MODIFICATION OF WORKING FLUID IN GEOTHERMAL ORGANIC RANKINE CYCLE ENGINES

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ABSTRACT

The organic Rankine cycle engines for geothermal water are in general fitted with full admission reaction turbines. Thus the rated power cannot be varied by altering the number of active nozzles. The power however has to be varied to match the grid requirements and the water temperature variation. To substitute the working fluid is an alternative to the power reduction by lamination of the vapour at the turbine inlet. The influence of fluid substitution vs throttling is discussed with reference to a 100 kW engine. Experiments were made on a 4 kW engine, using perchloro-ethylene-trichloro-ethylene mixtures to investigate if operational difficulties could arise due to erratic fluid composition at the interface. Mixtures seem promising, particularly if the variable change-of-state temperatures are properly exploited.

INTRODUCTION

Geothermal hot water as a heat source for electric power production is a variable source in many respects. In fact the temperature, the flow and the composition of water cannot be reliably predicted. Also, during the operating life of the well and of the associated power conversion module the water temperature generally drops from an initial value to a threshold value under which the well is no longer viable. Together with these heat-source related variations, the variation of the grid requirements should be considered. In fact the small power, hot-water geothermal plants, in general, are connected to small grids that are still in the growth phase. Hence the engine will receive the highest temperature water when the mean required power is still low. When steam turbines are used, the shaft power can be changed by altering the number of open nozzles at the turbine inlet. In the case of Organic Rankine Cycle (ORC) engines, the turbines are often of the full admission reaction type. The adoption of high reaction turbines is the main reason why the ORC engines allow a better exploitation of geothermal hot water, together with the favourable heat input curve. The admission arc of the reaction turbine cannot be partialized without large efficiency losses; thus part load operation has to be obtained by throttling the vapour at the turbine inlet.

In the present paper the change of working fluid is considered as an alternative to throttling in order to modify the ORC engine power level. Other items are available to adapt the power level, e.g., several modular engines can be adopted, to be installed in sequence, according to the growing demand. However, as the temperature drops the engines designed for the initial temperature will be underutilized.

REFERENCE PLANT

For this comparison, a 100 kW reference plant was considered, having 95°C initial evaporation temperature and 45°C constant condensation temperature. The data for the engine performance, both thermal and fluid dynamic, were obtained from experimental data measured on a low temperature, low pressure engine featuring a 3000 rpm turbine directly coupled to the electric generator (Angelino, Gaia and Macchi, 1984a). The fluid selection criteria for this kind of engine are

a) low vapor pressure (sub-atmospheric in both condensor and evaporator)

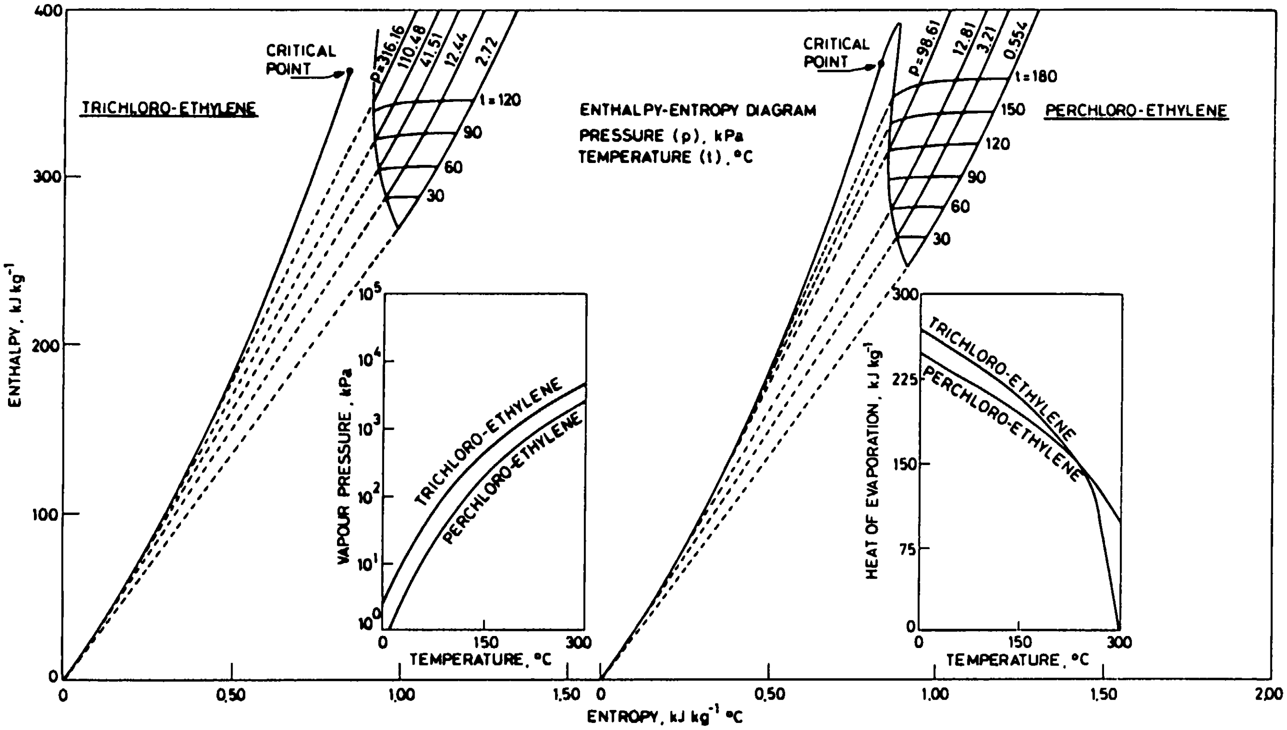

Figure 1. Mollier diagram, and vapour pressure and heat of vaporization vs. temperature for perchloro- and trichloro-ethylene

b) large molecular mass, yielding a small enthalpy drop in the turbine for the given temperature difference.

These criteria lead to large turbines, probably more expensive than fast, small chlorofluoro-carbon turbines but simple and reliable. The low evaporation pressure allows, in many cases, substitution of the feed pumps with gravity feed (Angelino, Gaia and Macchi, 1984b), further improving the engine reliability. Very high overall efficiencies have been obtained by adopting these design criteria for solar applications (Gaia and others, 1983), due to the high intrinsic efficiency of the saturated cycle, the high efficiency of the large low pressure turbine, and the low parasitic power consumption for the auxiliaries.

For the present comparison, perchloro-ethylene was adopted as a working fluid well suited for this kind of geothermal application while trichloro-ethylene was considered as a suitable higher pressure substitute. These fluids were practically employed several times by the authors, in different engines. The main data for the considered fluids are given in Table 1. Thermodynamic tables were prepared utilizing the procedure described in Invernizzi (1984). Simplified Mollier charts are given in Figure 1, together with the latent heat and vapour pressure as a function of temperature. The two fluids have similar properties, except for the vapour pressure of trichloro-ethylene which, e.g. for the adopted condensation temperature, is more than three times the pressure of perchloro-ethylene. The reference cycle in a T-s diagram and the plant scheme are given in Figure 2.

PART-LOAD OPERATION WITH CONSTANT EVAPORATION TEMPERATURE

The part-load performance curve for the reference engine at design evaporation temperature is given in Figure 3. The constant temperature assumption was made in order to allow a more straightforward comparison. In fact the economic optimization of the temperature differences in the exchangers yields for the geothermal plant a very small pinch-point temperature difference (around 3° C); as a consequence, the heat flow reduction at part-load operation does not involve a relevant modification of the change-of-state temperature. The turbine efficiency is mainly affected by the variation of the expansion ratio, the curve reported in Figure 4, which was obtained with the method discussed in Macchi and Perdichizzi (1981). The improvement in efficiency obtained by the fluid substitution is around 70 percent for the fully open valve condition with perchloro-ethylene. Evidently the fluid substitution is a way to adapt the engine maximum power to the grid requirements; it is not possible to adopt it as a way to obtain a fast modulation of the power produced as by throttling.

Table 1. Thermodynamic Data of the Considered Working Fluids

	C_2HCl_3	C_2Cl_4
Molecular mass °	131.39	165.83
Critical Temperature*	297.85	346.85
Critial Pressure+	4.91	4.46
Normal Boiling Point*	87.10	120.97

° kg kmole^{-1}
* °C
+ MPa

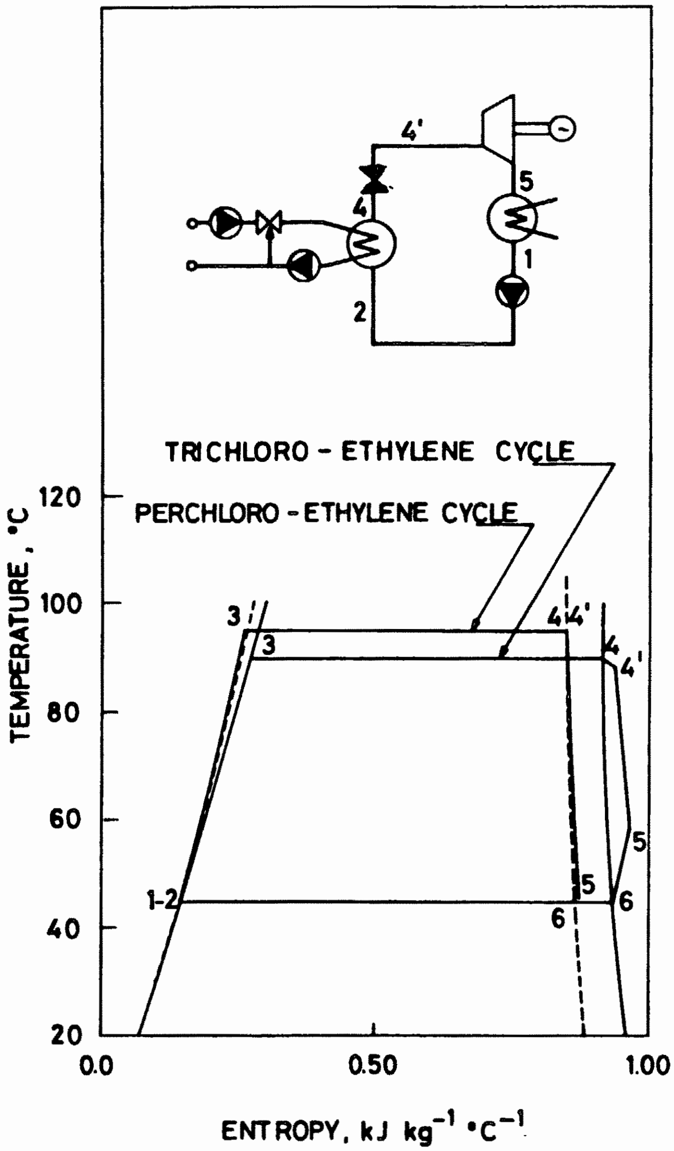

Figure 2. Reference cycle in the T-s coordinates and plant scheme

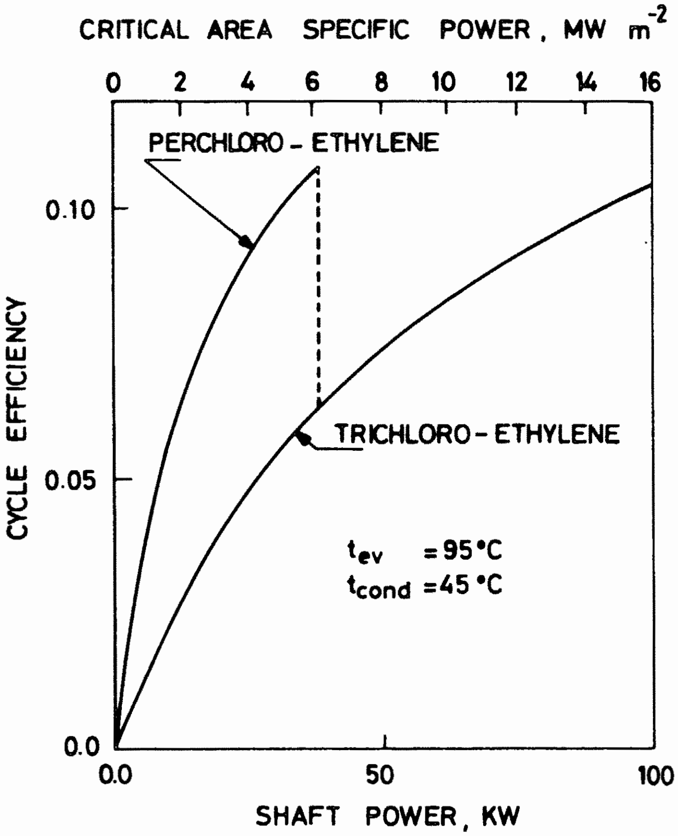

Figure 3. Part load efficiency, the power reduction being obtained by throttling and by fluid substitution

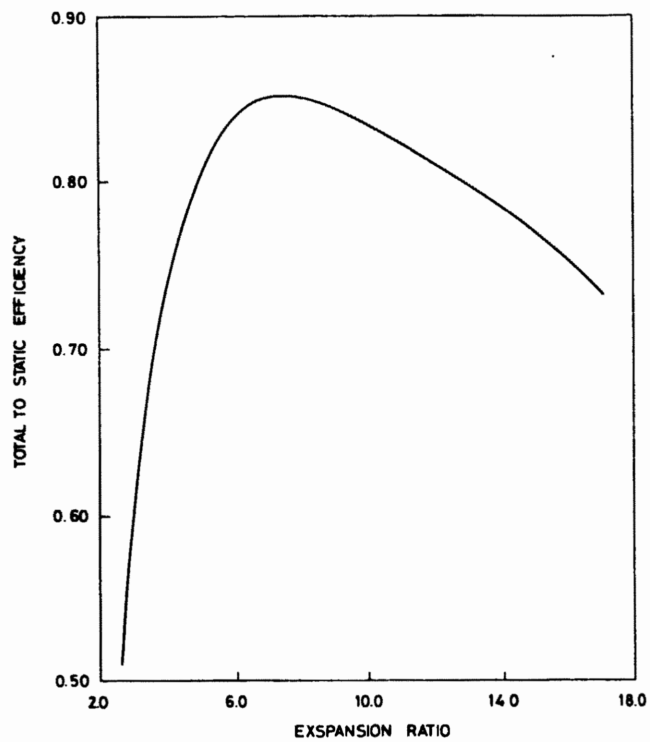

Figure 4. Turbine total to static efficiency for the reference engine vs. expansion ratio

The power reduction can be also obtained by reducing the evaporation temperature, i.e. by reducing the temperature of the water leaving the evaporator. Often however a large temperature drop of the water through the evaporator is not desirable in order to avoid separation of solute in the geothermal water and subsequent tube clogging.

CONSTANT POWER WITH DECREASING WATER TEMPERATURE

It is generally necessary to avoid a reduction of the produced power from a geothermal engine when the heat source temperature diminishes. In order to have a constant power either the turbine bladings have to be designed to produce the rate power at the minimum temperature to be used at full power (hence to be largely oversized), or substitution of the fluid has to be considered. The efficiency obtained in both ways is compared in Figure 5 as a

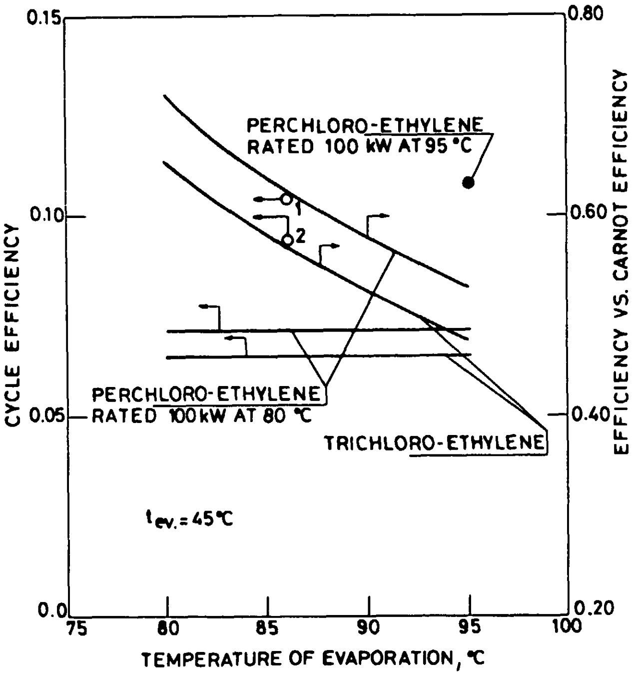

Figure 5. Efficiency vs. evaporation temperature, for oversize C_2Cl_4 turbine (260 kW, throttled to 100 kW) and with fluid substitution

function of the evaporation temperature for a constant 100 kW power. The dot at 95° C represents the design point operation with perchloro-ethylene while the curves nearly parallel to the abcissa represent the operation of the same engine with trichloro-ethylene and of the oversized perchloro-ethylene engine respectively. The throat area for the oversized C_2Cl_4 turbine giving 100 kW at 80° C would allow 260 kW at 95° C if the admission was not limited by the inlet valve to the required 100 kW: all the rotating components have to be designed to sustain that power during the valve operation transients. To avoid oversizing the fluid path is obviously an advantage for the turbine cost; moreover, if the field is expected to operate for a long time around the rated evaporation temperature the global produced energy can be significantly higher.

USE OF NONAZEOTROPIC MIXTURES

The advantages that can be obtained by switching to another working fluid when either the required power or the available temperature are different from design values are partially jeopardized by the lack of suitable working fluids. In fact the requirements for a working fluid are rather stringent and even for low temperature engines there is no such thing as a family of similar working fluids with a continuum of progressively rising vapour pressures at a given temperature. In particular the fluid for geothermal application should have a rather simple molecular structure to avoid superheating during the expansion from saturated vapour. This is to minimize the need for cooling during the condensation process, which is costly, due to the unfavourable mechanical work vs heat ratio and generally leads to a lower exploitation of the hot water stream. Also the fluid should be available at

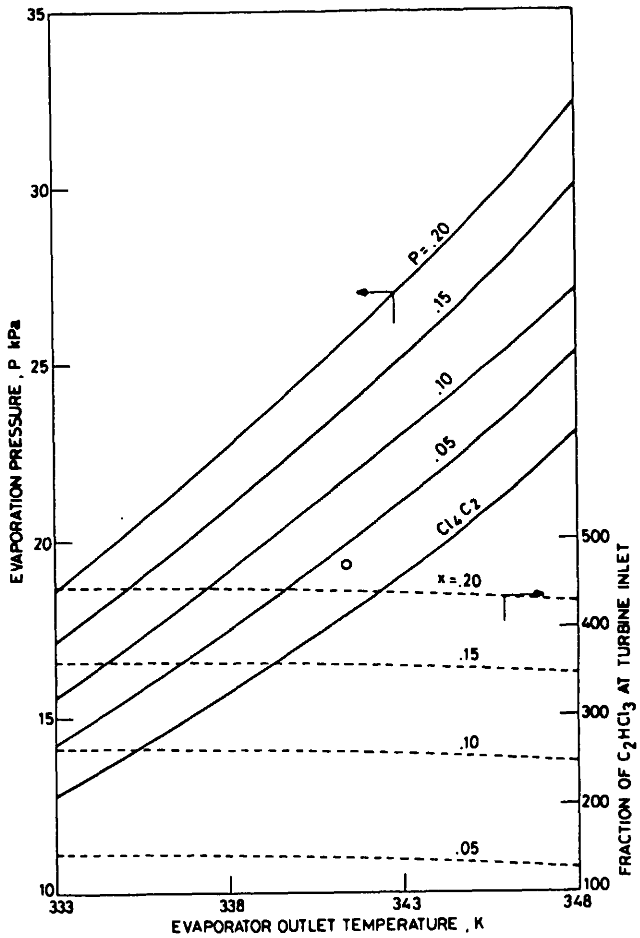

Figure 6. Evaporation pressure vs. temperature for liquid fraction

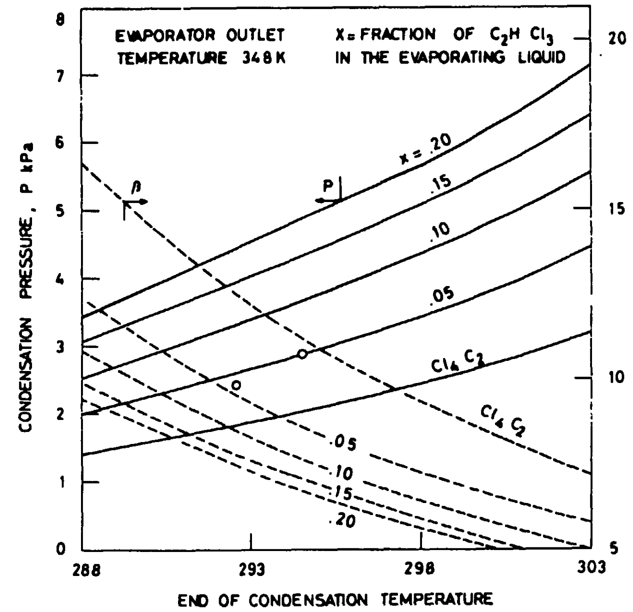

Figure 7. Condensation pressure vs. end of condensation temperature

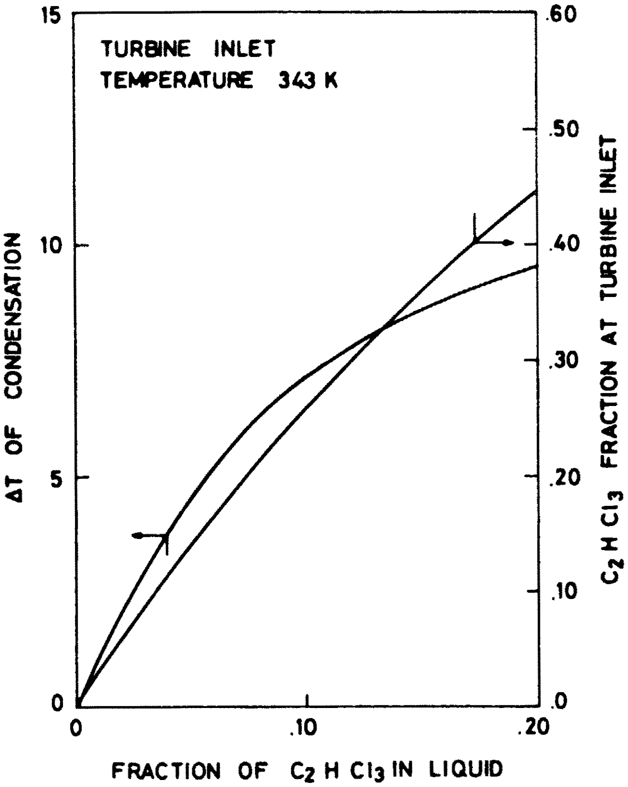

Figure 8. Temperature span during condensation

reasonable cost, not too dangerous and toxic, and noncorrosive for the metals to be used. All these requirements limit the choice to a few eligible fluids having rather large pressure differences at a given temperature.

A continuum of properties between pure fluids can be obtained by mixing the fluids in different amounts, provided the fluids do not give birth to azeotropes. Besides the advantage of obtaining intermediate properties it is easier to add known amounts of fluid than to substitute the fluid, which requires fully draining and disposing of the initial fluid.

Mixtures change phase at variable temperatures and this fact can be exploited to obtain better heat exchange performance and heat source exploitation, as often proposed for the heat pump application (IEA Heat Pump Center, 1984). The geothermal ORC engines are essentially comprised of two cross-flow shell-and-tube heat exchangers, which do not derive any advantage from a variable temperature heat exchange. However, the mixed fluid solution could give way to some improvement in off-design cycle efficiency, if compared to fluid substitution and throttle.

A preliminary evaluation of cycle performance with fluid mixtures was done using Raoult's law. Although it is known that the assumptions made in the derivation of Raoult's law are often unrealistic, such a model was used mainly with the target of achieving a rough estimation and the direction of change of the phenomena under consideration. It was supported by a certain degree of confidence in the fact that the similarity of the chemical nature of the mixture components would help to make the liquid phase behave like an ideal solution.

The predicted evaporation pressure as a function of vapour maximum temperature is plotted in Figure 6 for a number of vapour-liquid equilibrium compositions of the liquid phase, together with the composition of the vapour at the turbine inlet. The turbine expansion ratio is always decreased by the addition of a more volatile fluid because of the mixture behaviour in the condenser section (Figure 7). As it has been said, mixture performance can be consistently improved if counter-flow condensers are

Table 2. Design Features of the ORC Engine Used For Experiments

Working fluid	perchloro-ethylene
Evaporation temperature, °C	75
Condensation temperature, °C	30
Turbine shaft power, kW	4
Turbine angular velocity, r s^{-1}	1257
External heat exchange surface, m^2	
evaporator (nonbaffled, shell and tubes)	17
condenser (nonbaffled, shell and tubes)	17
Feed pump	none (gravity feed)

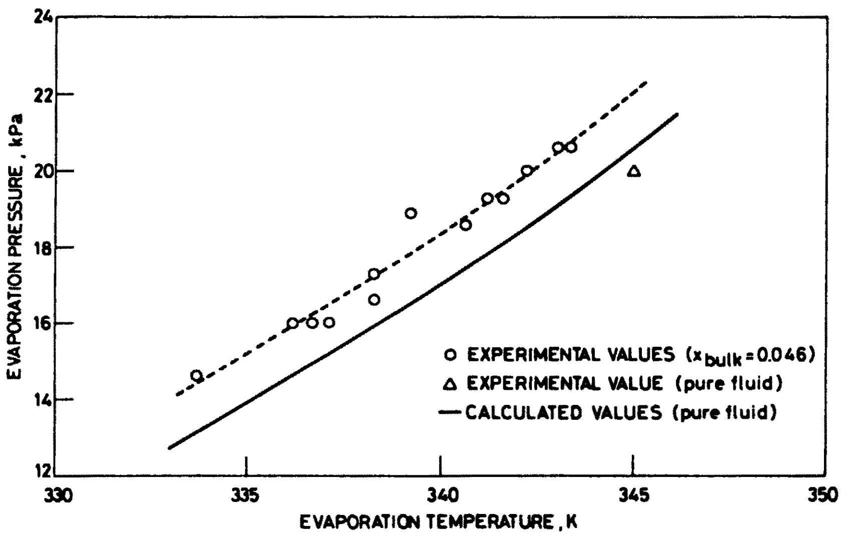

Figure 9.
Measured evaporation pressure vs. temperature

adopted. The difference between the temperatures of beginning and end of condensation is shown in Figure 8, for a given cycle maximum temperature, as a function of fraction of C_2HCl_3 in liquid (which means also an increasing fraction of C_2HCl_3 in the vapour, although less than proportional to the former one). If the entire life cycle of the geothermal plant is considered, it is clear that interest in adopting counter-flow condensers is higher for the cases when the source-temperature decrease in time is not such that a pure C_2HCl_3 cycle will rapidly become the preferred one.

Experiments were carried on with a small ORC engine, conceptually similar to larger ORC engines for geothermal use. The design data for the engine are given in Table 2.

The measured evaporation pressure vs temperature is given in Figure 9, and compared with a simple measured point for pure fluid. The engine was run several times under the conditions reported as circles in Figures 6 and 7. The experiments confirm that noninstability occurs even at start and fast transients. A substantial improvement in efficiency at part load should be obtained by using mixtures with open throttle instead of pure fluid both with and without the counter-flow design of heat exchangers (circles 1 and 2 in Figure 5 respectively).

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