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THERMODYNAMIC PRINCIPLES OF TOTAL FLOW POWER PRODUCTION FROM HOT-WATER GEOTHERMAL RESOURCES

Gu, Chujun

Lu, Caren

Thermal Energy Research institute, Tianjin University, Tianjin, China

ABSTRACT

This paper analyzes the energy utilization ratio of total flow power production for hot-water geothermal resources and develops its differential expansion process. The temperature effect and dryness effect concepts for geothermal total flow power production have been presented. The paper also studies the evaporating and condensing effects of the expansion process and their specifications.

Results of the study show that the ideal total flow power production system has the more potential for achieving of the energy utilization ratio from the geothermal resources, but its practice application depends on whether the higher internal efficiency of the total flow engine, will be achieved or not.

INTRODUCTION

To date, the recovery and conversion of geothermal energy in hot-water deposits is accomplished by using either the flashed steam or the binary methods. The disadvantages of these two methods are mainly the mechanical and thermodynamic complexity and lower efficiency. In order to have the potential for achieving higher thermal efficiency and mechanical and thermodynamic simplicity, a total flow method has been proposed by the Lawrence Livermore Laboratory (LLL) of the United States. The important advantage is that since the expansion process of the entire geothermal fluid can be accomplished in the engine, the more perfect utilization of geothermal energy in hot-water resources will be possible.

In this paper a maiden attempt to analyse the energy utilization ratio of total flow power production for steam-water geothermal resources is made and its other thermodynamic characteristics are also studied.

THE EXPANSION PROCESS OF TOTAL FLOW POWER PRODUCTION SYSTEM

The total flow power generation system appears very simple. It can be illustrated in Fig.1, Fig.2 shows the thermodynamic cycle of the system.

We consider a total flow thermodynamic system for the steam-water mixture and assume that G, D_1 are the initial weights of liquid and vapor in mixture, T_1 is the initial

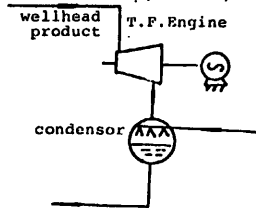


Figure 1

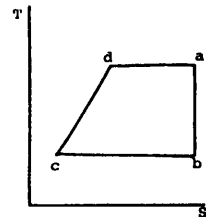


Figure 2

temperature of mixture, D is the weight of vapor evaporated in the expansion process. T is the temperature of mixture after expansion, $D, G-D$ are the weights of vapor and liquid in mixture respectively at a temperature of T . Considering a differential expansion process and temperature varies from T to $T-dT$, from the energy equilibrium relation the power output produced is

$$\begin{aligned} \delta W = & (G-D)[H_1(T) - H_1(T-dT)] + (D_1+D)[H_g(T) \\ & - H_g(T-dT)] - dD[H_g(T-dT) - H_l(T-dT)] \quad (1) \end{aligned}$$

where $H_1(T)$, $H_g(T)$ are the enthalpies of saturated liquid and vapor. dD is the increase in weight of vapor.

Assuming the expansion is ideal isentropic expansion, according to the entropy equation, we have:

$$\begin{aligned} (G-D)S_l(T) + (D_1+D)S_g(T) \\ = (G-D-dD)S_l(T-dT) + (D_1+D+dD)S_g(T-dT) \quad (2) \end{aligned}$$

where: $S_l(T)$ and $S_g(T)$ are the entropies of saturated liquid and vapor at temperature of T .

Transforming the forms of Eq(2) and considering:

$$(dS_l/dT)dT = S_l(T) - S_l(T-dT) \quad (3)$$

$$(dS_g/dT)dT = S_g(T) - S_g(T-dT) \quad (4)$$

we will have:

$$dD = \frac{(G-D)\frac{dS_l}{dT}dT + (D_1+D)\frac{dS_g}{dT}dT}{S_g(T) - S_l(T)} \quad (5)$$

Similarly to (3) and (4), using enthalpy

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differential formula to replace enthalpy difference, we can write:

$$\delta W = (G-D) \frac{dH_1}{dT} dT + (D_1+D) \frac{dH_g}{dT} dT - [H_g(T) - H_1(T)] dD \quad (6)$$

Introducing (5) into Eq.(6) and considering

$$H_g(T) - H_1(T) = T[S_g(T) - S_1(T)] \quad (7)$$

$$\begin{aligned} \delta W = & (G-D) \frac{dH_1}{dT} dT + (D_1+D) \frac{dH_g}{dT} dT \\ & - T \left[(G-D) \frac{dS_1}{dT} dT + (D_1+D) \frac{dS_g}{dT} dT \right] \quad (8) \end{aligned}$$

Making a simple differential transform:

$$T(dS_1/dT) = d(TS_1)/dT - S_1 \quad (9)$$

$$T(dS_g/dT) = d(TS_g)/dT - S_g \quad (10)$$

and considering the free enthalpy expression of saturated liquid and vapor:

$$G_1 = H_1 - TS_1 \quad (11)$$

$$G_g = H_g - TS_g \quad (12)$$

we can write:

$$\begin{aligned} \delta W = & (G-D) \frac{dG_1}{dT} dT + (D_1+D) \frac{dG_g}{dT} dT \\ & + (G-D) S_1 dT + (D_1+D) S_g dT \quad (13) \end{aligned}$$

Dividing formula (13) by $G+D_1$, $\delta W/(G+D_1)$ is differential work (ΔW) done per unit mass of water-steam mixture as

$$\Delta W = (1-x) \frac{dG_1}{dT} dT + x \frac{dG_g}{dT} dT + (1-x) S_1 dT + x S_g dT \quad (14)$$

where x is the dryness of mixture at temperature of T . In case of liquid-vapor equilibrium, we have $G_1 = G_g$, hence: $dG_1/dT = dG_g/dT$ and

$$\Delta W = dG_1 + S_1 dT + x(S_g - S_1) dT \quad (15)$$

In the isentropic expansion process, dryness is

$$x(T) = \frac{S_1 - S_1(T)}{S_g(T) - S_1(T)} \quad (16)$$

Where S_1 is entropy per unit mass of water-steam mixture at initial expansion state.

Introducing (16) into (15), we will have

$$\Delta W = dG_1 + S_1 dT \quad (17)$$

$$\text{Since } dG_1 = dH_1 - d(TS_1) \quad (18)$$

$$\text{Hence } \Delta W = dH_1 + d[T(S_1 - S_1)] \quad (19)$$

$$\text{i.e. } \Delta W = d[H_1 + T(S_1 - S_1)] \quad (20)$$

This differential process is correct during the overall process of the isentropic expansion.

Obviously, the differential work ΔW of this differential expansion can be divided into two parts. One is dH_1 , that which represents the decrease in enthalpy of saturated water. The other part is $d[T(S_1 - S_1)]$ and represents the differential area Λ as showing on the Fig.3.

ENERGY UTILIZATION OF TOTAL FLOW POWER PRODUCTION SYSTEM

We assume T_2 is the condensation temperature of total flow system. Integrating eq. (20) from T_2 to T_1 , the expansion work of the system for steam-water mixture is then

$$W_T = H_1(T_1) + T_1[S_1 - S_1] - T_2 S_1 - [H_1(T_2) - T_2 S_1(T_2)] \quad (21)$$

where

$$\begin{aligned} S_1 = & (1-x_1) S_1(T_1) + x_1 S_g(T_1) \\ = & S_1(T_1) + x_1/T_1 [H_g(T_1) - H_1(T_1)] \quad (22) \end{aligned}$$

here x_1 is the dryness of the mixture at initial state of the process.

The total enthalpy of the mixture of initial state is

$$H_1 = (1-x_1) H_1(T_1) + x_1 H_g(T_1) \quad (23)$$

Thus introducing eq. (22) and (23) into (21), we will have

$$\begin{aligned} W_T = & x_1 [H_g - H_1(T_1)] - T_2 [S_g(T_1) - S_1(T_1)] \\ & + [H_1(T_1) - T_2 S_1(T_1)] - [H_1(T_2) - T_2 S_1(T_2)] \quad (24) \end{aligned}$$

It is obvious that the power output per unit mass of mixture of the system is a linear function of dryness of the heat source.

From eq. (24) we have

$$W_T = (H_1 - T_2 S_1) - [H_1(T_2) - T_2 S_1(T_2)] \quad (25)$$

If the final temperature of the process were equal to the ambient temperature T_0 , the power output per unit mass of mixture would be maximized

$$W_{Tmax} = (H_1 - T_0 S_1) - [H_1(T_0) - T_0 S_1(T_0)] \quad (26)$$

From eq. (26) we find that the work done by the system is exactly equal to the exergy of the mixture, when $T_2 = T_0$.

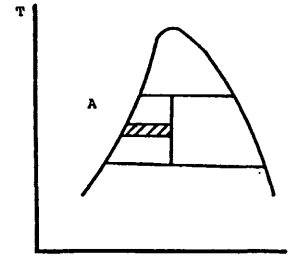


Figure 3

If the specific heat at a constant pressure of water C_p is const, then

$$S_1(T_1) - S_1(T_2) = AC_p \ln \frac{T_1}{T_2} \quad (27)$$

$$H_1(T_1) - H_1(T_2) = AC_p(T_1 - T_2) \quad (28)$$

where A is a conversion coefficient.

Introducing eq. (27), (28) into Eq(25), we have:

$$W_T = X_1 [H_g(T_1) - H_1(T_1)] \left[\frac{T_1}{T_2} - 1 \right] + AC_p [T_1 - T_2 - T_2 \ln T_1 / T_2] \quad (29)$$

we define

$$\Delta T_f = \frac{X_1 [H_g(T_1) - H_1(T_1)]}{AC_p} \quad (30)$$

$$T_f = T_1 + \Delta T_f \quad (31)$$

ΔT_f is called equivalent temperature difference of hot water of water-steam mixture source. T_f is called equivalent temperature of hot water of the mixture source.

Introducing ΔT_f , T_f to the Eq (29) we have

$$W_T = AC_p [T_f - T_2 - T_2 (\ln \frac{T_1}{T_2} + \frac{\Delta T_f}{T_1})] \quad (32)$$

when $T_2 = T_0$, we have:

$$W_{Tmax} = AC_p [T_f - T_0 - T_0 (\ln \frac{T_1}{T_0} + \frac{\Delta T_f}{T_1})] \quad (33)$$

The thermal efficiency off total flow system for water-steam mixture as:

$$\eta_T = W_T / Q_0 \quad (34)$$

Where Q_0 is the overall released heat when an unit mass of the mixture is cooled to the water at ambient temperature.

$$Q_0 = X_1 [H_g(T_1) - H_1(T_1)] + H_1(T_1) - H_1(T_0) = AC_p (T_f - T_0) \quad (35)$$

Thus the thermal efficiency of total flow power production is then:

$$\eta_T = \frac{(H_1 - T_2 S_1) - [H_1(T_2) - T_2 S_1(T_2)]}{X_1 [H_g(T_1) - H_1(T_1)] + H_1(T_1) - H_1(T_0)} \quad (36)$$

or

$$\eta_T = [T_f - T_2 - T_2 (\ln T_1 / T_2 + \Delta T_f / T_1)] / (T_f - T_0) \quad (37)$$

When $T_2 = T_0$, the maximum η_T is obtained as:

$$\eta_{Tmax} = \frac{T_f - T_0 - T_0 (\ln T_1 / T_0 + \Delta T_f / T_1)}{T_f - T_0} \quad (38)$$

The exergy efficiency of total flow system is defined as:

$$\eta_{eT} = W_T / E_0 \quad (39)$$

E_0 is the exergy of mixture and can be calculated

$$E_0 = [T_f - T_0 - T_0 (\ln \frac{T_1}{T_0} + \frac{\Delta T_f}{T_1})] AC_p \quad (40)$$

Hence the exergy efficiency of the total flow system for mixture is

$$\eta_{eT} = \frac{(H_1 - T_2 S_1) - [H_1(T_2) - T_2 S_1(T_2)]}{(H_1 - T_0 S_1) - [H_1(T_0) - T_0 S_1(T_0)]} \quad (41)$$

or

$$\eta_{eT} = \frac{T_f - T_2 - T_2 (\ln T_1 / T_2 + \Delta T_f / T_1)}{T_f - T_0 - T_0 (\ln T_1 / T_0 + \Delta T_f / T_1)} \quad (42)$$

When $T_2 = T_0$, the maximum η_{eT} is obtained

$$\eta_{emax} = 1$$

Fig.4 shows the relation between the power output per unit mass of mixture and temperature and dryness of heat source. In the high dryness region, first W_T increases with temperature of heat source, after reaching maximum value, it begins to decrease with the temperature of heat source. In the low dryness region, the extreme maximum point do not exist. The relation between W_T and dryness of heat source is linear.

Fig.5 represents the relation between the thermal efficiency of total flow system for mixture and temperature, dryness of heat source. In the high dryness region, there is a maximal value point, but the temperature of heat source at this point is generally higher than that of the point of maximum W_T . In the low region of dryness, the maximum value point disappears.

The power production value of total flow system is greatly influenced upon by the internal efficiency of total flow engine. The power output of practical total flow system is then

$$W_{TR} = \eta_0 W_T \quad (44)$$

η_0 is the internal efficiency of total flow engine.

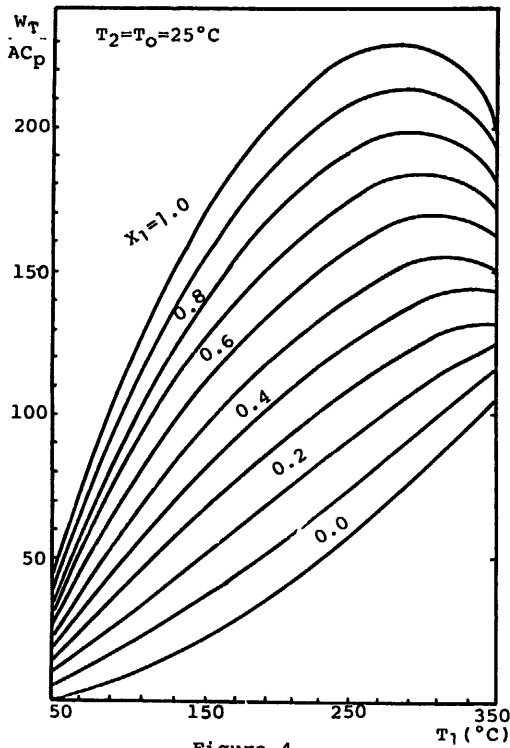


Figure 4

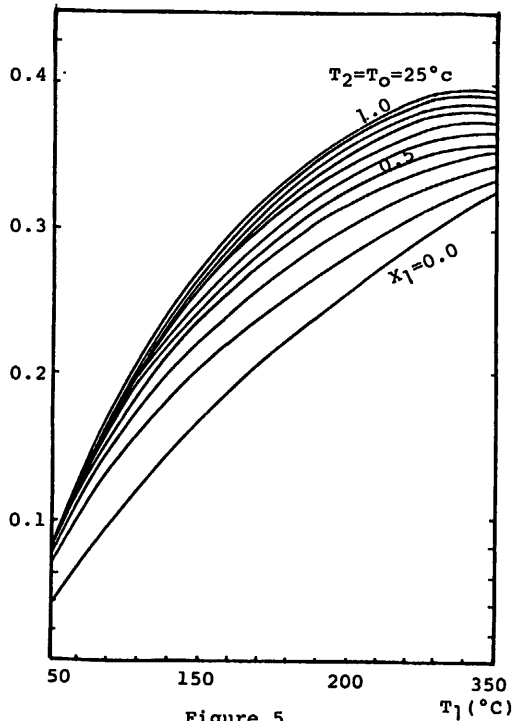


Figure 5

Introducing the above formula into Eq. (25), we have

$$W_{Tr} = \eta_0 [(H_1 - T_2 S_1) - (H_1(T_2) - T_2 S_1(T_2))] \quad (45)$$

The thermal efficiency of the practical system is

$$\eta_{Tr} = W_{Tr} / Q_0 = \eta_0 W_{Tr} / Q_0 = \eta_0 \eta_T \quad (46)$$

The exergy efficiency of the practical system is

$$\eta_{eTr} = \eta_0 \eta_{eT} \quad (47)$$

When the condensation temperature is an ambient temperature T_0 , we have

$$\eta_{eTrmax} = \eta_0 \quad (48)$$

This indicates that the thermodynamic characteristics of total flow system is strongly effected by the internal efficiency of total flow engine. Nevertheless, the internal efficiency of the total flow engine is generally very low at present, about 0.3 to 0.5 ordinarily. So the power output of practical total flow system is nearly 30-50% of idealized conditions.

Differentiating Eq. (32) relative to the condensation temperature T_2 , we will find the effect of the increase in condensation temperature of water on the power output of the total flow system

$$\frac{\partial W_{Tr}}{\partial T_2} = -AC_p \left(\ln \frac{T_1}{T_2} + \frac{\Delta T_f}{T_1} \right) < 0 \quad (49)$$

$$\frac{\partial^2 W_{Tr}}{\partial T_2^2} = AC_p / T_2 > 0 \quad (50)$$

Eq. (49) and (50) show that the power output would decrease with increase of condensation temperature, whereas the decreasing rate would decrease with increasing condensation temperature.

By the same way, we will have

$$\frac{\partial \eta_T}{\partial T_2} = - \frac{1}{(T_f - T_0)} \left(\ln \frac{T_1}{T_2} + \frac{\Delta T_f}{T_1} \right) < 0 \quad (51)$$

$$\frac{\partial^2 \eta_T}{\partial T_2^2} = 1 / [(T_f - T_2) T_2] > 0 \quad (52)$$

$$\frac{\partial \eta_{eT}}{\partial T_2} = \frac{1}{T_f - T_0 - T_0 \left(\ln \frac{T_1}{T_0} + \frac{\Delta T_f}{T_1} \right)} < 0 \quad (53)$$

$$\frac{\partial^2 \eta_{eT}}{\partial T_2^2} = \frac{1}{T_2 [T_f - T_0 - T_0 \left(\ln \frac{T_1}{T_0} + \frac{\Delta T_f}{T_1} \right)]} > 0 \quad (54)$$

From Eq. (51), (52), (53), (54) we know that the relations of η_T , η_{eT} and T_2 is similar to the relations of W_T and T_2 .

A TEMPERATURE EFFECT OF HEAT SOURCE OF TOTAL FLOW SYSTEM

The temperature level of heat source reflects its exergy value. In ordinary conditions, the higher the temperature of heat source is, the more the power is produced. But it is also related to the dryness of the mixture.

We define when the temperature of the heat source at a given dryness increases by 1°C, the increase in power output is called a temperature effect of the heat source of total flow system (HTTE) as

$$HTTE = (\partial W_T / \partial T_1)_{X_1 = \text{const}} \quad (55)$$

If $C_p = \text{const}$, we will have

$$HTTE = AC_p [(1 - T_2/T_1) (1 + \partial \Delta T_f / \partial T_1) + (T_2/T_1)^2 \Delta T_f] \quad (56)$$

When the temperature of heat source is relatively high, introducing eq. (25) into (55), we have

$$HTTE = X_1 [dLh(T_1)/dT_1] (1 - T_2/T_1) + X_1 Lh(T_1) T_2/T_1^2 + [dH_1(T_1)/dT_1] - T_2 [dS_1(T_1)/dT_1] \quad (57)$$

Where $Lh(T_1)$ is the latent heat of water at temperature of T_1 , i.e. $Lh(T_1) = Hg(T_1) - H_1(T_1)$.

Fig.6 and Fig.7 show the relation between the temperature effect of heat source and its temperature or dryness respectively. It should be pointed out that the relation between the temperature effect of heat source and its dryness is linear.

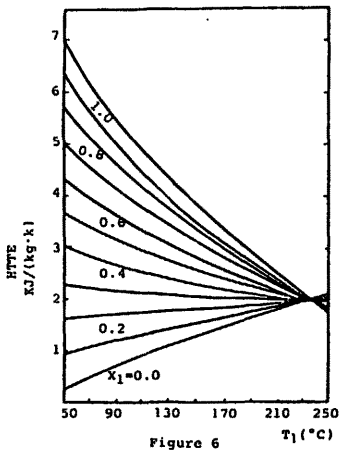


Figure 6

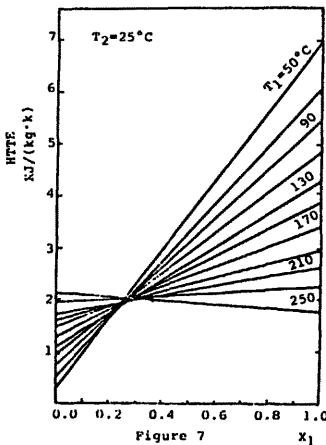


Figure 7

A DRYNESS EFFECT OF HEAT SOURCE OF TOTAL FLOW SYSTEM

As noted before, the power output is related not only to the temperature but also to the dryness of the heat source. So we define when

the dryness of heat source at a given temperature increases by a differential value, the increase in power of the system is called the dryness effect of heat source of total flow system (HTXE) as

$$HTXE = (\partial W_T / \partial X_1)_{T_1 = \text{const}} \quad (58)$$

Introducing eq. (29) into above formulæ, we have

$$HTXE = [H_g(T_1 - H_1(T_1))] (1 - T_2/T_1) = Lh(T_1) (1 - T_2/T_1) \quad (59)$$

HTXE is merely concerned with the temperature of heat source T_1 , condensation temperature T_2 and the latent heat of water Lh and independent on dryness of heat source X_1 . This conclusion is unanimously with linear relations between W_T and X_1 .

Fig.8 shows the relation between HTXE and the temperature of heat source.

It can be found that there is a maximal point on the curve. At the temperature of this point, (about 240°C) the increment per unit dryness devotes the most to W_T .

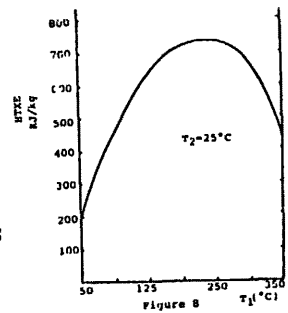


Figure 8

EVAPORATING AND CONDENSING EFFECTS OF TOTAL FLOW EXPANSION

For the total flow system, the expansion process is accompanied with the change of dryness whether it is idealized or practical. If the dryness increases during the expansion, this phenomenon is called the evaporating effect of total flow expansion. The phenomenon is called condensing effect, if the dryness decreases

The evaporating and condensing effect of the total flow expansion can be reflected by the difference between initial and final dryness as

$$EEC = X_2 - X_1 \quad (60)$$

$EEC > 0$, represents evaporating effect
 $EEC < 0$, represents condensing effect
 or

$$EEC = \frac{T_1 T_2 [S_1(T_1) - S_1(T_2)] + X_1 [T_2 Lh(T_1) - T_1 Lh(T_2)]}{T_1 Lh(T_2)} \quad (61)$$

For the case of $C_p = \text{const}$

$$EEC = \frac{X_1 T_1 T_2 \ln T_1 / T_2 + X_1 (T_2 \Delta T_{f1} - T_1 \Delta T_{f2})}{T_1 \Delta T_{f2}} \quad (62)$$

Where the definition of ΔT_{f1} is the same as former one. ΔT_{f2} is

$$\Delta T_{f2} = [X_1 Lh(T_2)/AC_p] \quad (63)$$

Fig.9 shows the relations of dryness X_2 to the different temperatures and drynesses of heat source and condition temperatures of the isetropic expansion. The tips of the curves represent the initial temperatures and drynesses and the any point§in curves represent the final temperatures and drynesses of the process. From the figure we can see that the dryness of the mixture is approximately unchanged throughout the proces when the dryness is equal to 0.5 or so. When the dryness is less than 0.4, the evaporating effect of idealized expansion is decreasing with the dryness. When the dryness is greater than 0.6, the condensing effect is increasing with the dryness.

Considering the efficiency of the total flow engine we will have

$$BEC = \frac{T_1(1-\eta_0)[H_1(T_1)-H_1(T_2)] + X_1[Lh(T_1)\{T_1(1-\eta_0) + T_2\eta_0\} - T_1 Lh(T_2)] + \eta_0 T_1 T_2 [S_1(T_1) - S_1(T_2)]}{T_1 Lh(T_2)} \quad (65)$$

For the case of $C_p = \text{const}$

$$BEC = \chi \frac{T_1(1-\eta_0)(T_1 - T_2) + \eta_0 T_1 T_2 \ln T_1 / T_2 + \Delta T_{f1} [T_1(1-\eta_0) + T_2\eta_0] - T_1 \Delta T_{f2}}{T_1 \Delta T_{f2}} \quad (66)$$

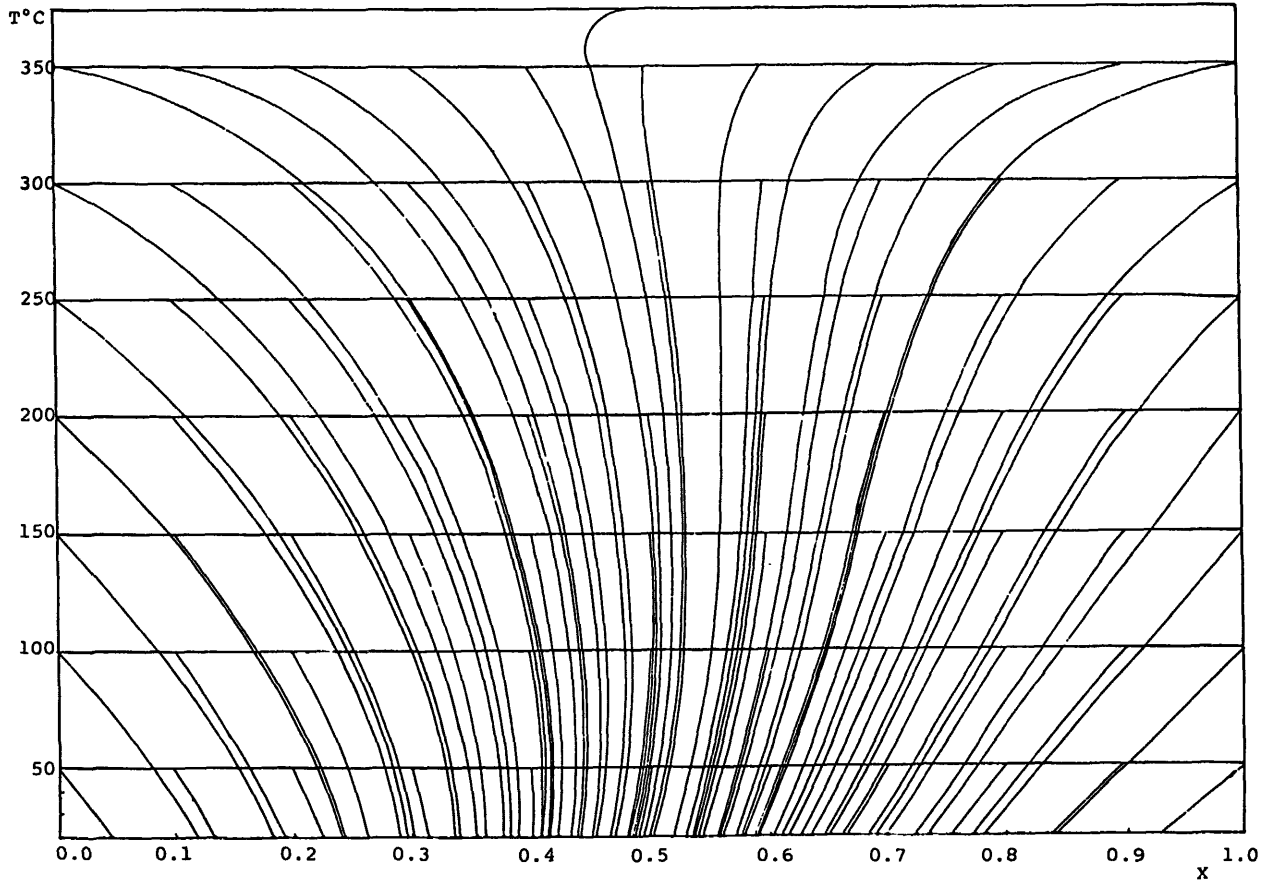


Figure 9

When $\eta_0=0$, the evaporating effect in the evaporating effect regions is the strongest.

When $\eta_0=0-1$, the evaporating effect is changing from the strongest to the weakest. In the condensing effect regions, $\eta_0=0$ means that the condensing effect is weakest. When $\eta_0=0-1$ the condensing effect is changing from the weakest to the strongest. It should be particularly emphasized that there exists a turning internal efficiency η_0 for any regions of condensing effect for water except when it is near regions of critical point and the temperature difference of heat source and condenser is not too small. When $\eta_0' < \eta_0 < 1$, condensing effect still appears. When $\eta_0 < \eta_0'$, condensing effect is turned into evaporating effect.

The evaporating and condensing effects of total flow expansion is very important for the researching work of multistage total flow power production system, the total flow-flashed steam and combined system.

From above-mentioned analysis we may summarize as follows:

1. In the idealized process of total flow expansion, when the initial dryness is near zero, the evaporating effect, must appear. The evaporating effect would have finally the decreasing tendency with the increase of initial dryness for the same final expansion temperature.

2. In the idealized process of expansion, when the initial dryness is near one, the condensing effect must emerge at least in some temperature regions (e.g. near critical region). The condensing effect would finally have the decreasing tendency with decreasing the initial dryness on the whole regions of condensing effect, for the some final expansion temperature.

3. When the efficiency of total flow expansion decreases, the evaporating effect in its regions is made stronger, and in the regions of condensing effect, the condensing effect is made weaker even changed from condensing effect into evaporating effect. But the condensing effect can not be turned into evaporating effect at least in some regions (e.g. near critical region) even if $\eta_0=0$.

These principles are generally correct for all kinds of fluids.

CONCLUSIONS

1. For the ideal isentropic expansion of total flow system the power produced is equal to the exergy of heat source, i.e. the exergy efficiency is 100%.

2. The internal efficiency of total flow engine influenced greatly on the thermodynamic characteristics of total flow system. It is still a vital research problem to raise the efficiency to total flow engine.

3. No matter how high the inlet temperature of heat source is, it is necessary to decrease backpressure as low as possible in order to decrease condensing temperature.

4. Although it is possible that the exergy

efficiency of total flow power production is very high, its available thermal efficiency is still low.

5. The temperature effect of heat source is mainly determined by thermodynamic properties of hot fluid in heat source. The concept of the temperature effect of heat source is significant for the economic analysis of total flow system.

6. The dryness effect of heat source of total flow system is completely determined by the temperature of heat source, condensing temperature and latent heat of heat-carried fluid.

7. The characteristics of evaporating and condensing effect of total flow expansion determines the evaporating and condensing phenomenon of total flow expansion. It is of significance to analyze multistage total flow power production and combined power systems.

REFERENCES

- Austin, A.L., House, P.A., 1978, new concepts for converting the energy in low-to medium-temperature liquids, with emphasis on geothermae Applications, UCRL-52583.
- Wang Zhian, Hu Liangguang, 1986, Performance Analysis of Liquid-Vapor Flow of Helical Rotor-Screw expanders, Engineering Thermodynamics Conference of Chinese Institute of Engineering Thermodynamics (Chinese).