Optimization of the Fuel Consumption of an Evaporation Salt Plant with the Aid of the Exergy Concept

P. Jongema
*Akzo Salt Chemicals Division,
Hengelo, Netherlands*

ABSTRACT

The ever increasing price of fuel in the last decade has been the stimulus to undertake a study of the factors that govern the fuel consumption of the vacuum salt-making process. By making use of the exergy concept, a method has been devised that provides an answer to the question of how a plant should be constructed in order to achieve an optimum fuel consumption. It has been found that at the 1982 price level, and independent of the number of effects, the optimum fuel consumption is 1.20 GJ per metric ton of salt, or 550 Btu (HHV) per lb of salt.

INTRODUCTION

Because of the steep rise of the fuel cost in the past decade as against a more gradual one of plant construction cost (see Figure 1) it may be expected, that the optimal energy consumption for the production of bulk chemicals e.g. like vacuum salt has shifted to a value much lower than 10 years ago. To know this optimum is not only of interest in those cases where a new plant is envisaged, but is also of importance to a comparison with the actual consumption of an existing one. Once this comparison has been made, an improvement may be initiated.

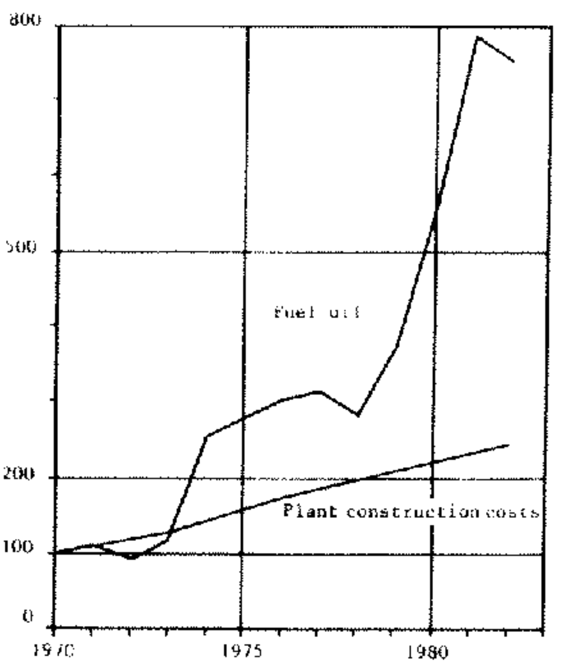

Figure 1. The rise of fuel oil and plant construction costs (in the Netherlands) vs time.

ENERGY CONSUMPTION OF THE VACUUM SALT PROCESS

As the process of making salt from brine does not occur spontaneously, work is required to carry it out. The minimum amount can be derived from the free energies of formation:

$$\begin{array}{lll} NaCl\ (c){:}\ \Delta F = & -91.894 & \text{kcal/mole (at 25°C).} \\ NaCl\ (aq){:}\quad\ = & \underline{-93.92} & \text{kcal/mole (at 25°C).} \\ \text{Ideal work}\quad = & 2.026 & \text{kcal/mole} = 40\ \text{kWh/t.s.} \end{array}$$

(tonne of salt or 1000 kg).

In a real process this figure is much higher, as can be seen from the following example of a quadruple effect evaporation plant (Figure 2) designed in the 1960s.

The heat released by the condensing steam is not lost, of course, but is transferred for over 85% to the cooling water, while the remainder is absorbed by the outgoing process streams (vapour condensates, mother liquor). However, by transferring the heat from steam to cooling water, work is extracted from it which is used to drive the process.

The maximum amount of work that can be obtained from the heat delivered by the steam is calculated with Carnot's well-known expression by assuming a reversible

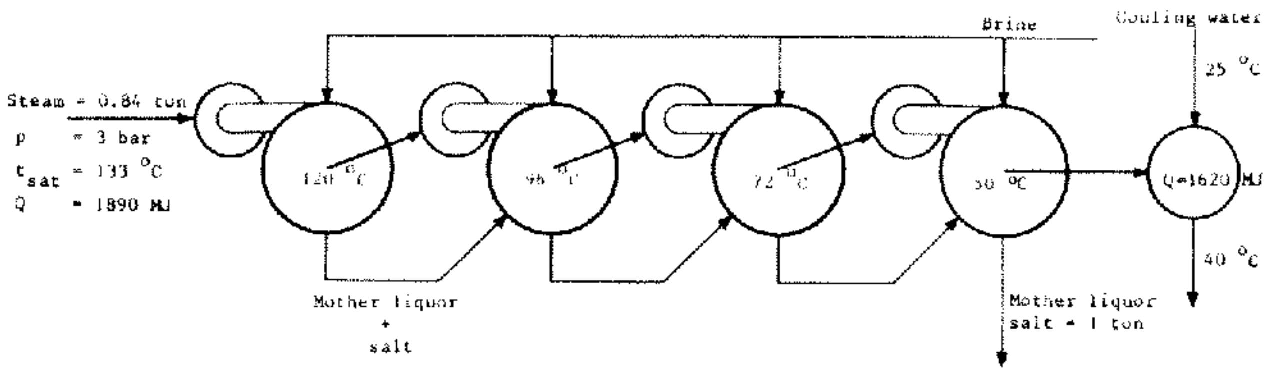

Figure 2. Simplified flowsheet of a quadruple effect evaporation plant.

heat engine operating between a source at 133°C and a sink at 25°C (Figure 3):

$$\text{Maximum work: } W = \left(1 - \frac{T_2}{T_1}\right)Q \qquad (1)$$

$$W = \left(1 - \frac{298}{406}\right)1890 = 503 \text{ MWs} (= \text{MJ})$$

per ton of salt = 140 kWh/t.s.

After adding about 20 kWh/t.s. for the shaft work done in th process (pumps, centrifuges), one arrives at a figure for the total work required to run the plant equal to about 160 kWh/t.s. So the thermodynamic efficiency of the process is only 40/160 = 25%.

THE EXERGY BALANCE

To analyse the cause of the rather low efficiency, use will be made of the exergy concept. As first defined by Rant (1956) the exergy E of a quantity of heat Q available at T equals the maximum amount of work that can be obtained by transferring it to the surroundings at a temperature of T_0. So by applying expression (1):

$$E = \left(1 - \frac{T_0}{T}\right)Q \qquad (2)$$

Exergy may, therefore, be considered the work potential of a quantity of heat. By valuing heat or the medium transferring it, like steam, at its exergy instead of at its number of Joules (or Btu's) or its enthalpy, it is brought on an equal footing with mechanical or electrical power. Generally spoken, exergy and shaft power are the same, they may be interchanged or added to each other.

Looking into more detail of the quadruple-effect plant presented above it is obvious that the surroundings are defined by the cooling water temperature, so $T_0 = 298$ K. The heat discharged to the cooling water (1620 MJ/t.s.) is transferred at a level of about 40°C, so the exergy of this heat is:

$$\left(1 - \frac{298}{273 + 40}\right)\frac{1620}{3.6} = 22 \text{ kWh/t.s.}$$

As has been calculated already, the exergy delivered by the condensing steam $E_D = 140$ kWh/t.s. However, the exergy of the same quantity of heat absorbed at 120°C by the 1st effect:

$$E_A = \left(1 - \frac{298}{393}\right)\frac{1890}{3.6} = 127 \text{ kWh/t.s.}$$

So the transfer of heat, made possible by a driving force of: 133 − 120 = 13°C is accompanied by an exergy loss:

$$\Delta E = E_D - E_A = 140 - 127 = 13 \text{ kWh/t.s.}$$

Moreover, in order to enhance the heat transfer, forced circulation is applied, which consumes about 2 kWh/t.s. of shaft power per effect.

Now it is a well-established fact, that the vapour generated in an evaporator is not quite in equilibrium with its contents. Depending on several factors the vapour is

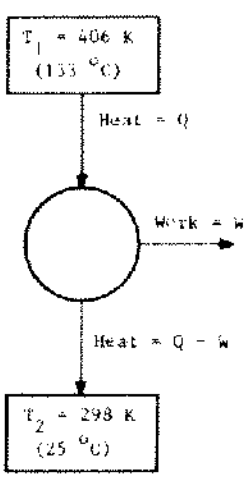

Figure 3. Reversible heat engine.

about 2°C lower in temperature. This gives rise to an additional loss of:

$$\Delta E = \left[\left(1 - \frac{298}{393}\right) - \left(1 - \frac{298}{393 - 2}\right)\right] \frac{1890}{3.6}$$
$$= 2 \text{ kWh/t.s.,}$$

which brings the total exergy loss and power use related to the heat transfer and boiling in the evaporators to 13 + 2 + 2 = 17 kWh/t.s. per effect, that is roughly 68 kWh/t.s. for 4 effects.

Summarizing, the actual work required to drive the process may be divided as follows:

Ideal work:	40 kWh/t.s.
Heat transfer and boiling (in evaporators):	68 kWh/t.s.
Exergy discharged to cooling water:	22 kWh/t.s.
Shaft power except circulators (20 − 4 × 2):	12 kWh/t.s.
Balance (= other exergy losses):	18 kWh/t.s.
	160 kWh/t.s.

So this "work and power balance", or, to use a better expression, this "exergy balance," clearly indicates that the major part of the losses occurring within the battery limits of the process are attributable to the evaporators. Therefore, prime attention should be directed to the design of this part of the plant.

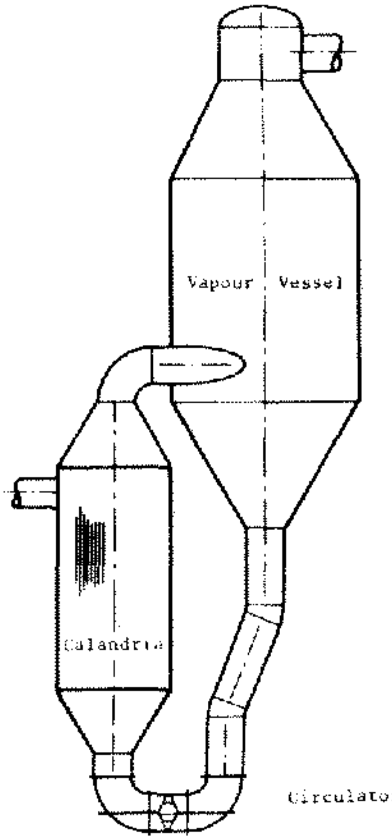

Figure 4. Evaporator.

THE OPTIMIZATION PROBLEM

A forced-circulation type of evaporator, such as the one in Figure 4, consists of 3 pieces of equipment:

1. a calandria
2. a circulator
3. a so-called "vapour vessel" (it provides room for both releasing the vapour and crystallizing the salt.)

Qualitatively it may be said, that if the size of the calandria is increased, the LMTD[1] for heat transfer will drop and so will the exergy loss. If the pump size is increased the Δt per pass[1] becomes less, the available LMTD increases slightly and, what is more important, the boiling equilibrium is approached better. On the other hand this reduction of the exergy loss must be paid for by an increase of shaft power. These examples illustrate again the generally known fact that an increase in investment cost, and consequently the annual fixed (= capital and maintenance) cost, is accompanied by a decrease in the variable cost. The problem now is to find the set of conditions where the total annual costs are at a minimum.

Assuming a certain heat load and fixing the diameter and length of the calandria tubes will allow deriving a number of exergy loss and power consumption functions with two variables only, by applying the appropriate chemical engineering relationships for the design of heat exchangers and pumps. For this purpose, the selection of both the tube side velocity and the heat flux as the two variables has been found to be a very convenient one.

Reference is made to the Appendix for a summary of the equations. They have been based on an evaporator temperature of 92°C. The reason for choosing this particular temperature is, that it is fairly representative of the average conditions in an evaporator train.

Scrutinizing cost data on calandrias and circulators of a great number of vacuum salt plants built over the past 25 years, has led to the formation of two investment cost functions expressed in the same two variables. The basics having been dealt with, only a price remains to be tagged to the exergy in order to solve the optimization problem.

THE PRICE OF ENERGY

The price to be attached to exergy is strongly dependent on the way it is made available to the user. A few numerical examples will make this clear. The simplest

[1]LMTD = log mean temperature difference. Δt per pass = temperature rise in the calandria.

[2]This is the pressure at the intersection of the turbine state line and the saturated steam line on a Mollier-chart.

manner of raising steam for the use in the afore-mentioned evaporation plant is doing so in a boiler working at a low pressure. A simplified flowsheet of such a cycle is given in Figure 5.

Saturated steam is generated in a steambloc at a pressure of, e.g., 10 bar. After throttling to the required pressure of 3.0 bar it is condensed in the salt plant. The condensate is returned uncooled. Heat and material balance data, based on a supply of 1 ton of steam have been included. Remembering Equation 2 the exergy delivered to the customer

$$E = \left(1 - \frac{298}{406}\right)\frac{2219}{3.6} = 164 \text{ kWh}.$$

Substracting the (small) use of power of the boiler plant itself, the net exergy production is 161 kWh at the cost of 2522 MJ of fuel. So the exergy efficiency of the LP-boiler plant is:

$$\frac{161 \times 3.6}{2522} = 23\%$$

By defining the exergy efficiency in this way it is presupposed that the exergy of primary fuels = 1 MWs/MJ = 1 kWh per 3.6 MJ Low Heating Value (LHV). This has been found to be more or less true. For a detailed argument, see Baehr (1979).

This rather low efficiency may be improved by applying co-generation of steam and power. An example of a cycle incorporating a high-pressure boiler and an extraction back-pressure turbine is presented in Figure 6. Again, the quantities have been based on a steam raising capacity of 1 ton. In this case the combined steam-power plant produces:

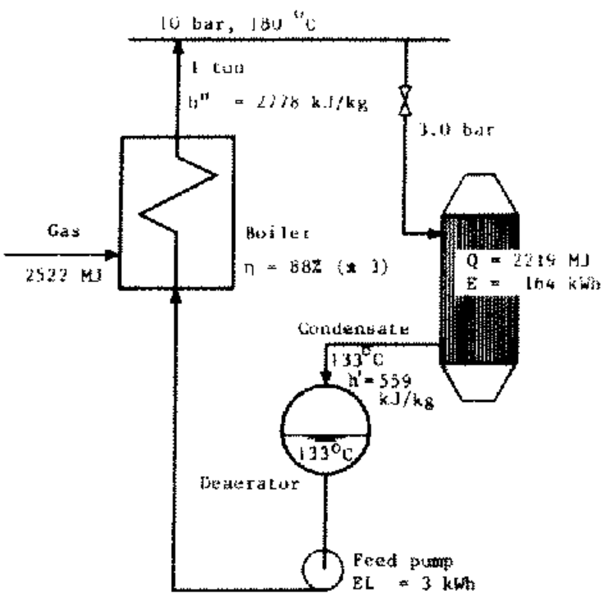

Figure 5. Steam supply with LP-boiler. Boiler efficiency (η = 88%) is based on the low heating value (LHV).

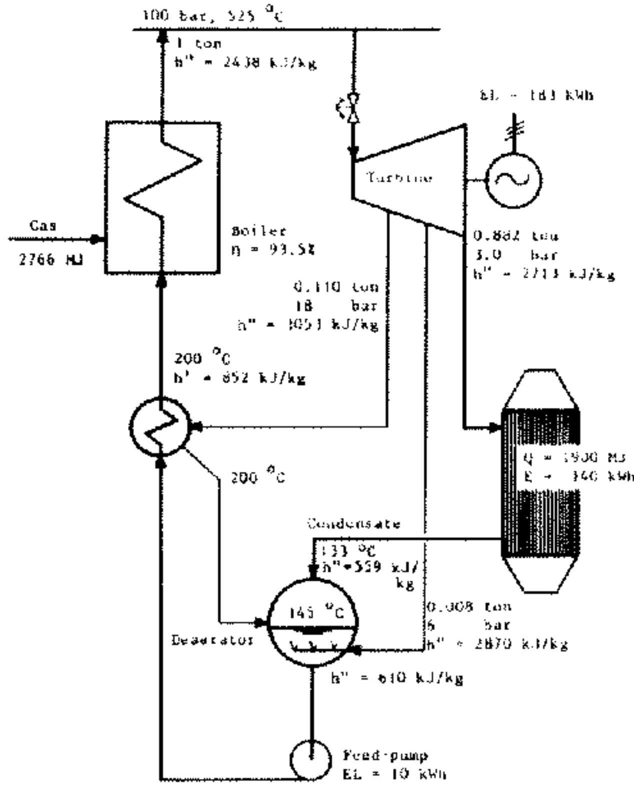

Figure 6. Cogeneration of heat and power.

— exergy: 140 kWh.
— electricity: 183 − 10 = 173 kWh
313 kWh

by firing 2766 MJ of fuel.

Therefore, the exergy efficiency is:

$$\frac{313 \times 3.6}{2760} = 40.7\%$$

This figure is well near to the efficiency of a modern public power plant. Taking transmission line losses into account, such a station uses about 9.2 MJ/kWh, which is equivalent to an efficiency of 3.6/9.2 = 39%.

The net electricity output of the steam-power plant in Figure 6 is

$$\frac{173}{0.882} = 196 \text{ kWh per ton of LP-steam.}$$

After substracting the power consumption of the evaporation plant (Figure 2) of

$$\frac{20}{0.85} = 25 \text{ kWh per ton of steam.}$$

there still remains a surplus of about 170 kWh. In general it will not pose too much of a problem to sell this to either another plant or to the public grid.

Efficiency could be stepped up still further to at least 45% by choosing a cycle consisting of a gas turbine, a dual-pressure waste heat boiler and a back-pressure steam turbine. However, with such a cycle the electricity output per ton of LP-steam is over 500 kWh. Getting rid of such a power surplus at a reasonable price could create some difficulties, although it is not impossible.

Two conclusions can be drawn from the foregoing examples:

1. Generating steam together with power can be carried out with an exergy efficiency of about 40%. Under certain conditions even 45% could be attained.
2. As this efficiency is about the same as the one of a public power plant it may be expected that the price of self-generated exergy—either in the form of steam or power—will be more or less equal to the price of the electricity interchanged with the public grid. To express it more precisely, by "price" it is meant the variable cost of generation.

This means that the exergy of heat and electricity are not only equal in a thermodynamical sense but also in the sense of money value.

These conclusions come in very handy, because the task as outlined before, namely, the optimization of the evaporator, can be done now without paying any attention to the way the exergy is supplied, so long as its generating efficiency is about 40%.

Founded on these two conclusions the exergy price can now be fixed. With a fuel price of P_E guilders (or dollars) per GJ of LHV (low heating value) and an exergy generating efficiency of 40%, the exergy price P_E amounts to:

$$P_E = \frac{P_F}{0.40} \times 3.6 = 9.0\ P_F\ \text{f/Mwh}. \qquad (3)$$

SOLVING THE PROBLEM

The optimization problem defined above has been solved for the conditions prevailing at the end of 1982, viz. a fuel price (in the Netherlands) of P_F = f 12.70/GJ. Application of Equation 3 sets the exergy (and electricity) price to be P_E = f 114.-/MWh. For the scale of production has been chosen an annual salt output of 200.000 tons per single effect, which corresponds to a production rate of 25 tph and a heat load Q = 46 MJ/s. The results have been plotted in Figure 7.

At the optimum thus established the saturation temperature of the condensing steam (or vapour) is found to be 98.0°C and the temperature of the vapour emerging from the evaporator: 90.2°C.

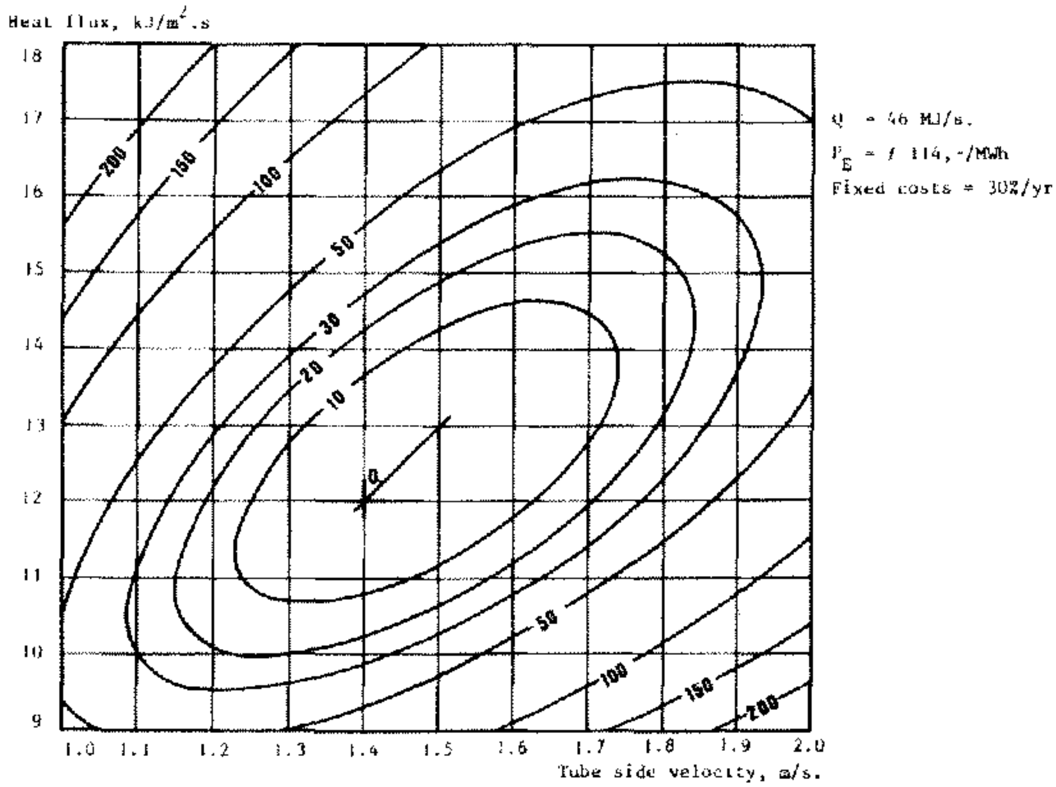

Figure 7. Total annual cost "valley", relative to the bottom at 0-level. Contours in *f* 1000,-.

At this temperature difference of 7.8°C, total exergy loss amounts to 0.79 MW and power consumption to 0.23 MW. Since the hourly production rate (in one effect) is 25 tons, the specific figures are 32 and 9 kWh/t.s., respectively, giving a total consumption related to heat transfer and boiling in the evaporator of 41 kWh/t.s.

Referring to the "exergy balance" given above, where this specific part of the consumption was rated at 68 kWh/t.s., it is seen that the total exergy consumption is reduced by 27 kWh/t.s. to about 133 kWh/t.s. This means that the optimal thermodynamic efficiency has gone up to: 40/133 = 30%. At the steam-power plant, efficiency of 40% mentioned before this is equivalent to the use of $133/0.40 \times 3600 \times 10^{-6} = 1.20$ GJ/t.s. of primary fuel (or 550 Btu (HHV)/lb salt in Anglosaxon units).

THE NUMBER OF EFFECTS

Up to now the number of effects has been left out of consideration completely. This is allowable, because what has been done boils down to finding the optimum temperature difference to transfer the latent heat of condensation and evaporation. Addition of an average value for the boiling point elevation of about 8.5°C brings the optimum drop in vapour saturation temperature to 7.8 +8.5 = 16.3°C per effect. So long as this temperature drop is applied, the energy consumption will be equal to the figure calculated in the foregoing paragraph, irrespective of the number of effects. The only requisite that has to be fulfilled for a multiple effect plant is that the steam is expanded in a turbine as far as the pressure demanded by the plant, thereby producing useful work.

If the cooling water outlet temperature is again taken at 40°C, the saturation temperature of the steam for a N-effect plant should be: $t_s = 40 + 16.3.N$°C And putting it in reverse:

$$N = \frac{t_s - 40}{16.3} \quad (4)$$

In making the choice of the number of effects to be installed, attention should be paid to some practical considerations, viz.:

1. In order to benefit by the economy of scale of the individual evaporators, it is advisable to restrict the number of effects, implying that the steam pressure should be as low as possible.
2. Expansion of steam well into the wet region should be avoided, which puts a lower limit to this pressure.

The lowest steam pressure applicable depends to some extent on the steam turbine inlet conditions, as is shown by the following examples:

Steam-power plant	HP-boiler + Steam turbine		Gas turbine + W.H.-boiler + Steam turbine
Turbine inlet conditions			
pressure, bar	100	70	30
temperature, °C	525	510	390
Internal efficiency = %	87	87	82
Back pressure at saturation, bar.[2]	3.2	2.0	1.8
Saturation temperature, t_s °C	136	120	117
With Eq. (4) N =	5.9	4.9	4.7

So at a minimum steam pressure of about 2 bar, the obvious choice for the number of effects is 5.

THERMOCOMPRESSION

As a first approximation it may be said that the optimum temperature drop of 16.3°C, as mentioned in the preceding section, will be valid for an electrically driven thermocompression plant as well as one driven by steam. It is approximate only because the compressor has not been included in the optimization calculations.

The advantage of a thermocompression unit lies in the fact that no heat has to be rejected to the surroundings, so the exergy of this heat to the amount of 22 kWh/t.s. (par. 2) is saved. However, the work to be done to overcome the temperature drop of 16.3°C, which is equal to the sum of the ideal work of 40 kWh/t.s. (par. 2), and the optimum exergy loss of 32 kWh/t.s. (par. 6) is delivered by electric power not at 100% efficiency but at, for example, 80%. So there occur extra losses of about:

$$\frac{20}{80} \times (40 + 32) = 18 \text{ kWh/t.s.}$$

Therefore, the total energy consumption will not differ much from the figure mentioned in the section labeled "Solving the Problem."

RESULTS AND CONCLUSIONS

It has been demonstrated that the exergy concept is an adequate tool to determine the optimum temperature difference by which the evaporation process should be carried out. Once this temperature difference has been ascertained, the thermodynamic efficiency of the process and consequently the actual work required is known. Depending on the efficiency of generating work (or exergy) the work required may be converted to a fuel consumption.

By way of a numerical example it has been shown, that at the present-day fuel price the optimum-process thermodynamic efficiency is 30%, compared to 25% a decade ago. At a minimum work requirement of 40 kWh/t.s. this means that actually:

$$\frac{40}{0.33} = 133 \text{ kWh/t.s. of work is needed.}$$

As cogeneration of steam and power can be performed at an exergetic efficiency of about 40%, the optimum fuel consumption of an evaporation salt plant is found to be:

$$\frac{133}{0.40} \times 3600 \times 10^{-6} = 1.20 \text{ GJ/t.s.}$$

In contrast with what one would assume at first glance, this figure is not dependent on the number of effects so long as full use is made of the power generating potential of the steam. However, in order to avoid the extraction of wet steam, a lower limit is set to the back pressure. From this practical point of view one arrives at the conclusion that the minimum number of effects should be 5.

In principle, the optimization of a thermocompression evaporator does not pose a different problem. It is, therefore, not astonishing that the optimum primary fuel consumption to generate the shaft power requirements is about the same as the above-mentioned figure.

ACKNOWLEDGMENT

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APPENDIX

Exergy Loss and Power Consumption Functions

Basis:

Evaporator temperature (= temperature at the calandria inlet) = 92°C (= 365 K).
Temperature of the surroundings = 25°C (298 K).

Calandria tube diameter	= 32 mm.
tube length	= L m.
Heat Load	= Q MJ/s.
Tube side velocity	= V m/s.
Heat flux	= F kJ/m² · s.

Equations:

Overall heat transfer coefficient:

$$U = \frac{100}{312(FL)^{0.33} + 4.8 + 25.5\, V^{-0.8}} \ (\mathrm{kJ/m^2\ ^\circ C\ s.})$$

Pressure drop $\Delta P = 21 + (1 + 0.62\, LV^{-0.16})\, V^2$ (kN/m²)

Heat transfer surface: $A = \dfrac{1000Q}{F}$ (m²)

No. of tubes: $N = \dfrac{10A}{L}$

Circulator rate: $W = 0.66 \times 10^{-3}\, NV$ (m³/s).

Temperature rise in the calandria: $\Delta t = \dfrac{0.285Q}{W}$ (°C)

Effective temperature difference (= difference between the saturation temperature of the condensing steam or vapour and the evaporator temperature):

$$\Delta t_e = \frac{e^m \Delta t}{e^m - 1} \ (^\circ C) \qquad \text{Exponent } m = \frac{\Delta t \cdot U}{F}$$

Non-equilibrium temperature drop $\Delta t_{ne} = 0.67\, \Delta t$ (°C)

Exergy loss (heat transfer and boiling):

$$\left(\frac{1}{365 - \Delta t_{ne}} - \frac{1}{365 + \Delta t_e}\right) 298\, Q \ (\mathrm{MW})$$

Circulator power consumption: $1.5 \times 10^{-3}\, W\, \Delta P$ (MW).