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An Organic Total Flow System for Geothermal Energy and Waste Heat Conversion

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ABSTRACT

A two phase flow expander modified by an air helical compressor has been successfully tested. This makes possible to realize the triangle cycle, e.g. a total flow power system in the wet-vapor rigion. The total flow system with water-steam working fluid has been extensively discussed in open literature. This paper describes a new total flow system with organic working fluid. Thermodynamic analysis of the system and its comparison with other common organic systems have been performed. The performance characteristics of a geothermal power generating unit using the proposed method with capacity of 500kw are presented. It is concluded that the organic total flow system may be singnificantly more efficient (30-60%) than other organic systems. In adition, the efficency advantage of the system increases with decreasing the heat source temperature. Conseqently, the "economic utilization temperature" of resources might be greatly decreased.

1. Introduction

To date, the organic Rankine cycle and flashed steam system are successfully in use for geothermal energy and industrial waste heat conversion. In the early 1970's, a more efficient total flow system for hot water or steam-water mixture power generation has been proposed[1] and attained great success in research and development. In order to introduce greater flexibility in application of the total method not only for steam-water fluids but other energy resources (eg: exhaust gases and petrochemical hot fluids), an organic total flow system (OTFS) is suggested. Using two phase expansion machines and organic media as its working fluid, the OTFS may be realized. This paper studies the possibilities and advantages of OTFS. Thermodynamic specifications of the system are analysed. The performance characteristics of a geothermal power generating unit using the proposed method with capacity of 500kw are presented.

2. Progress of the Organic Cycle

The most widely used organic thermodynamic cycle for low-temperature energy power generation is the organic Rankine cycle(ORC), as shown in Figure 1. Packaged, modular ORC power units are available commercially from many countries in the world. Nevertheless, from thermodynamics point of view, in the ORC, there is difficult to have a high utilization efficiency because of badly matching between the heat source cooling

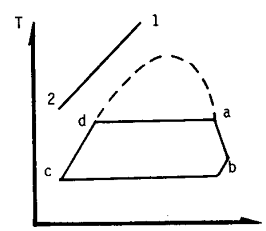

Fig.1 Organic Rankine Cycle

curve(1-2) and working fluid heating curve(c-d-a)(see Figure 1). The temperature differenctial is uneven and large throughout the heat exchanger. In addition, there is also difficult to cool the heat source below the evaporation temperature of the working fluid owing to cycle limitations. Thus, less heat can be recovered from the heat source in the cycle.

Another thermodynamic cycle is the organic supercritical cycle (OSC), as shown in Figure 2. Theoretically, OSC can be nearly reach the ideal one--"Triangle cycle" by means of technology. But actually, it is too costly to realize the cycle because of pump power. As the pressure level is very high, the working fluid pump may generally require as much as 30-40% of the shaft power of the turbine. Thus the net power output of the cycle will be greatly decreased. In some cases(eg: for certain fluids and pressure

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is too high), quite the reverse of what we expect might be happened.

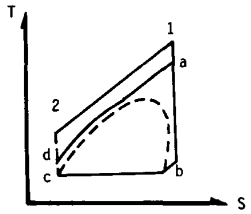


Fig.2 Organic Supercritical Cycle

To convert the thermal energy of the heat sources into mechanical shaft work, the steam turbines are used in two cycles mentioned above. If the two-phase expansion machines were adopted, the ideal cycle could be realized without difficulties. Figure 3 shows an organic vapor-liquid two-phase triangle cycle using total flow expander. It is so

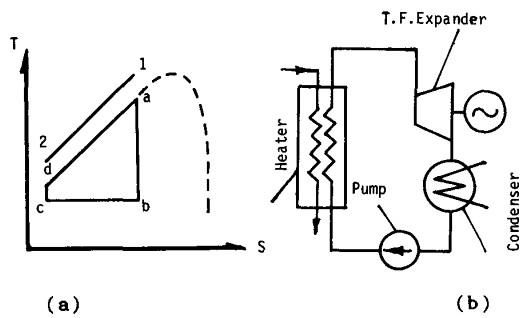


Fig.3 Organic Total Flow System

called an Organic Total Flow System. As shown by Fig.3, there is a considerably better macth between the heat source cooling curve(1-2) and working fluid heating curve(d-a). The temperature difference is small and constant. Heat can be recovered from the heat source over its entire range of allowable temperature as low as the environmental one. Thus, the OTFS has a higher utilization efficiency than both ORC and OSC. It will be proved that the efficiency advantage of the OTFS increases with decreasing heat source temperature. The "economic utilization temperature" of resources may be greatly decreased. Possibly, OTFS will become an effective approach to advance the state-of-the-art of the organic cycle.

3. Thermodynamic Analysis of Organic Total Flow System

From Fig.3, for the organic total flow system the power output per unit mass of the working fluid can be calculated by

$$W_e = h_a - h_b = AC_{pr}\left[(T_a - T_b) - T_b \mathrm{Ln}\frac{T_a}{T_b}\right] \quad (1)$$

Where h_a, h_b are the enthalpies of the working fluid at point a and b, T_a, T_b are the temperatures of the point a and b in cycle respectively, A is a conversion constant and C_{pr} is specific heat of the fluid.

Pump work as indicated in Fig.3 is:

$$W_c = V_c (P_a - P_b)$$

Where V_c is the specific volume of the fluid at point c, P_a, P_b, are the condensing and heating pressures.

The net work from the system is:

$$W_O = W_e - W_c = AC_{pr}\left[(T_a - T_b) - T_b \mathrm{Ln}\frac{T_a}{T_b}\right] - V_c (P_a - P_b) \quad (2)$$

From heat equivolent we have:

$$G_w (h_1 - h_2) = G_r (h_a - h_b) \quad (3)$$

Where G_w, G_r are the mass flow rates of the heat source and the working fluid respectively, h_1, h_2 are the inlet and outlet enthalpies of the heat source.

The power output per unit mass of the heat source, neglecting the pump work, is:

$$W = \frac{AC_{pw}(T_1 - T_2)}{T_a - T_b}\left[(T_a - T_b) - T_b \mathrm{Ln}\frac{T_a}{T_b}\right] \quad (4)$$

Where T_1, T_2 are the inlet and exhaust temperatures of the heat source.

Assuming the heat exchange temperature difference δT_e, we have

$$T_a = T_1 - \delta T_e \quad \text{and} \quad T_2 = T_b + \delta T_e$$

Then equation(4) becomes:

$$W = AC_{pw}\left[(T_1 - \delta T_e - T_b) - T_b \mathrm{Ln}\frac{T_1 - \delta T_e}{T_b}\right] \quad (5)$$

The thermal efficiency of the cycle is defined(neglecting the pump work):

$$\eta_t = \frac{W}{Q_o} = \frac{(T_1 - \delta T_e - T_b) - T_b \, Ln \dfrac{T_1 - \delta T_e}{T_b}}{T_1 - T_o} \tag{6}$$

Where Q_o is the overall released heat when an unit mass of the heat source is cooled from the initial temperature T_1 to the ambient temperature T_o.

When $T_b = T_o$, the maximun efficiency is obtained as:

$$\eta_{t\,max} = \frac{(T_1 - \delta T_e - T_o) - T_o \, Ln \dfrac{T_1 - \delta T_e}{T_o}}{T_1 - T_o} \tag{7}$$

The exergy efficiency of the heat source is:

$$\eta_e = \frac{W}{E_o} = \frac{(T_1 - \delta T_e - T_b) - T_b \, Ln \dfrac{T_1 - \delta T_e}{T_b}}{T_1 - T_o - T_o \, Ln \dfrac{T_1}{T_o}} \tag{8}$$

Where E_o is the exergy of the heat source.

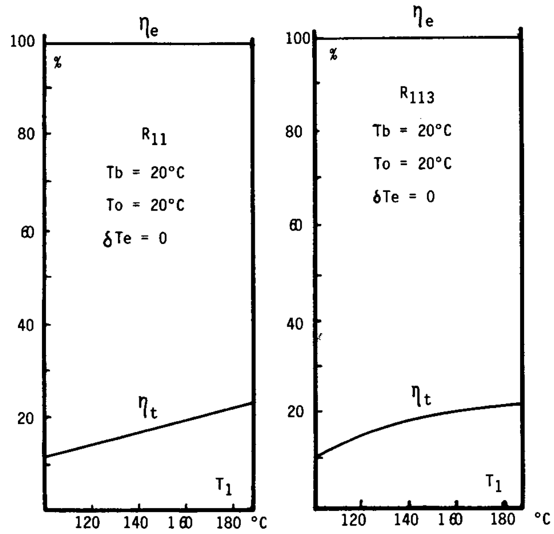
The maximun exergy efficiency is obtained as:

$$\eta_{e\,max} = \frac{(T_1 - \delta T_e - T_o) - T_o \, Ln \dfrac{T_1 - \delta T_e}{T_o}}{T_1 - T_o - T_o \, Ln \dfrac{T_1}{T_o}} \tag{9}$$

For the ideal cycle, $\delta T_e = 0$, $T_b = T_o$, then we have:

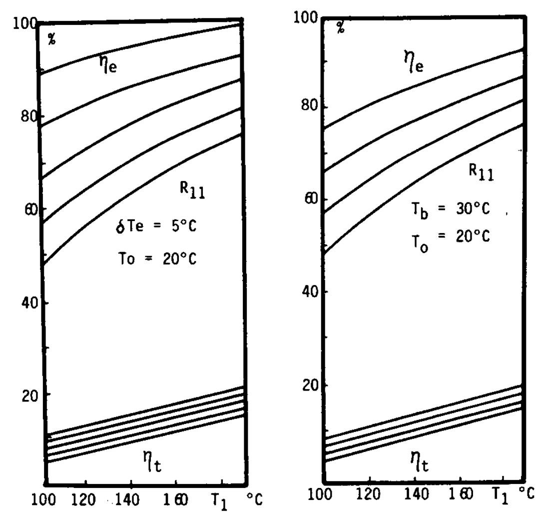
$$\eta_e = 1 \tag{10}$$

Figure 4 and 5 illustrate the thermal and exergy efficiencies of organic total flow system as a function of the heat source temperature. From the figures we can see the effect of the condensing temperature and heat exchange temperature difference to the efficiencies.



(a) (b)

Fig.4 Efficiencies of OTFS



(a) (b)

Fig.5 Efficiencies of OTFS
From top to bottom, (a): T_b=20, 25, 30, 35, 40 °C and (b): δT_e=0, 5, 10, 15 °C, respectively.

4. Performance Comparison

In order to evaluate the efficiency advantage of OTFS, the performance comparison with ORC is made. For the Rankine cycle, the power output per unit mass of the heat source can be written as:

$$W_R = AC_p(T_1 - \delta T_e)\left(1 - \sqrt{\frac{T_b}{T_1 - \delta T_e}}\right)^2$$

Thus, the increase in power output of OTFS is:

$$\Delta W = \frac{W_{TF} - W_R}{W_R} = \frac{\left[(T_1 - \delta T_e - T_b) - T_b \ln\frac{T_1 - \delta T_e}{T_b}\right]}{(T_1 - \delta T_e)\left(1 - \sqrt{\frac{T_b}{T_1 - \delta T_e}}\right)^2} - 1 \qquad (11)$$

The increase in efficiency of OTFS is:

$$\Delta\eta_t = \frac{\eta_{tTF} - \eta_{tR}}{\eta_{tR}} \quad \text{and} \quad \Delta\eta_e = \frac{\eta_{eTF} - \eta_{eR}}{\eta_{eR}}$$

It is obvious that the relations between increases in power output and efficiencies are:

$$\Delta\eta_t = \Delta\eta_e = \Delta W \qquad (12)$$

Fig.6 shows the increase in power output of OTFS as a function of the heat source temperature. It is observed that the performance advantage of the system increases with decreasing heat source temperature.

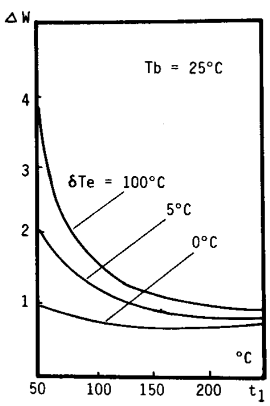


Fig.6 Increase in Power Output of OTFS

In general, the exergy efficiency of the organic Rankine cycle and supercritical cycle for low temperature heat source is about 30-40% and 40-50% respectively. In this case, the exergy efficiency of OTFS may reach to 60-80%.

To date, the efficiency of the total flow expander is still lower than that of steam turbines[2]. But, as analysed above, the OTFS has the potential for economically generating electric power from geothermal and waste heat source at low temperature than would be practical using Rankine cycle. It is interesting to calculate the minimun efficiency which the expander of OTFS must have to compete with ORC for generating the same power output and from the same conditions of the resources as:

$$\eta_{TF} = \frac{(T_1 - \delta T_e)\left(1 - \sqrt{\frac{T_b}{T_1 - \delta T_e}}\right)^2 \eta_{ST}}{(T_1 - \delta T_e - T_b) - T_b \ln\frac{T_1 - \delta T_e}{T_b}} \qquad (13)$$

Fig.7 presents the mimimun total flow expander efficiency as function of the heat source temperature. It is concluded that OTFS will have the efficiency advantage if the efficiency of the expander will be higher than that as indicated by minimun efficiency curve in Fig.7 at a given heat source temperature.

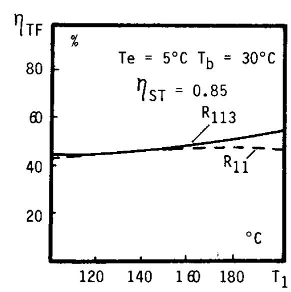


Fig.7 Min. efficiencies of T.F.expander

At the present time, the total flow expansion machines are in the development stage. Several expander concepts have been suggested. Among all candidate machines, the helical screw expander(HSE) may be a practicable one and suitable for use as a condensing machine in organic fluid case. Test results indicate that 60 to 70% isentropic(or 50 to 60% effect.) efficiency of the expander are achivable[3][4]. Fig.8 shows the performance results of a HSE, tested in laboratory of our university. The effect. efficiency of the expander is about 55%[5].(In Fig.8, η_i, η_e are the theoretical isentropic and experimental effective efficiencies respectively)

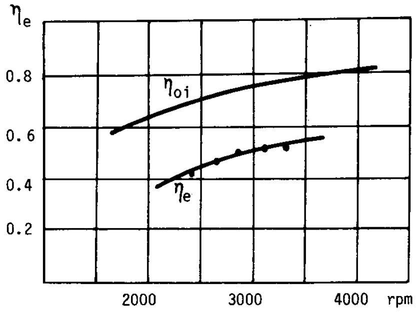


Fig.8 HSE Efficiency

It is of value to estimate the practical performance advantage of OTFS. For sake of illustration, we assume that the isetropic efficiency of Rankine cycle turbine and total flow expander is 80% and 60% respectively, the working fluid is Freon-11, mechanical and generator efficiency is 90%, the condensing temperature is 22 °C, heat exchange temperature difference is 5 °C for OTFS and 7 °C for ORC and the power output is 500kw. Fig.9 summarizes the results of calculations. The diameter of the turbine and expander is 977 and 510mm respectively.

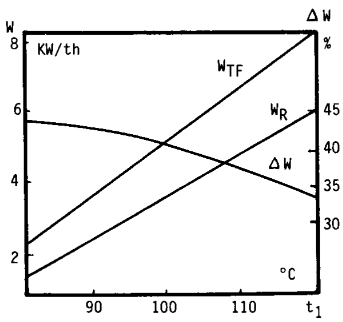


Fig.9 Performance Comparison of OTFS and ORC

5. Conclusion

The organic total flow system is an advanced energy conversion technology. It may provide 30 to 60% more efficiency from low temperature geothermal and waste heat sources than Rankine cycle. The efficiency advantage increases with dereasing the heat source temperature. The economics should also be proven attractive. Thus, such system offering significant energy and economic advantages should have good prespects.

References

(1) Austin,A.L., Higgins,G.H, Howard,J.H, 1973. The Total Flow Concept for Recovery of Energy from Geothermal Hot Deposits, UCRL-51366.

(2) Hu Liangguang, Gu Chujun, etal., 1987. A Comparison of Energy Use efficiency of Total Flow and Flashed Steam Methods for Geothermal Power Generation, International Symposium on Geothermal Energy.

(3) Mckay,L.S., Sprankel,R.S., 1974. Helical Rotary Screw Expander Power System, Proceedings on Research for the Development of Geothermal Energy Resources.

(4) Taniguchi,H., 1984. Test and Performance Analysis of Two Phase Flow Helical Screw Expander, Symposium on Thermodynamic Properties of Working Fluids for Banary Cycle(Japanese).

(5) Xue Huijiun, 1989. Theoretical and Experimental Studies on Characteristics of Two Phase Flow Helical Screw Expander, Master Degree Thesis of Tianjin University.