperatures can be considered to be equal to 20°C. Two straight lines between the points defined above approximate T_{ui} and T_{uo} for the entire range of temperature demand intensity.

In order to calculate the annual heat requirements of a single user or group of users and also the power input from different sources for every temperature demand intensity, it is necessary to know the variation of the total heat rate (or thermal power) demand over the year. The usual method is to determine first the variation of the temperature demand intensity with time during one year. This diagram for the Oradea area, based on recorded meteorological data, is given in Figure 3. Usually, the heating systems in Romania are turned off when the daily mean temperature of the outside air is above 10°C for three days in a row. Following this procedure, the average heating season for the Oradea area is 172 days and the minimum temperature demand intensity encountered 5°C.

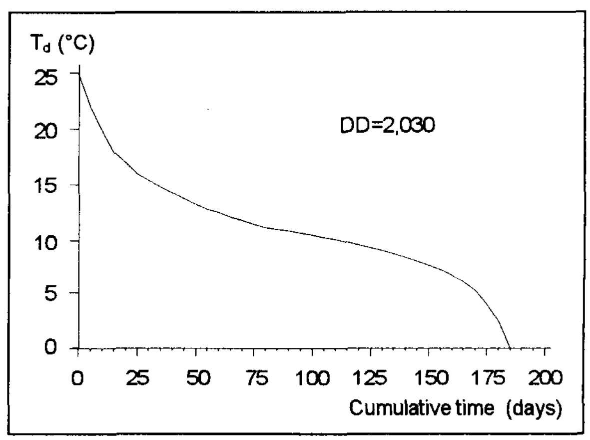

Fig. 3: Temperature demand intensity duration curve for the Oradea area

A hotel with 200 rooms is considered an average sized one for the Felix Spa resort. A standard room is defined as a double-room with a bathroom, having a total volume $V = 70\ m^3$. The additional volume of all ancillary facilities is 25% of the total room volume and can be considered as an equivalent number of standard rooms. The thermal power required for space heating is then:

$$P_u = N \cdot V \cdot G \cdot T_d \tag{1}$$

where:

$G = 1\ W/°Cm^3$ = volumetric heat loss coefficient (typical value assumed),

N = total number of standard rooms (including facilities),

The maximum thermal power demand for space heating is: $P_{u\,max.} = 437.5\ kW$

The maximum number of guests this average hotel can accommodate is 400, twice the number of rooms. The average hot tap water consumption per capita is about 100 l/s/day. In a hotel, the total demand is typically 50% higher (excluding geothermal water for health bathing). The fresh water and standard hot tap water temperatures in Romania are, respectively:

$T_{cw} = 15°C$ = temperature of fresh water (cold)

$T_{hw} = 55°C$ = standard domestic hot water temperature

The heat capacity of the mass flow rate is defined as the product of the flow rate and its specific heat:

$$M = f \cdot \gamma \tag{2}$$

where:

M [W/°C] = heat capacity of the mass flow rate

f [kg/s] = mass flow rate

γ [J/kg·°C] = heat capacity

The thermal power required for tap water heating is:

$$P_w = M_w \cdot (T_{hw} - T_{cw}) \tag{3}$$

where:

$M_w = 2.9\ kW/°C$ = heat capacity of the tap water mass flow rate

The heat capacity of the mass flow rate in the network for space heating is:

$$M_n = \frac{P_{u\,max}}{T_{ui\,max} - T_{uo\,max}} \tag{4}$$

and the calculated value is: $M_n = 14.6\ kW/°C$

The parameters N, V and G are building constants and so the temperature demand intensity duration curve (Fig. 3) is equivalent to the thermal power demand duration curve (Fig. 4), at the scale factor:

$N \cdot V \cdot G = 17.5\ kW/°C$

The thermal power required for heating the tap water (P_w) can be added at the bottom of the graph as a stripe of a constant width for the whole year, at the same scale factor. The total heat demand is thus proportional to the area below the thermal power demand duration curve. The purpose of the technical calculations is to depict which part of the total energy demand is supplied by the various energy sources, such as geothermal energy, electricity and fossil fuel combustion. This will provide the data for the economical assessment of the heating system.

Even if shell and tube heat exchanger already exist at the hotel, they will have to be changed, because these are designed to use hotter water supplied by the co-generation power plant. It will be further assumed that both heat exchangers (HX1 and HX2 in Fig. 1) are stainless steel plate heat exchangers and used in counter flow. The equations used for this type of heat exchangers are identical to those for common counter flow shell and tube heat exchangers.

The power balance for a vapor compression heat pump can be written as:

$$P_h = P_c + P_m \tag{5}$$

where:

P_h [W] = heat rate supplied in the condenser

P_c [W] = heat rate extracted in the evaporator

P_m [W] = mechanical power supplied by the compressor

The thermal powers are given by:

$$P_{h(c)} = M_{h(c)} \cdot (T_{ho(ci)} - T_{hi(co)}) \tag{6}$$

where the subscripts denote the following:

h - the fluid heated in the condenser

c - the fluid cooled in the evaporator

i - inflow　　o - outflow

For a real heat pump, in order to enable the heat transfer over the entire area of the condenser, the condensation temperature of the working fluid has to be higher than the outlet temperature of heated fluid. For the same reason, the evaporation temperature of the working fluid has to be lower than the outlet temperature of the cooled fluid. The temperature difference is, usually, 4°C for both the condenser and the evaporator. Also, the coefficient of performance of a real heat pump is reduced roughly by half by the irreversibility of the thermodynamic processes and the mechanical and hydrodynamic losses. An empirical equation for the coefficient of cooling performance of a real heat pump is (Harrison, 1990):

$$C_c = \frac{0.5 \cdot (T_{co} - 4 + 273.15)}{T_{ho} - T_{co} + 8} \tag{7}$$

The average waste geothermal water temperature can be calculated from the energy balance equation:

$$T_{go} = \frac{M_{gn} \cdot T_{gn} + M_{gw} \cdot T_{gw}}{M_{gn} + M_{gw}} \tag{8}$$

where:

M_{gn} [kW/°C] = heat capacity of the mass flow rate of geothermal water for space heating
M_{gw} [kW/°C] = heat capacity of the mass flow rate of geothermal water for tap water
T_{gn} [°C] = temperature of the waste geothermal water used for space heating
T_{gw} [°C] = temperature of the waste geothermal water used for tap water heating

Using the equations given above and the diagrams shown in Figures 2 and 3, calculations were carried out for specific load values, mainly for T_d values for which the operation mode of the system is changed. The T_d values for which calculations were made and their significance for the operation mode, are:

- $T_d = 0°C$ - only the tap water system is turned on at full capacity (193 days/year);
- $T_d = 9°C$ - heat is transferred to the space heating water through direct heat exchange by HX1 only;
- $T_d = 12.5°C$ - the maximum value for which HX1 is still used and HP1 is operated at half speed;
- $T_d = 16°C$ - HP1 is operated at full speed and PLB is not yet started (HX1 is by-passed);
- $T_d = 20°C$ - HP1 is operated at full speed, PLB is working roughly at half load (HX1 is by-passed);
- $T_d = 25°C$ - maximum load, the HP1 is operated at full speed and the peak load boiler at full capacity (the HX1 is by-passed).

The thermal power demand duration curve plotted by using the calculated data is presented in Figure 4.

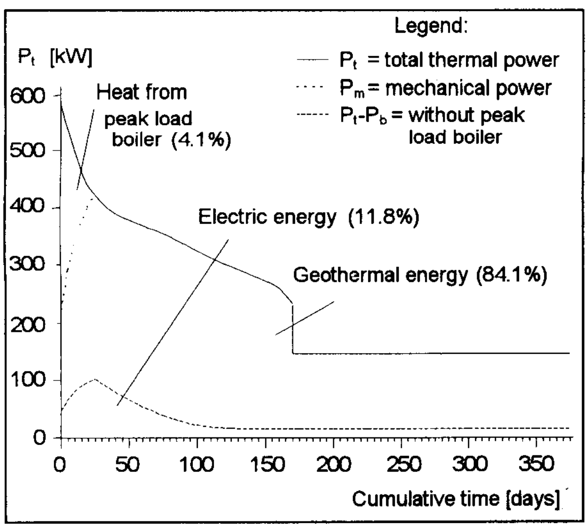

Fig. 4: Thermal power demand duration curve

The annual heat demand is: E_t = 2,128.13 MWh, which comprises: thermal energy from the peak load boiler (E_b = 86.64 MWh), mechanical energy from the heat pump compressors (E_m = 251.28 MWh), and mainly thermal energy from geothermal water (E_g = 1,790.21 MWh).

The design powers of the two heat pumps, accepting a mechanical efficiency for the compressor of 90% and the electrical motor efficiency of 95%, are:

HP1: -heating power: P_{h1} = 321 kW
-mechanical power: P_{m1} = 86 kW
-electrical power: P_{e1} = 101 kW
HP2: -heating power: P_{h2} = 59 kW
-mechanical power: P_{m2} = 16 kW
-electrical power: P_{e2} = 19 kW

Considering the peak load boiler as a common one, with an efficiency of 75%, fired by heavy fuel oil with a low calorific value of 11.8 kWh/kg, the annual fuel consumption is: F = 9,790 kg

4. ECONOMICAL APPRAISAL

The economical appraisal was carried out basically following the methodologies presented by Harrison et al., (1990) and Piatti et al. (1992). The Romanian economy is now suffering a transition process, from being a centrally planned system towards a free market economy. In this situation prices are changing fast and at an uneven rate, due to a high inflation rate, changes in the subsidizing policy, etc. The general tendency of prices is to approach international market values, energy prices being among the quickest to follow this trend. By the time the possibility to implement a project of this type qualifies for consideration, the Romanian economy will probably be fairly stabilized and the problems outlined above less acute. For the above reasons, it was considered appropriate to carry out the economical appraisal of the project on the basis of economical conditions prevailing in the European Economic Community, all costs being given in ECU.

The capital investment comprises purchasing and installation costs for all equipment required for the new heating system, engineering cost and additional costs due to contingencies. The new equipment required for the geothermal heating system are:

- main piping network for the geothermal water supply, including valves and meters (for flow, temperatures and pressures) = C_{MN}
- heat pumps for tap and radiator water heating (regulation and control systems included) = C_{HP}
- heat plant network, including stainless steel plate heat exchangers, valves and control system = C_{SN}

All other equipment, such as storage tanks, user supply and return pipelines complete with valves (for rooms and ancillaries), circulation pumps and the boiler, already exist at the hotel. The total capital investment cost (C_T) can thus be calculated as:

$$C_T = C_{MN} + C_{HP} + C_{SN} \tag{9}$$

with:

$$C_{MN} = c_n \cdot P_t \cdot K_1 \tag{10}$$

where:

c_n = 150 ECU/kW = specific cost of a reference network

P_t = 585 kW = design power of the system

K_1 = non-dimensional correction coefficient depending on the difference between the design supply and return temperatures

$$K_1 = 1 + 0.2 \cdot \frac{30}{\Delta T} \tag{11}$$

The costs of the heat pumps and their prime movers (i.e. electric motors) and also the cost of the heating station are estimated as follows:

$$C_i = C_{0i} \cdot \left(\frac{P_i}{P_{0i}} \right)^{n_i} \tag{12}$$

where:

C_{0h} = 700 kECU = reference cost of a heat pump with the P_{0h} thermal power

C_{0m} = 100 kECU = reference cost of an electric motor with the P_{0m} shaft power

C_{0s} = 250 kECU = reference cost of a heating station with the P_{0t} thermal power

P_{0h} = 4 MW = thermal power of the reference heat pump

P_{0m} = 1 MW = shaft power of the reference electric motor

P_{0s} = 10 MW = thermal power of the reference heating station

P_h [MW] = total design thermal power of the heat pumps

P_m [MW] = total design shaft power of the electric motors

P_t [MW] = total design thermal power of the heating station

n_h = 0.8 = scale factor for heat pumps

n_m = 0.7 = scale factor for electric motors

n_s = 0.65 = scale factor for the heating station

The total engineering cost for a heating system with a design thermal power of less then 10 MW is estimated as 8% of the investment cost. Additional costs due to contingencies are estimated as 5% of the total investment cost, including engineering.

The total capital investment is thus: C = 270 kECU.

The annual running cost of the project comprises the costs of electricity, boiler fuel and geothermal water, maintenance costs and wages for the personnel required to operate the geothermal heating system. The total annual maintenance costs are estimated as 2% of the capital investment. The geothermal heating system for an average hotel is not a large one, so that a single worker is required to operate it. The annual wage of this worker is 25 kECU. The specific costs of the different forms of energy are:

c_e = 0.050 ECU/kWh = specific cost of electric energy

c_g = 0.020 ECU/kWh = specific cost of geothermal energy

c_b = 0.020 ECU/kWh = specific cost of thermal energy supplied by boiler

The annual running cost is: R = 82.6 kECU

The annual earnings are considered to be the costs of continued running of the former heating system which is to be discontinued. The maintenance costs and wages are considered to be approximately the same. The specific cost of the thermal energy supplied by the power plant is: c_p = 0.043 ECU/kWh
The annual earnings are then: E = 121.9 kECU

The annual running cost and earnings are considered constant for the entire estimated life span of the project. Since these payments are made at different times, they have different economic values. The discounted cash flow analysis is usually adopted for the economical appraisal:

For the purpose of this study, it has been assumed that the company finances the investment to the tune of 50% through equity contribution (Q) and 50% by a fixed interest bank loan (debt contribution - D). The equity and debt values are: Q = D = 135 kECU.

The company owning the hotel is a taxable company. The annual taxation rate for a company of this type is t = 30%. All expenses, such as running cost, debt repayment and usually the annual depreciation of equity, are tax deductible, called tax allowances. The annual earnings of the project, as defined above, are tax allowances while the current heating system is in use, but because it is not paid any more when the new system is employed, it is added to the revenues of the company and so becomes a taxable quantity.

It is further assumed that the whole of the investment cost is committed at the beginning of the project, before its operation starts. After commissioning, only running costs, debt charges and taxes have to be paid. The project lifetime is assumed to be n = 20 years and inflation is not considered. This means that all payments remain constant in real values over the entire project lifetime. All the financial rates are also considered constant over the project lifetime. A yearly compound period is considered for all payments. This may not reflect real life practice, but is sufficiently accurate for this study.

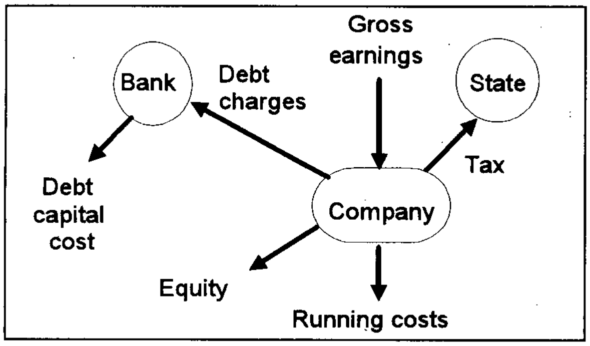

Fig. 5: Financial system of the project

The financial system of this project is presented in Figure 5. The discount rate (r) required to calculate the Capital Recovery Factor (CRF) depends on how the company perceives the worth of money. It should compensate the company for future risk (expected payments that may not materialize) and for lost opportunity (to spend the money on other, more profitable, ventures).

$$CRF(n,r) = \frac{r \cdot (1+r)^n}{(1+r)^n - 1} \tag{13}$$

For a hotel owning company a discount rate of r=9% is considered to be reasonable. Then, for a lifetime of n=20 years and the system defined above, the CRF comes out as: $CRF(n,r) = 0.1095$

Assuming that the pay back time for the bank loan equals the project lifetime (i.e. n = 20 years) and an annual interest rate of i = 8%, the CRF for the loan becomes: $CRF(n,i) = 0.1019$

Annual debts charges (annuity):

$C = D \cdot CRF(n,i) = 13{,}756.5$ ECU

Annual depreciation of equity:

$p = Q/n = 6{,}750$ ECU

Total annual tax allowances:

$A = C + p + R = 103{,}106.5$ ECU

Taxable annual earnings:

$X = E - A = 18{,}793.5$ ECU

Annual tax:

$T = t \cdot X = 5{,}638$ ECU

Net earnings after tax:

$N = E - R - T - C = 19{,}905.5$ ECU

Indices for evaluating the economical feasibility of this system can now be defined and calculated. These are the Net Present Value (NPV), Internal Rate of Return (IRR) and Discounted Pay-back Time (DPT).

The NPV is defined as the present value of the total earnings of the project over its lifetime, after the present values of all expenses have been deducted. A positive value of this index means that the project is economically viable. For the financial system defined above, the NPV can be calculated with the following equation using the CRF for the considered discount rate (r):

$$NPV = \frac{N}{CRF(n,r)} - Q \tag{14}$$

The IRR is defined as the discount rate which ensures that the project breaks even over its lifetime, for a fixed level of revenue. It is calculated by a trial and error method or graphically. The IRR is considered to equal the value of the discount rate (d) for which the following function is zero:

$$(NPV)_d = \frac{N}{CRF(n,d)} - Q \tag{15}$$

The DPT is defined as the number of years required to pay back the initial investment, at the discount rate (r). After this time, the present value of the net earnings equals the equity (Q). A value of this coefficient, which is lower than the project lifetime, indicates that the project is economically viable. For the financial system defined above, the DPT can be calculated graphically or by trial and error, using the following equation:

$$(PVN)_j = \frac{N}{CRF(j,r)} \tag{17}$$

The calculated values are:

NPV = 46,785.4 ECU
IRR = 13.6%
DPT = 11 years

5. DISCUSSION

Technical and economical calculations have only been carried out for the system presented in Figure 1. A basic assumption was that required modifications of the existing heating system should be minimized, implying that the room heaters should not be changed. Other technical solutions are also viable. The well head temperature of the geothermal water in the Felix reservoir is one that suits various low temperature room heater systems, i.e. floor, wall and/or ceiling heating. Since no scaling problems will be encountered provided the geothermal water is not cooled to below 20°C, it can be used directly in these heating systems. This type of heating requires no heat exchangers and incidental temperature losses will be minimal. The variations in heat demand can be met by regulating the geothermal water flow rate and mixing a portion of the return water with the inlet water. The return water temperature can also be regulated by temperature controlled valves, insuring optimal use of the geothermal water. This requires, on the other hand, a total refurbishing of the existing heating system, since floor heating is currently not used in Romania.

The outlet temperature from the evaporators of both heat pumps was selected as 20°C. The main reason was to avoid silica scaling, whilst maintaining a reasonable coefficient of performance for the heat pumps. The maximum outlet temperature from the heat pump condensers have been selected as 60°C for space heating and 55°C for tap water heating. This reduces the thermal energy required from the peak load boiler, providing also reasonable values for the coefficients of performance. The energy balance for the heat pumps has been made on a theoretical basis, assuming that the heat pump operates at optimum parameters for every load. Under real conditions, when the heat pump is operated at partial loads the coefficient of performance will be slightly lower than the theoretical value. For these reasons, the curve for the mechanical power in Figure 4 is only of informative value. For a more accurate calculation, certain heat pumps have to be selected.

With the financial premises considered above, the project is shown to be economically viable. The net present value over the 20 years of project lifetime, although not very high, is still positive. This means that the project is profitable, so the company will not lose money if the decision is made to change the heating system to a geothermal one. The discounted pay back time of 11 years is fairly reasonable at about half the project lifetime. The internal rate of return (IRR = 13.6%) is also a reasonable one, higher than the considered discount rate (r = 9%), even if it is probably on the low side for a small company, particularly during the initial stages in the operation of a hotel. Before a binding decision can be made, a more detailed economic appraisal is recommended. The study should be based upon the financial situation existing at the specific time and also take account of available financial forecasts. When the investment can be made from own capital resources, as equity, the internal rate of return is higher in inflationary conditions. The influence of inflation should also be considered in an economic feasibility study. Changes of energy prices will also affect the economical viability of this project. Fossil fuel prices are expected to increase in the future, due to depletion of the resources, combined with announced environmental energy taxes. This will make a geothermal heating system more profitable.

By maintaining the pressure in the geothermal loop above 4 bar (too keep the gasses in solution) and by re-injecting the spent geothermal water pollution is completely avoided. The pollutant emissions from a co-generation power plant fired by low grade coal, as those in Oradea city, for producing the heat that can be supplied by geothermal water, are: 24,000 t/year of CO_2, 34 t/year of SO_2, 38 t/year of NO_x, 6 t/year of solid particles and 2,182 t/year of ash, which requires expensive pollution free means of disposal.

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