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POWER GENERATION COSTS AND ULTIMATE THERMAL HYDRAULIC POWER LIMITS IN HYPOTHETICAL ADVANCED DESIGNS WITH NATURAL CIRCULATION

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INTRODUCTION

In assessing the design and operating power output of a plant, the real issue is the comparison to alternate power generation costs, or replacement costs. Obviously, the maximum power must be generated to minimize the cost per kWh, where the limit on the power output is usually set by the limit on the peak allowable fuel rating (or equivalent assembly power). The average core power output is set by the relationship between the peak to average power limit, and is a combination of the core and fuel assembly peaking factors. In most designs, the ultimate limit on power output is set by thermal hydraulic limits, rather than by plant thermal limitations of turbine output, heat exchanger rating, or fuel absolute temperature (i.e., center line melting) constraints. In actual operation, reductions in design margin in tube plugging, turbine efficiency and fuel burnup history may limit the output via reductions in plant availability, lifetime or capacity factor.

To reduce the peak power, various ingenious designs methods and incentives are used to flatten the core radial and axial power peaking factors, including the use of burnable poison distributions, and water and absorber rods to preferentially absorb neutrons. On line monitoring and control, optimized refueling strategies, and refined differential enrichments and pin designs are also used for the same reason. Similarly, various physical design strategies are used to reduce the peak rating or power density, including increased pin number, reduced diameter, increased length, and grid spacer optimization. In forced convection plants, the limits have usually been set by accident conditions, where there is a direct relationship between the allowable operating power rating and the maximum allowable peak rod or assembly power as determined by maximum allowable temperatures. Also, in transients and normal operation, the limits set by conventional thermal hydraulic CHF or DNB margins are commonly adopted.

In natural convection plants, there are opportunities for the limits to be set by the absolute power output available from natural convective flow, and the onset of instability in that flow. We are interested in the ultimate or maximum power output in order to both minimize power generation costs (both capital and operating), and to decide or determine how far the natural circulation designs can be developed we

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call this a hypothetical design, to indicate the conceptual nature of the analysis.

To illustrate the approach, simple analytical expressions are derived to illustrate the ultimate or maximum heat removal. We can then relate the maximum thermal hydraulic limits to hypothetical reactor power output and cost. The relationship between some of the various enhanced design features is then clear when seeking the ultimate or maximum safe power output at least cost. A hypothetical natural circulation design with boiling is discussed as a basis for competitive generation costs.

NATURAL CIRCULATION POWER LIMITS

In the following, we assume that the measures to flatten the core power profile and reduce peaking have all been employed. Thus peaking has been minimized through enrichment, fuel cycle, power shape, and fuel design innovations and methods specific to a particular fuel bundle design. The geometrically related thermal hydraulic parameters (inlet orificing, outlet loss, and friction and form losses) are however known or variable for a particular fuel design or bundle configuration.

It is important to note that stability limits in natural circulation systems arise before (and as a prelude to) CHF or DNB. Indeed, conventional forced flow CHF and DNB correlations cannot be applied to natural circulation and parallel channel systems if either the loss coefficients are unknown or not reported, or the appropriate constant pressure drop was not maintained or achieved in the tests. Throttling the inlet flow to set a flow boundary condition artificially stabilizes the channel. In actual plants, it is well known that the plant maintains a constant pressure drop, by having multiple parallel channels and/or a controlled downcomer hydrostatic head.

(a) Pressurized systems

The common designs of pressurized systems limit the heat removal to that determined when there is no bulk boiling. The flow is always subcooled, and the heat exchange is by single phase (liquid) flow in the heat exchanger. Examples of this design are PWRs, VVERs and HWPRs, of the typical forced convection design with loop pumps, and the so-called advanced designs of the AP600 and EPR. Heat removal in normal and accident conditions is set by the convective heat removal by natural circulation.

We would like to establish the maximum or ultimate heat removal in a natural circulation design where there is an elevation difference between the heat source (core) and sink (HX). For the present purposes it does not matter if the HX is a vertical or horizontal design. The maximum power limit is when the heat generated is completely removed by the heat exchanger loop, and the HX outlet temperature is close or equal to the secondary side (boiling) temperature. The secondary side temperature is of course set by the turbine stop valve (design) pressure. Thus, there is a relation between the core maximum (saturation or subcooled) temperature, and hence the power, and the secondary pressure.

DISCLAIMER

For the purposes of the maximum design output evaluation, we define the maximum core outlet temperature as the saturation temperature at the primary pressure. Thus the ultimate limit is taken as bulk boiling at the core exit and not conventional DNB. We consider the two phase (boiling) limit later and use results available from the literature (Duffey and Sursock, 1987).

For this *single phase case*, the natural circulation flow rate is found by integration of the loop momentum equation and coupling this to the energy equation. Thus, the primary temperature increase is given by the standard result,

$$\Delta T = \left(\frac{Q_c}{\rho_l c_l} \right)^{2/3} \left(\frac{K_f}{2g\beta\Delta L} \right)^{1/3} \qquad (1)$$

where the fluid thermal expansion drives the convective flow around the loop. Also, from the heat balance across the HX, we have, assuming a mean temperature as a good approximation to the average of the non-linear profile,

$$\begin{aligned} Q_c &= \epsilon A_H n_t \langle h \rangle (T_p - T_S)/2 \\ &\approx \epsilon A_H n_t \left(\frac{K_t}{\delta_t} \right) (T_p - T_S)/2 \end{aligned} \qquad (2)$$

Here, we have taken the primary to secondary temperature difference as close to that due to conduction across the tube wall only, with a correction coefficient for any surface, plugging, corrosion, and thermal resistance effects. Now, to maximize the heat removal we assume the HX to not be limiting in capacity, and hence may take the core outlet temperature as saturation, and the inlet temperature as close to the HX secondary temperature.

Equating the two expressions for the power [Equations (1) and (2)], and noting the steam is saturated on the secondary side (superheat can be accounted for if needed), we obtain the following result for the maximum heat removal in a natural convection loop where onset of bulk boiling at the core exit is the limit,

$$Q_M = \left(\frac{\epsilon A_h n_t K_t}{2\delta_t} \right)^3 \left(\frac{K_f}{2g\beta\rho_l^2 c_l^2 \Delta L} \right) \qquad (3)$$

It is immediately evident that the loop design parameters of elevation and losses are important, but even more sensitive are the HX characteristics of material and wall thickness (the effective heat transfer coefficient). Now, recasting this result in terms of the primary to secondary pressure drop, via the Clausius-Clapeyron relation, we have the result for the maximum core power as,

$$\begin{aligned} Q_m &= \epsilon A_H n_t \langle h \rangle \; (T_p - T_S)/2 \\ &= \left(\frac{\epsilon A_H n_t K_t G \bar{T}_{sat}^2}{\delta_t h_{fg}} \right) \left(\frac{P_p - P_S}{P_p + P_S} \right) \end{aligned} \qquad (4)$$

These results are somewhat counter intuitive for several reasons. The maximum heat removal is very sensitive to the HX design, and relies on maximizing the primary to secondary temperature drop, and hence on minimizing the core to HX elevation difference, and also maximizing the loop flow resistance.

We can also derive the equivalent expression for the primary to secondary pressure ratio, which we write as P^*, and is from Equations (3) and (4),

$$P^* = \left(\frac{P_P - P_S}{P_P + P_S}\right) = \left(\frac{\epsilon A_H n_t K_t}{2\rho_f c_f \delta_t}\right)^2 \left(\frac{h_{fg} K_f}{4g\beta G \overline{T}_s^2 \Delta L}\right) \quad (5)$$

Note that decreasing the elevation difference increases the pressure ratio, and that again, the effects of the HX design parameters are non-linear.

(b) Boiling systems

The other common design is when there is two-phase boiling flow, and usually no HX to minimize the cost or to use direct steam to the turbine. Examples of these designs are the BWR, ABWR, and SBWR. When steam is fed directly to the turbine, again there is a direct relation of the maximum power output to the secondary (turbine stop valve) pressure.

Without a HX to condense the steam the maximum heat removal is due to the circulation of two-phase mixture when the downcomer is liquid and the core is two-phase. The heat removal is then totally evaporative and given by,

$$Q_m \approx H_{fg} W_m \approx \left(\frac{g\rho_f \Delta L}{C_O \rho_g}\right)^{1/2} h_{fg} \rho_f A_c \quad (6)$$

This result shows clearly the important influence of the downcomer elevation and the core area, which helps to maximize the heat removal for this case.

We note that the maximum power output in a natural circulation boiling system without a HX is *not* that derived on the basis when the natural circulation driving head is equal to the two-phase losses, i.e.,

$$Q_m \neq h_{fg} W_O \approx h_{fg} \left(\frac{2A_c^2 g \rho_l \rho_g \Delta z_D}{k_f}\right)^{1/2} \quad (7)$$

Instead, the ultimate or maximum power output is set by the onset of flow instability and hence subsequent CHF. The expression for the stability condition is derived by solving for the minimum in the pressure drop versus flow relation (Rohatgi and Duffey, 1994; 1996). The resulting equation is then solved simultaneously with the natural circulation loop flow equation for the intersection of the stability and flow solutions for the stable limit. The form of the natural circulation line is found to be $N_p/N_s \sim$ *constant* for a given downcomer head to core height ratio, L*. The limiting region of stability is given to a good approximation by the solution of the quadratic,

$$N_p^2 N_{fr}(1 + k_e) + 2N_p N_{fr} \left[(1 - N_s)(2 + k_e) + k_i\right] + 3N_{fr} N_s^2 + 2N_f(2 + N_s) = 0 \quad (8)$$

Limiting maximum power solutions for the unstable case are of order,

$$N_p/N_s \sim 3 \quad (9a)$$

with a residual dependency on the loss coefficients, consistent with subcooled flow instability data. When there is a natural circulation loop, then the intersection of the stability region with the natural circulation flow is very nearly, for typical design values, given by,

$$N_p/N_s \sim 2 \qquad \textbf{(9b)}$$

Thus, we can write generally that the maximum power is a function of, say, the form,

$$N_p/N_s = N* \qquad \textbf{(9c)}$$

We find that by comparing a wide range of parallel or multichannel instability data at high pressure (5MPa) on a N_p versus N_s plot, that the data do indeed group around a line given by N*= 3 (Rohatgi and Duffey, 1996).

Now the maximum two-phase flow in the whole natural circulation loop at intermediate inventories, is given by:

$$W_O \approx \left(\frac{A_c^2 \rho_l^3 g \Delta L}{k_f C_O \rho_g h_{fg}} \right)^{1/3} Q_c^{1/3} \qquad \textbf{(10)}$$

Combining the Equations (9) and (10) for the maximum flow rate at the instability limiting case, we find the hypothetical maximum core power at this maximum flow rate to be,

$$Q_m \approx \left(\frac{g \Delta L A_c^2 \rho_l^3 c_p^3 \Delta T^3}{k_f C_O \rho_g h_{fg}} \right)^{1/2} \left(\frac{2}{2+k_e} \right)^{3/2} \qquad \textbf{(11)}$$

This result clearly shows the influence of both the total loop and peak power bundle outlet loss coefficients in determining the maximum core power, the former by limiting the maximum flow in the loop and the latter the maximum stable power. To optimize the design output, the minimum loss coefficients, and the maximum elevation (driving) head and flow area should be obtained.

SCALING THE DESIGN

The above derivations imply some interesting effects of scale (i.e., maximum power output) if we for the moment set aside the purely physical properties . Thus, for the single phase case, the maximum power scales as

$$Q_m \propto \left(\frac{A_h n_t K_t}{\delta_t} \right)^3 \left(\frac{k_f}{\Delta L} \right) \qquad \textbf{(12a)}$$

whereas, for the boiling case,

$$Q_m \propto A_c \left(\frac{\Delta L \Delta T_m^3}{k_f \rho_g k_e^3} \right)^{1/2} \qquad \textbf{(12b)}$$

Now generally, the cost to construct the plant will increase with the volume of materials used, including concrete, steel and pipe work. We can take the *relative* cost of the reactor, C, as scaling with the characteristic loop dimensions such that,

$$C \propto A \Delta L \qquad \textbf{(13)}$$

So, for an estimate of the *relative* cost changes, C/Q_m, we see these vary as,

$$\frac{C_s}{Q_m} \propto \left[\frac{\Delta L}{A_h} \right]^2 \qquad \textbf{(14)}$$

for the single-phase system, where the dimension is the elevation of the heat sink, and as

$$\frac{C_B}{Q_m} \propto [\Delta L]^{1/2} \qquad (15)$$

for the boiling system, where the dimension is that of the loop. Thus, the physical dimensions of the chosen system for these hypothetical designs directly influences the cost per unit power output, but in fundamentally different ways.

THE GENERATING COST EQUATION

We are now in a position to write down the generating cost equations for these maximum hypothetical designs. Given the usual relationships between peak and average power, we have the relation between limiting or maximum core power and the electrical output cost of the usual functional form,

$$C_G = F\left(\frac{\sum_n C_n}{A\eta Q_m}\right) \qquad (16)$$

Here, for simplicity, the annual average capacity factor, A, is the fraction of time producing the rated output per year, and includes all downtime (scheduled or not); the thermal to electrical conversion efficiency is η, and includes all the turbine and generator losses; and the numerator is the sum of all generating costs (capital, operating, fuel, and maintenance) in a given year, weighted by merit and other factors as needed. The least cost is clearly when the output is maximum, and substitution from Equations (3), (6) or (11) for example shows how the design parameters directly affect the cost, without for the moment including secondary effects on the capital and engineering costs, such as materials.

Now to evaluate the design parameters in the cost equation, we need a target capacity factor. Purely as an example, the data given in the Table are all the average *national* capacity factor data up to 1991, for a population of 368 reactors in 16 countries (ANS., 1992). They represent actual operating experience, repairs, replacement, generating policy, national load variations and economic fluctuations, but there is a clear trend. We can see that the national capacity factor is of the form,

$$A(\%) = 83 - 12 \log N \qquad (17)$$

for a national population of N reactors. These data imply a single unit can achieve a lifetime average of 83%. The world average capacity factor over all countries and populations is 65% from these same data, and we note that the averages are over all reactor designs and types.

Reinforcing this observation and analysis, recent operating experience in the U.S. and France indicates that 80% is achievable. Thus, the data presented by Fertel (1996) show a recent *yearly average* of 78%, and many plants exceed this value in both countries. Therefore, we take a value of order 80% as the highest achievable and desirable for the hypothetical design. A natural circulation design that exceeded the national average in any country should be perceived as having a competitive advantage.

For the generation cost, we take the lowest competitive rate (usually a natural gas or co-generation value, but from any substitute or market power), which is a fixed target cost per kWh. Other approaches adopt a fixed cost advantage percentage as a target, say 10-20%. The data of Fertel (1996) indicates that between 2 to 3.5 c/kWh is the target range for competitive power generation in the U.S., without allowing for regional variations. Taking, as an example, the maximum power output for a boiling design limited by stability (Equation (11)), we can substitute this result in Equation (16) to evaluate the competitive cost for an assumed plant lifetime (or for the equivalent net generating hours). Typical target lifetimes are in excess of 30 or more years, with appropriate equipment maintenance and replacement.

Table 1 National Average Capacity Factors up to 1991

N (reactors)	Capacity Factor(%)	Country
112	62	USA
55	63	France
41	68	Japan
36	50	UK
31	67	Russia
20	75	Canada
20	73	Germany
11	71	Sweden
10	74	Spain
6	42	India
6	76	Czech/Slovak
5	82	Belgium
5	67	Taiwan
4	82	Switzerland
3	84	Hungary
3	83	Finland

HYPOTHETICAL DESIGNS

To provide a working estimate, we proceed to evaluate hypothetical designs with the maximum power output, and thus try to avoid any preference or commercial considerations for any current design. In the hypothetical designs, the number of "free" overall thermal hydraulic design parameters are quite small, and include the loop loss coefficient, the elevation difference between the core and the HX, the operating pressure (also governed by turbine considerations), and the core flow area. This calculation assumes water as the working fluid.

First evaluating the single-phase case, we insert typical values into Equation (4), assuming water/steam as the working fluid. The result for assumed parameters of a heat sink elevation of 10 meters, 10,000 tubes in the HX, and a loop loss coefficient of order 10, is a maximum power of order $Q_m \approx 50P^*$ MW. This implies that, for this case, the primary to secondary pressure ratio is indeed a sensitive design parameter.

Now for the boiling case, we can insert typical values into Equation (11). With assumed parameters of an elevation difference of 20 meters and a flow area of order $1m^2$, we obtain a 5000 MW(t) output. The value taken for N* in a natural circulation loop was 2 , which is also consistent with the lower bound from the data comparisons for boiling parallel channels at 5MPa. It would be prudent to check the range and values thoroughly with *multichannel* design data.

Assuming just 30% thermal to electric conversion efficiency, and a design life of

30 years at 80% capacity factor, the 1500 MW(e) plant revenue will be of order $9B, generating competitive power at under 3 c/kWh. Thus the total capital, O&M, fuel and interest charges must be less than this value.

Operating and safety margins to the maximum power can be factored in as necessary, and increase the incentive for longer operating lifetime and higher efficiency.

CONCLUSION

Maximum power limits for hypothetical designs of natural circulation plants can be described analytically. The thermal hydraulic design parameters are those which limit the flow, being the elevations, flow areas, and loss coefficients. We have found some simple "design" equations for natural circulation flow to power ratio, and for the stability limit.

The analysis of historical and available data for maximum capacity factor estimation shows 80% to be reasonable and achievable.

The least cost is obtained by optimizing *both* hypothetical plant performance for a given output, and the plant layout and design. There is also scope to increase output and reduce cost by considering design variations of primary and secondary pressure, and by optimizing component elevations and loss coefficients. The design limits for each system are set by stability and maximum flow considerations, which deserve close and careful evaluation.

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NOMENCLATURE

A	Flow area; Average load factor(%)
β	Liquid volume expansion coefficient
c	Specific heat
C	Cost
D	Hydraulic diameter
δ_t	Tube wall thickness
f	Friction factor
g	Acceleration due to gravity
G	Mass flux
h	Heat transfer coefficient
h_{fg}	Heat of vaporization

k	Loss coefficient, non-dimensional $\bar{k}/N_{fr}$
K_t	Thermal conductivity of HX tubes
L	Heated length scale
ΔL	Elevation difference
N	Number of reactors
N*	Unstable power to flow ratio, N_p N_s
N_f	Froude number, $gL\rho_l^2/G^2$
N_{fr}	Friction number, fL/2D
N_p	Phase change number, $Q\rho_l/AGh_{fg}\rho_g$
N_s	Subcooling number, $c_p\Delta T\rho_l/h_{fg}\rho_g$
n_t	Number of HX tubes
P	Pressure
Q	Power
ρ	Density
T	Temperature
ΔT	Subcooling
W	Mass flow rate
W_o	Maximum flow rate
Z	Vertical location
Z_D	Downcomer liquid level

Subscript:

a	average
c	core
e	exit
g	vapor
i	inlet
l	liquid
m	maximum
p	primary
s	secondary