ENERGY GENERATION AND UTILIZATION

By

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1. Considerations in the Design of Steam Generating Units

1.1 Boiler heating surface and heat transfer equipment design:

Some of the more important factors in the selection of the two type of unit are the following:

1. Steam flow, pressure, and temperature.
2. Reheat steam flow, pressure and temperature, entering and leaving.
3. Feedwater temperature and feedwater conditions.
4. Load characteristics and type of service.
5. Fuel and firing.
6. Draft.
7. Efficiency and economics.

The objective in the design of boiler heating surface and other so called heating surfaces such as superheater, economizer and air heater isn't establish the combination of tube diameter, tube spacing, length of tubes, number of tubes wides and deep, and gas baffling that will give the desired gas temperature drop with an acceptable draft loss. In other words, the design should provide an optimum combination of heating surface and gas mass flow to give the desired results. A practical approach, based primarily on experience, must be followed in view of the fact that no one relationship can be provided the so-called optimum interrelationship among the variables mentioned. The problem must therefore be approached in two steps:

1. For a certain gas mass flow, establish (by trial and error computation) the true relationship between surface and the temperature drop (n rise) desired.

2. To establish the draft loss (pressure drop) for this gas mass flow this surface.

The general relation which relates heat transfer across an area with the heat "gained" by the fluid is:

$$UA\ T_m = w'\ c'\ (t'_2 - t'_1)$$

where t'_2 = exist temperature of the fluid

t'_1 = inlet temperature of the fluid

w' = fluid rate of flow
c' = fluid specific heat
U = overall coefficient of heat transfer
A = total surface area
T_m = long mean temperature difference

Similarly,

$UA\ T_m$ must also equal the heat out of the hotter fluid, or $UA\ T_m = w'\ c'\ t'_2 - t'_1) = wc\ (t_1 - t_2)$

where U, A, and T_m have the same meaning mentioned above, and

w = mass flow rate of hotter fluid
c = specific heat of hotter fluid
t_1 = inlet temperature of hotter fluid
t_2 = outlet temperature of hotter fluid

It should be noted that we have mentioned earlier in our discussion that the overall coefficient which actually includes both inner and outer surface coefficients and the thermal conductivity of the wall separating the hot and the cold fluid. For practical purposes however, we can neglect the resistance to heat transfer to the wall and unite the overall coefficient as:

$$U = \frac{1}{\frac{1}{h_1} + \frac{1}{h_2}} = \frac{h_1 h_2}{h_1 + h_2}$$

(here we are assuming that even for cylindrical wall, $r_1 = r_2$)

where: h_1 = surface coefficient of heat transfer on one side of the wall

h_2 = surface coefficient of heat transfer on the other side of the wall

Further simplification in the calculation of the overall coefficient U may be accomplished for some specific cases, such as boiler and economizer surface. Coefficient (all this h_w) is very much higher than the hot gas side coefficient, h_g. Therefore the quantity,

$\frac{1}{h_w} << \frac{1}{h_g}$ so that $U = H_g$ overall gas-side coeffiicient of heat transfer.

The overall gas side coefficient, will be composed of a convection coefficient and a radiation coefficient. The radiation coefficient will be defined as follows:

if q_r = heat transferred throught radiation to the surface under consideration, then h_r, radiation coefficient is defined as

$$h_r = \frac{q_r}{A(t_r - t_s)\ \text{mean}} = \frac{q_r}{A(T)\ \text{mean}}$$

$$\text{where} \quad = \quad A(T_r^4 - T_s^4)\ F_A F_c$$

Let us now outline the sequence of calculations involved in the design and sizing of the variouus equipment we mentioned above.

1.2 Mass flow:

Based on past experience it is known that for a given set of conditions such as fuel used (amount and characteristics and allowable overall draft loss, surface, boiler, superheater, economizer or air heater is arranged as tubes of a suitable diameter disposed in a bank and spaced in both directions) within a fairly limited range. Tube length is generally determined by the drum or header locations. In heat transfer and fluid flow problems it is essential to know the limits set by good practice for mass velocity ($G = w/A$, where w in, say lb/hr, A in ft^2). With tube diameter, spacing and length established, the width of the tube bank is based on an acceptable mass flow of the fluid in which the free flow area is a factor.

The "depth" of bank, or the number of rows deep, is frequently limited by the usable circumference of the drums or headers to which the tubes are connected. Other considerations, such as the maximum distance foe effective soot blowing, will sometimes limit the depth of bank. In any case, since it is necessary to start with a tube arrangement, experience will permit a tentative selection of the height and width of the receiving surface and the tube diameter of this bank for a given mass flow of the entering gases. Having this selected specific arrangement of surface, it is possible to calculate the total amount, of surface required for the desired temperature drop by the use of the heat balanced mentioned.

1.3 Boiler surface; temperature drop

As an example of the use of the heat balance, a simple case of boiler surface to cool combustion gases will be discussed.

From the heat balance the total boiler surface is

$$A = \frac{w_g c_g T_g}{U \Delta T_m}$$

where the total surface, A, and the number of rows of the tube bank are still to be determined.

To determine the total surface, A, required to cool the gases to the desired temperature, it is necessary, as mentioned earlier, to set certain conditions, known from experience to be fairly applicable to the case selected.

Economizer Surface:

The surface for a non-steaming economizer for (a) desired gas temperature drop, T_g, or for (b) desired water temperature rise T_w, can also

$$UA \Delta T_m = W_g C_g \Delta T_g$$

$$\text{and } W_g C_g \Delta T_g = W_w C_w \Delta T_w$$

For the condition (a), the values W_g, C_g and ΔT_g are known. So that if W_w and the water inlet temperature are known we can determine the water outlet temperature. The log-mean-temperature difference ΔT_m is then established for a particular heat exchange flow pattern (parallel flow, counter flow cross flow or multi-pass).

For the condition (b), if the quantities W_g, C_g, gas inlet temperature, W_w, C_w and ΔT_w are known, we can determine ΔT_g that the mean temperature $(\Delta T)_m$ is also established. However if C_g is not known, we can, by trial and error assume gas exit temperature (with known gas inlet temperature) we can find a trial C_g. The right values of temperature and C_g will balance the energy equation, $W_g C_g \Delta T_g = W_w C_w T_w$. ΔT_m is then established for a particular heat exchange arrangement. This leaves, as in case (a), the determination of U and A. Or,

$$A = \frac{Wg\ Cg\ Tg}{U \Delta T_m}$$

After the tube diameter, arrangement of surface and tube bank dimensions suitable for economizer practice (economizers use small diameter, thin-walled tubes, closely spaced), we can proceed in determining A and U. Analogous to the previous example, $U = Ug = h_{com} + h_{rad}$, in as much as the water side coefficient is considered very much higher than the gas side coefficient. The procedure is determining A is also analogous to the previous example, that is, by trial and error, we need to assume an area to find U. This value of U is substantiated into the above equation to check whether the assumed value of A is closed enough to the calculated value. If the two differ from each other significantly, a reiterative processes followed, where the new value for A is used in recalculating a new value for U. This in turn is used in finding another approximation for A.

The draft loss across an economizer is calculated using the same equation as that for the draft loss for gass cross-flow through a bank of tubes.

In practice, water inlet temperature to an economizer is kept above 175°F to avoid cooling the boiler gas to the dew point, which could cause severe corrosion with most fuels. Proper feed water conditioning avoids internal corrosion.

Economizer usually use finned tubes, where there are in the neighborhood of fifteen tubes per row.

Superheater Surface:

For the superheater surface, the two basic heat balance equations (repeated here for convenience) used are,

$$U\,A\,\Delta\,T_m = Wg\;Cg\,\Delta\,Tg$$

and

$$Wg\;Cg\,\Delta\,Tg = W_s\,C_s\,\Delta\,T_s$$

where the subscript $_s$ stands for steam, and the usual symbols stand for those already defined earlier.

For the superheater, the steam side and the gas side heat transfer coefficients are comparable in magnitude, so that,

$$U = \frac{U_g\,U_s}{U_g + U_s}$$

where U_g is defined as the gas side coefficient

U_s = steam side coefficient

For a desired temperature rise, ΔT_s, it is possible, after finding the value ΔT_g U_g, U_s and $(\Delta T)_m$, to determine the surface A.

The surface so calculated, however will be reduced somewhat, when account is taken of cavity radiation from the mass of gas approaching the bank.

1.4 Air Heater Surface:

Arrangement of the surface is based on good practice from experience. The mass-flow of both gas and air should be kept within economical limits usually from 5000 to 10,000 lb/hr ft^2 for the gas, and from 3000 to 5000 lb/hr ft^2 for air. Tube spacing usually may be closer than for a superheater, since air only passes across the outside of the tubes. It should be noted (are mentioned this earlier) that the metal temperature should be kept above the dew-point. Usually the most economical arrangement is with long tubes and multipass on the air side.

The usual heat balance equations,

$$U\,A\,\Delta\,T_m = W_g\,C_g\,\Delta\,T_g$$

$$= W_a\,C_a\,\Delta\,T_a \text{ are used.}$$

For the air heater, the overall coefficient is,

$$U = \frac{U_g\,U_a\,k}{U_g + U_a} \text{, where}$$

$$U_g = h_{rad} + h_{conv},\ g$$

$$U_a = h_{conv},\ g$$

and the other quantities have been previously defined.

GENERAL GUIDE TO FURNACE DESIGN

Furnace design depends on:

- Steam-generator operating conditions
- Heat-transfer principles
- Combustion process
- Available materials
- Empirical or experience factors.

Furnace volume and proportions selected are fixed by:

- Type of fuel
- Steam capacity and conditions—maximum load, load range, pressure and temperature
- Percent of heat recovery that proves economical
- Type of firing
- Excess air
- Flame length
- Ash fusion temperature
- Stoker or grate area
- Furnace wall and arch construction

ENERGY UTILIZATION

1. *Considerations in the Design of Heat Exchangers*

1.1 *Review of Fundamental Definition.*

Heat exchanger is a device used to transfer heat from a hot fluid to a cold fluid through a solid wall separating the two fluids. This transfer of heat is brought about by virtue of a temperature gradient. It is one of the most important processes frequently encountered in engineering practice. Power plants, refrigerating plants, air conditioning systems, petrochemical plants to name a few-all these depend their operations mostly on the exchange of heat between two fluids. The boiler transfer heat from the hot combustion gases to the water which finally turns into steam; the condenser in a steam power plant or refrigerating plant transfers heat from a condensing fluid to a coolant such as air or water; the automobile radiator transfers heat from the water (an intermediate coolant) to the surrounding air; the economizer of a steam generator transfers heat from the combustion gases to the feedwater; the generator in a gas turbine transfers heat from the turbine exhaust gas to the compressed air before entering the combustor.

1.2 *Classification*

Heat exchangers may be classified into two general types depending on the relative motion of the two fluids: In-live heat exchangers and cross-flow heat exchangers.

In-live heat exchangers have the two fluid streams move parallel to each other. They are further subclassified into parallel-flow or counter-flow depending on the relative directions of the two fluid streams. Fig. 1 (a) shows the cross section of an in-line shell-and-tube heat exchanger. Fig. 1 (b) and 1 (c) schematically show respectively parallel-flow and counterflow arrangement. The number of times a fluid traverse the entire length of the exchanger constitutes the number of "passes" Fig. 1 (a) shows a single-pass tube side. Figs. 2 (a), (b), and (c) show several multi-pass arrangements.

Fig. 3 (a) shows the temperature variations in parallel-flow. Note that the coldest portion of the cold fluid comes into communication with the hottest portion of the hot fluid and the coldest portion of the hot fluid communication with the hottest portion of the cold fluid. This means that initially there is a large temperature difference at the entrance but as the fluids move through the exchanger the temperature difference drops rapidly until it reaches its minimum temperature difference at exit. The temperature of the two streams approach each other symptotically. Theoretically, an infinitely large area of heat transfer is required to raise the cold fluid to the temperature of the hot fluid or vice versa.

Counterflow heat exchangers on the other hand, places the hottest portion of the cold fluid in communication with the hottest portion of the heat fluid at one end of the exchanger and the coldest portion of the cold fluid in communication with the coldest portion of the hot fluid at the other end of the exchanger.

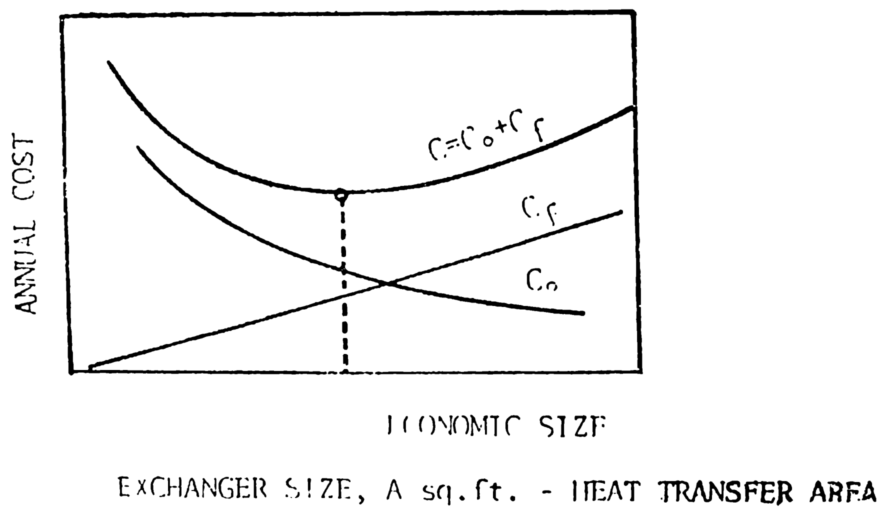


FIG. 1 *General Relationn Between Costs And Heat Exchanger Size.*

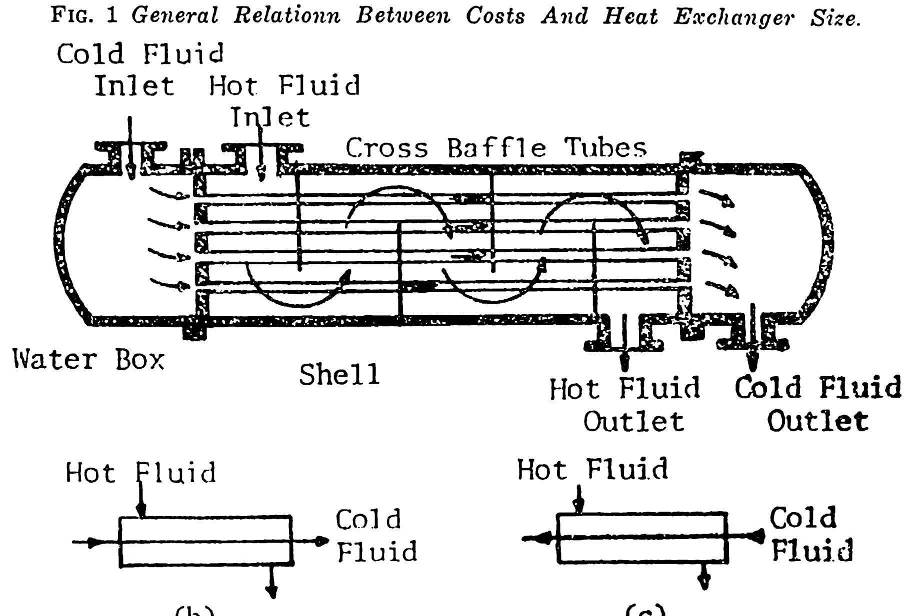


FIG. 2. *Scheme drawing of (a) 1 one-shell-pass, one-tube pass heat exchanger (b) parallel flow (c) counterflow.*

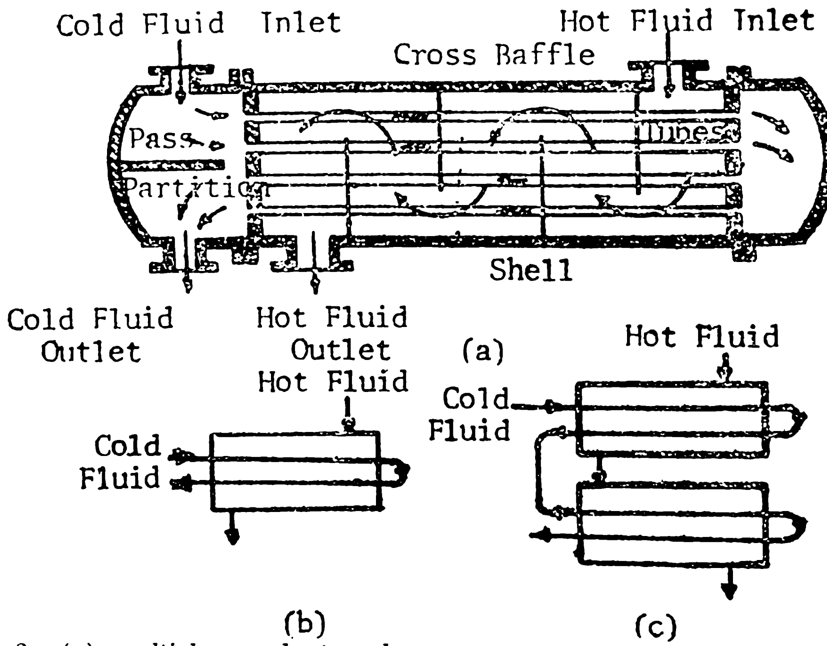


FIG. 2. *(a) multiple-pass heat exchangers (b) one-shell pass and two-tube passes, (c) two-shell passes and four-tube passes.*

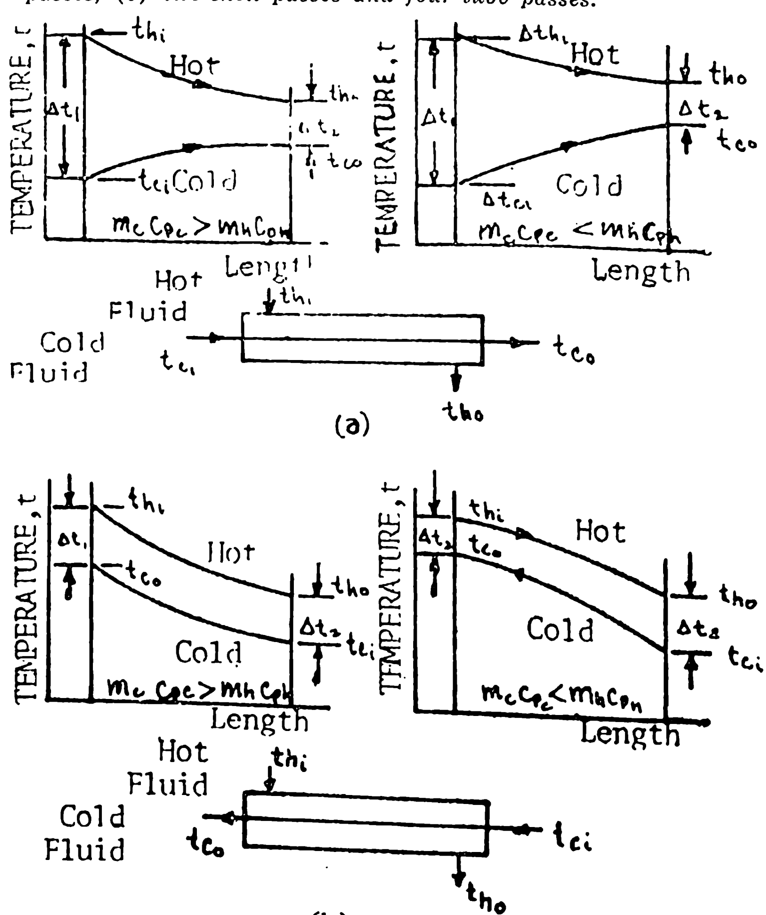


FIG. 3. *Temperature variations in (a) parallel-flow and (b) counterflow heat exchangers.*

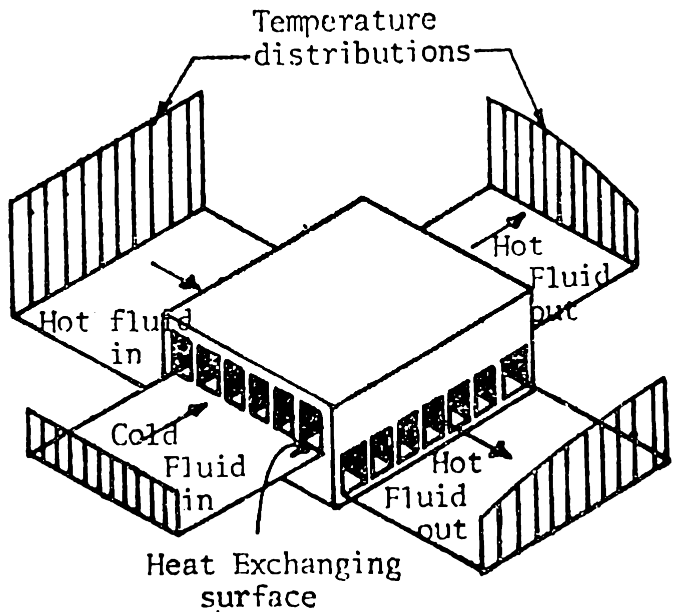


FIG. 4. *Temperature variation nof uncountered fluids in a cross-flow heat exchangers.*

This provides a nearly constant temperature difference between the two fluids along the heat exchanger and allows the possibility of heating the cold fluid to a temperature greater than the outlet temperature of the hot fluid or vice versa. In general the counterflow exchanger results in a greater heat transfer rate than the corresponding parallel flow case.

Fig. 4 shows a simplified plate type cross-flow heat exchanger. A look at the temperature profiles at the inlet and outlet of each fluid will show that the analysis of this type of heat exchanger is slightly more complex than in-line exchanger. The automobile radiator is a common example of cross-flow heat exchanger.

1.3 *Heat Transfer Consideration*

The heat transfer rate from the hot fluid to the cold fluid may be expressed by the following equations:

$$q = U \Delta t_m \quad (1)$$

$$q = M_c C_{pc} \ (t_{co} - t_c) \quad (2)$$

$$\text{and } q = M_h C_{ph} \ (t_h - t_{ho}) \quad (3)$$

where U = a suitable mean overall heat transfer coefficient based on Area A.

A = total area of heat transfer

Δtm = a mean temperature difference

m = mass flow rate

Cp = specific heat at constant pressure

t = fluid temperature

subscripts c and h denote cold and hot fluids respectively i and o denote inlet and outlet conditions respectively.

The reciprocal of the overall heat transfer coefficient 1/U may be thought of as a total resistance to heat transfer consisting of several resistances in series. Lower values of these resistances results to higher overall coefficient of heat transfer.

For a given heat exchanger, increasing the overall coefficient U (decreasing the total resistance R_t) increases the capacity of the exchanger and may also decrease the Δ tm. The value of U may be increased by increasing the velocity of either or both fluids because increasing the velocity of the fluid increases the convective coefficient of heat transfer. For example, for forced convection inside cylindrical tubes the convection coefficient is proportional to $V^{1/3}$ for laminar flow and is proportional to $V^{0.8}$ for turbulent flow.

However, increasing the velocity of the fluid means larger pumping requirements for that fluid. Thus, the positive gains (higher capacity or smaller size of exchanger required and lower Δ tm) due to increased U should be compared with the negative effects such as higher pumping cost. Actual numerical calculations should be made to provide the engineer with a firm basis for decision.

Scale formation on the heat transfer area decreases the overall coefficient U. It is usual or normal to design the heat exchanger taking into account the effect of scale formation, water treatment and/or regular de-scaling maintenance procedures may be required. These of course will involve additional investment for the water treatment system and/or additional operating cost for the water treatment system or maintenance cost for regular de-scaling requirement not to mention plant downtime in case the plant has to be shut off.

1.4 *Cost Consideration*

The performance of particular heat exchanger in a given system may affect the total power consumption of the whole system. For example a decrease in the effectiveness of the condenser or evaporator of a refrigeration system increases the compressor power requirement; the efficiency of a steam power generating plant is affected by the performance of the difference heat exchangers in the system such as the boiler, economizer, reheater, condensor, etc.

The economic design or selection of heat exchanger or any engineering undertaking for that matter should consider as much as possible among other things all the factors or at least the major factors that contribute to the total cost, For convenience this total cost (Capital Cost) C_f and operating cost C_o. The general relations of these costs with respect to exchanger size are shown in Fig. 1.

Note that the fixed cost increases with the exchanger size while the operating cost decreases as shown. The total annual cost is the sum of fixed and operating costs and has a minimum which can be considered the economic size. If the total cost curve is relatively flat at the bottom then the range of economic size is relatively large which means that the minimum annual cost is not so sensitive to the exchanger size in that size range. On the other hand if the bottom of the curve is sharp then the choice of economic size is very limited.

The installed costs of heat exchanger equipment depends on many factors. In general it depends on the type and design and whether it can bought off the shelf or it has to be made to order because of unusual design or duty requirements. For a given type and design however, the installed cost is proportional to the heat transfer area raised to an exponent ranging from about 0.6 to 0.7. This exponent can be determined by plotting on a semi-log paper the cost versus heat transfer area. The exponent can then be evaluated from the slope of the line.

The mean temperature difference (Δ tm) is some kind of an average temperature difference between the hot fluid temperature and cold fluid temperature. For a pure counterflow heat exchanger, this Δ tm is equal to the logarithmic mean temperature difference (LMTD). This relation is also true for a heat exchanger process where at least one of the fluid practicaly remains at constant temperature (when there is a change of phase such as condensing or evaporating process or when the capacity rate MC_p of one fluid is very large compared to that of the other fluid) regardless of the flow arrangement — be it counterflow, parallel flow or cross-flow.

1.5 *Economics of Air Versus Water — Cooled Condensers.*

Oftentimes, the engineer is faced with a problem of choosing between air-cooled or water-colled condenser, say for a refrigeration system. The final choice may be arrived at by considering just a few restraining factors such as availability of space, supply of water as coolant (this may require a cooling tower) or it could even be based purely on personal learnings of the engineer. On the other hand he could consider as many factors as he can think of. Of course, a knowledge of the relative magnitude or effects of the different factors is desirable so that he can at the start disregard the negligible ones. He can make his choice purely on the basis of the system power (energy) requirements regardless of initial investment or he can base it on the total cost of the system, operating plus fixed costs.

The following discussions may serve as guiding principles in making the "right" decision. These basic principles however must be translated into quantitative figures so that he will have some numerical basis for his decision. The specific problem under study revolves around whether to use an air-cooled or a water-cooled condenser for a given refrigeration system.

a) *Condensing Temperature.* Generally the condensing temperature in an air-cooled condenser is higher than in a water-cooled condenser. This is because the air dry-bulb temperature is the controlling factor for the air-colled condenser while it is the air wet-bulb temperature for the water-cooled condenser. Furthermore, the typical mean temperature difference for air-cooled condensers is generally larger than that for water-cooled condensers. Typical condensing temperature during summer here in the greater Manila area about 125°F and 100°F. Respectively for air-cooled and water-cooled condensers. The usual maximum outdoor air temperature is about 95°F and cooling water temperature from a cooling tower is around 85°F when the air wet-bulb temperature is 80°F WB.

b) *Effect of Condensing Temperature on Compressor Power Requirement.* An analysis of a conventional vapor compression refrigeration system will show that for a given refrigeration load at a given evaporating temperature the compressor power requirement increases with increasing condensing temperature. This can easily be verified from the thermodynamic analysis of the refrigeration cycle. It can also be verified from manufactures performance data. Thus more compressor power is generally required by an air-cooled refrigerating system than by a water-cooled system to produce the same refrigeration capacity at a given evaporator temperature.

c) *Capital Cost Comparison.* Generally, an air-cooled condenser is more expensive than a water-cooled condenser of the same capacity. However, the cost of the cooling tower for the water-cooled condenser may more than offset the price difference. Additional minor items such as pipings, foundations, electrical installations, etc. for both types of condensers and water pipings, water pumps, water treatment plant, etc. for the water-cooled condenser must be included in estimating the total capital cost for each type. Actual costings of the different items are necessary to arrive at the capital costs.

2. *Steam Prime Movers*

2.1 *Power From Steam*

Mechanical power from steam is derived by expanding the steam in a piston-cylinder system (steam engine), by allowing the steam to expand in static any nozzles thereby attaining in high velocity steam jet which flows over moving blades without further expansion (impulse turbine), or by partially expanding the steam through fixed blades and finally completing the expansion through moving blades (reaction turbine). The expanded steam or exhaust steam is either thrown away which is wasteful, or it is used as process steam, or it is recovered by condensing it and then pumping it back to the steam generator.

In the steam engine the reciprocating motion of the piston in a cylinder is converted in the rotary motion by means of a slider-crank mechanism consisting at a connecting rod and a crank. In the turbine, the moving blades or buckets are attached to the rim of a rotating drum. It is this rotary motion that mechanical power from the steam engine or turbine is used to drive devices or equipment such as pumps, blowers, fans, compressors, or electric generators for electric power generation.

Steam turbines are more compact than steam engines since they run at speeds about ten(10) times the steam engine speeds. Steam turbines are designed to run at a few thousand rpm (1200 to 4000 rpm) while steam engines are sually designed to run at much lower speeds (150 to 350 rpm). Steam engines are normally used to drive small loads up to 1,000 hp. They have better heat rates at capacities less than 1,000 hp. Although steam turbines are commercially available at sizes as low as 10 or so horsepower, their usual application is for commercial generation of electric power from a few thousand kilowatt to as large as 500,000 kw per single turbo-generator unit.

Steam engines operate at low throttle pressures which are in the order of 80 to 200 psig while turbines can operate up to much higher pressures in the order or 5,500 psig. The present maximum inlet steam temperature of 1150°F is dictated by metallurgical considerations.

2.2 *The Steam Power Cycles*

a) *Rankine Cycle.* A schematic diagram of a simple steam power plant operating on a simple Rankine cycle is shown in Fig. 6. Supperheated steam from the boiler expands through the turbine or steam engine producing mechanical energy. The exhaust steam is condensed in the condenser and the condensate is then pumped back to the boiler where heat energy from the combustion of fuel is used to produced superheated steam.

b) *Reheat Cycle.* Metallurgical limitations usually determine the maximum temperature of the superheater in a Rankine cycle. Likewise, metallurgical problems associated with erosion usually limit the amount of entrained liquid that can be allowed to form in the steam as it expands through the turbine or engine. These two limitations from the thermodynamic standpoint, greatly reduce the amount of work that can be realized from a simple Ranking vapor cycle. To satisfy both limitations and at the same time increase the work output and the thermal efficiency, steam would be withdrawn from the turbine at some intermediate static stage, returned to the steam generator for resuperheating and reintroduced into the turbine at the following stage. Fig. 7 shows a Rankine Reheat Cycle with one reheater. Fig 8 shows the variation of efficiency with reheat pressure in a reheat cycle.

c) Regenerative Cycle. This cycle uses extraction steam for feedwater heating (Fig. 9). The major irreversible process of the Rankine cycle is the mixing of the cold condensate with the saturated water and steam mixture in the boiler. Feedwater heating by steam extracted from the turbine minimizes this irreversibility. The thermal efficiency of a regenerative cycle increases with the number of extractions (see Fig. 10) and would reach a limiting value as the number of extractions or feedwater heaters reaches infinity. In practice it is rarely economical to use more than four or five stages of feedwater heating.

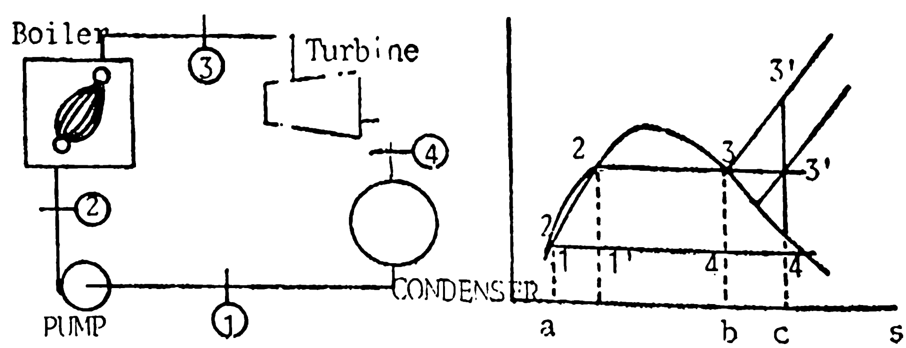


FIG. 5. *Simple power plant which operates on the Rankine cycle.*

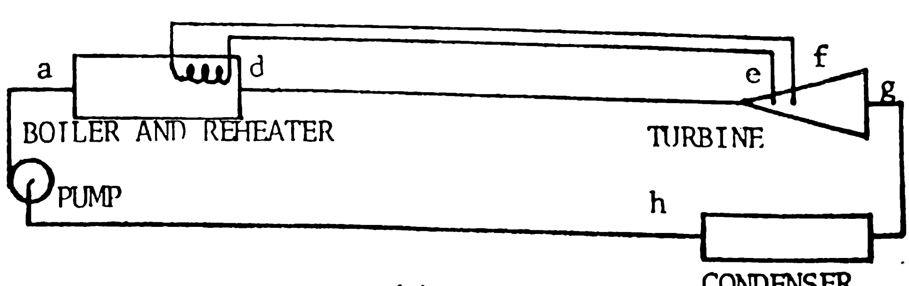


FIG. 6. *The Rankine Reheat Ccycle.*

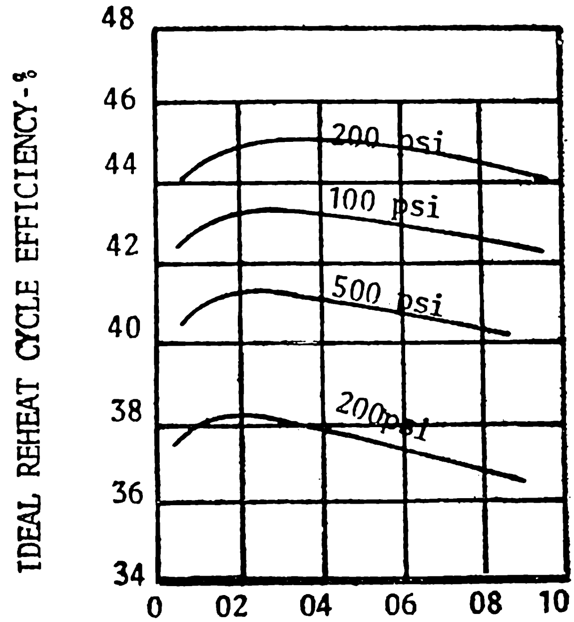


FIG. 7. *Reheat-cycle efficiency variation with reheat pressure. Curves are for ideal cycle.*

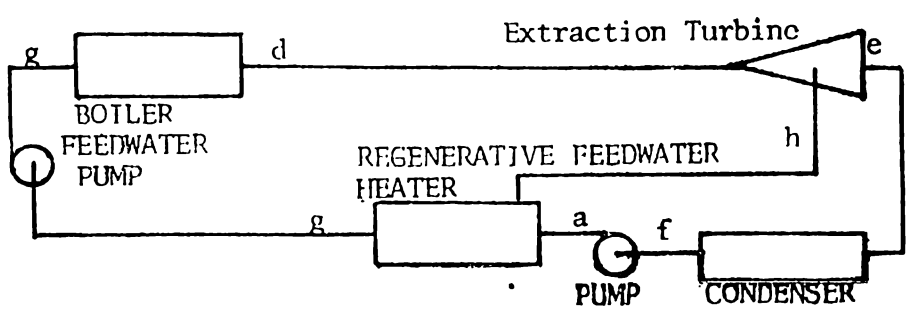


FIG. 8. *Regenerative cycle with staged feedwater pumps.*

d) *Reheat Regenerative Cycle.* In many central station steam power plants a combination reheating and regenerative cycle is used, the number of reheats and the number of regenerative feedwater heaters being determined by the size of the station and by the cost of fuel. Fig. 11 shows the cycle and the equipment arrangement for a system in which there are two extraction points, each point serving a reheater and a regenerative feedwater heater.

e) *Binary Vapor Cycle.* This cycle essentially uses two Rankine cycles using two different working fluids (water and mercury) coupled together with a heat transfer link. Fig. 12 shows a schematic diagram of a binary vapor cycle.

2.3 *Improving the Performance of Steam Power Plants*

Rising fuel costs and the steady increase in unit rating have given impetus to the demand for more efficient power plants. Inevitably efficiency improvements involve greater initial expenditures for additional or improved equipment. Equipment which cannot be economically justified at low fuel costs becomes attractive at higher fuel costs because of the larger savings on the operating cost. As the size of unit increases, the station cost per kilowatt declines as does the cost of additional or improved equipment associated with it. Thus both rising fuel costs and increasing unit size work powerfully toward the introduction of more efficient power plants.

Steam power plants may be improved in a number of ways. Some of these are listed below. The potential improvement in some categories may be small but in others it may be large. Limitations on the improvements may be dictated by economic considerations and/or technical difficulties.

a) *Thermodynamic Improvements.*

Higher initial pressure has been given much attention as a means of improving plant efficiency. The present maximum pressures considered are in the range 4500 to 5500 psig. The difficulty and expense of designing for extremely high pressures mount steadily as the pressure is increased while gains in thermal efficiency tend to decline (see Fig. 13).

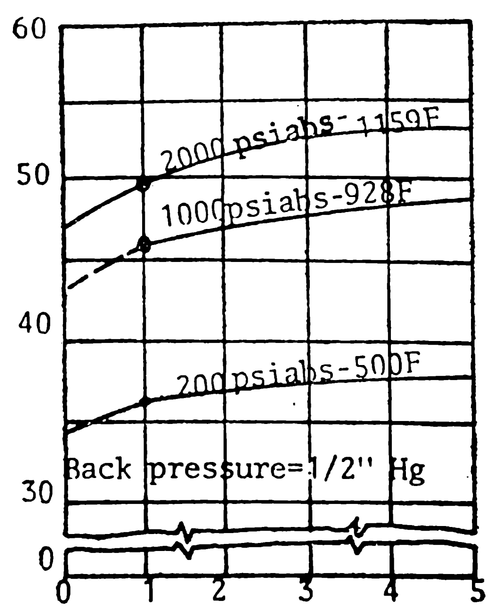
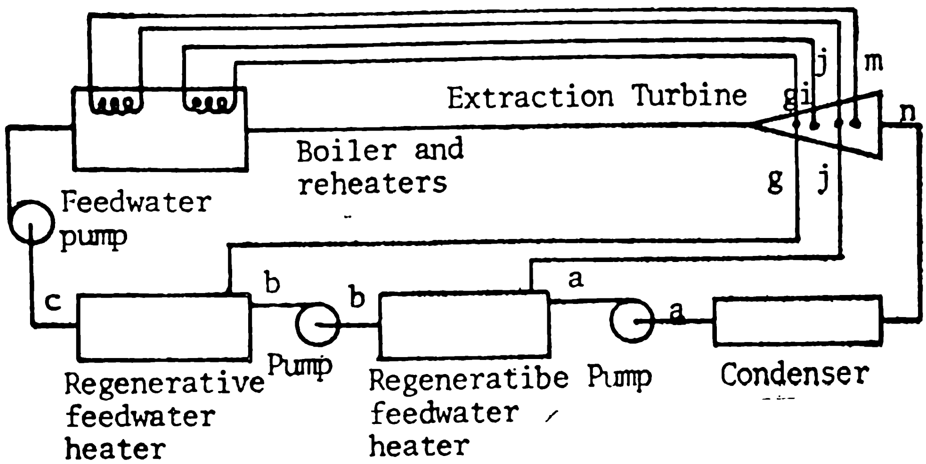


FIG. 9. *Regenerative-cycle efficiency variation with number of heaters. Curve are for ideal cycle.*



(c)

FIG. 10. *Regenerative reheat cycle.*

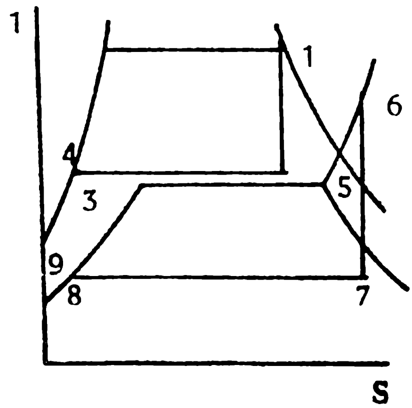


FIG. 11.

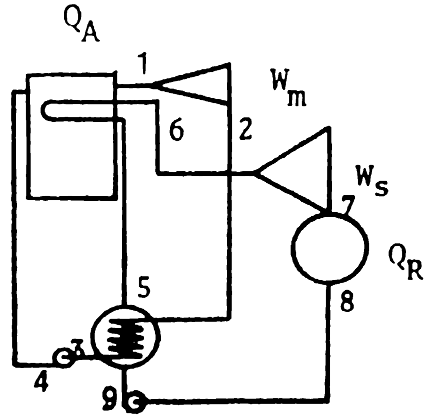


FIG. 12

Higher initial temperature offers a fruitful field for improvements in thermal efficiency (See Fig. 13). The basic problem involves the sharp decline in strength at high temperatures of currently available materials which can be economically by justified in boiler and turbine construction. At present 1050°F is the maximum initial temperature in general use although units with initial temperature as high as 1150°F have been built.

Reheating to improve thermal efficiency is a generally accepted practice. The efficiency gains to be realized from a second reheating are much less than for the first.

Higher reheat temperature results in gains comparable with those of higher initial temperatures. Again the rapid decline in the strength of materials at high temperatures limits this to 1050°F.

Binary Cycles have been used to overcome some of the disadvantages of water as the only working substance. Mercury-steam plants offer considerable improvement in thermal efficiency over plants using steam alone.

Combined Cycles. The rapid development of the gas turbine has stimulated the interest in the practicability of a combined gas-turbine-steam-turbine system. The thermal gains which can be realized are comparable to those obtainable from reheating.

Regenerative feedwater heating is a widely accepted means of improving thermal efficiency (See Fig.). Optimizing the cycle arrangement to provide the greatest efficiency gains with minimum investment continues to be a fruitful area for reducing the cost of power production.

Lower Turbine or Steam Engine Exhaust Pressure. Lowering the exhaust pressure increases the energy available, increasing the output of the turbine as well as improving the plant thermal efficiency. (See Figs. 14 and 15). Assuming that the available cooling water temperature depends on factors beyond our control, the exhaust pressure may be lowered, by increasing the condensing surface area or improving the design.

Reduced piping pressure drops and heat losses result in improved efficiency. However, the possible gains over present normal practice is small. Decreasing pressure drops and heat losses beyond normal practice is a limited and costly area in which to seek improvement.

Minimized Make-up Feedwater Flow Requirements. The introduction of make-up feedwater to a steam-power-plant cycle results in a loss of thermal efficiency. The reduction in make-up water requirements can yield some improvements in thermal efficiency.

b) *Operation and Equipment-efficiency Improvements.*

Improved boiler efficiency. Improved boiler efficiency is a direct method of increasing plant thermal efficiency. This is discussed by another speaker in this seminar.

Improved turbine internal efficiency also has a direct effect on plant thermal efficiency. Manufacturers have steadily improved the quality of the turbine-steam flow path. These improvements generally result from technical advances in the sciences of fluid flow and aerodynamics. The high level of turbine internal efficiency already attained makes it certain that progress in this area will be very slow.

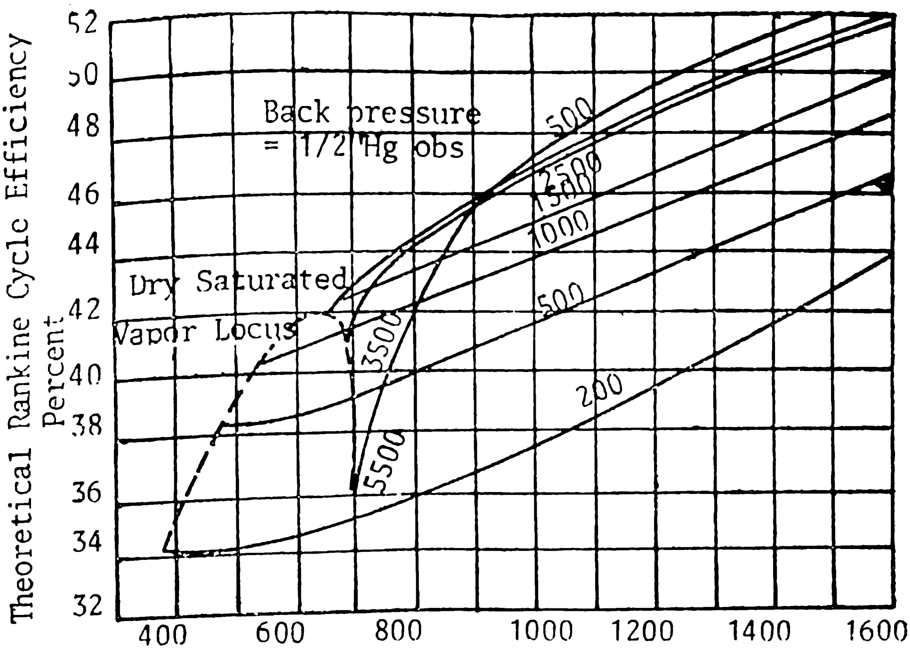


FIG. 13. *Effect of throttle steam conditions on Rankine-cycle efficiency. Note supercritical curves.*

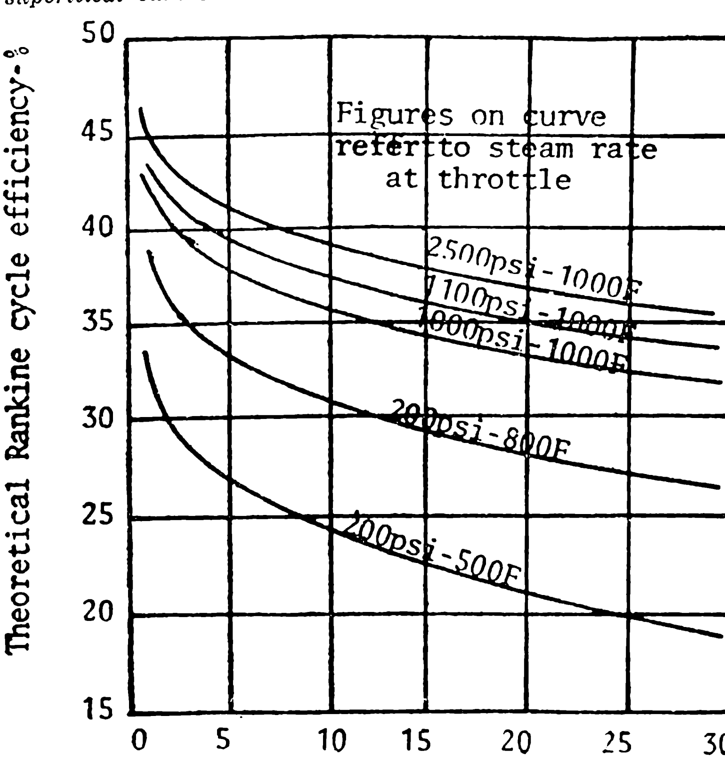


FIG. 14. *Effect of back pressuure on Rankine-cycle efficiency. Curves are for ideal cycle.*

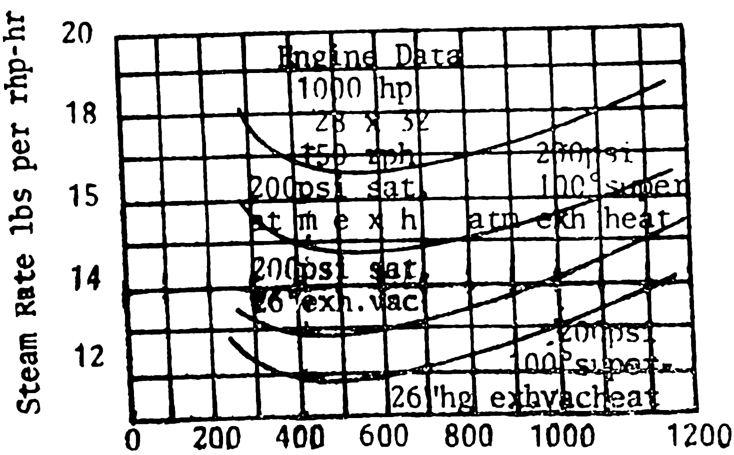


FIG. 15 *Uniflow-Engine Steam Rates.*

Operate Steam engine at minimum steam rate as much as possible. If possible schedule the load so that steam engine will operate at minimum steam rate. Fig. 15 shows the steam rate of a 1,000 hp uniflow engine with different inlet and exhaust conditions. This engine performs best at about half load. Design could make the engine perform best at full or three fourth load is so desired.

Reduced Turbine Exhaust Loss. Exhaust loss is primarily a function of the annulus area of the last-stage buckets and the exhaust pressure. Few large turbines are selected without careful attention to matching the last stage annulus area with the conditions under which the turbines are expected to operate.

Reduced Mechanical, Generator, and Boiler-feed-pump Losses. Reduction of the mechanical losses of the prime mover, the electrical losses of the generator, the hydraulic and mechanical losses of the boiler feed pump, and minor equipment losses are all means of improving the plant efficiency. In general these are not overriding factors in the selection of equipment since the room for improvement is small. Reliability is usually the determining consideration for these items.