ENERGY CONSERVATION

Energy Conservation Tools for the Process Engineer

Upgrading Existing Evaporators to Reduce Energy Consumption

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Technology Applications Manual
Technology Transfer

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Upgrading Existing Evaporators to Reduce Energy Consumption

A Technology Applications Manual by
the Energy Research and Development Administration
Division of Industrial Energy Conservation

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Chapter 1

POTENTIALS OF UPGRADING EVAPORATORS

OVERVIEW

Evaporation is one of the most energy-intensive unit operations used by industry. A recent estimate places the equivalent energy use for evaporation above 110 million barrels of fuel oil per year. Because of the importance of energy to evaporation, it is likely that even the oldest evaporators in a manufacturing plant were at some time subjected to an economic analysis comparing the energy cost and consumption to the capital costs for additional hardware that would conserve energy. However, because energy was relatively cheap, the economics and increased operating convenience of relatively simple units favored the installation of the less energy-efficient systems.

The radical increases in energy costs in relation to capital equipment costs have made all past analyses obsolete. As shown in Fig. 1, energy costs have escalated far more since 1973 than have process hardware and installation costs. Although the total installed costs and the additional costs for retrofitting will be higher than those indicated by the index in Fig. 1, the reevaluation of additional evaporator hardware must show significantly different and usually more attractive economics for realizing optimal energy consumption in the existing plant because the rise in energy costs has been so much greater than the rise in total installed costs.

The engineering evaluation of more energy-efficient equipment may have many constraints and in most cases will be more complicated than designing a new plant. The design rationale and performance of the existing plant must be reconstructed and then modified to meet the design criterion of the new technology. The changes in the process conditions with each modification must be evaluated not only to determine the energy savings but also to meet the demands of maintaining product quality and capacity. The evaluation of performance and the applicability of the technology must also consider physical layout and retrofit problems. For designs of new evaporators, there is considerably more latitude for design changes at any point in the system to meet the energy and performance objectives and to incorporate any option that will optimize the operating and investment costs.

The economic analysis for upgrading existing evaporators also is more complex than are cost estimates for new systems. Many individual evaluations must be made and then compared to provide the best alternative. The analysis may need to consider the effects that upgrading the evaporator will have on the remainder of the manufacturing facility. The installed costs for retrofitting are not easy to calculate since standard estimation procedures are not applicable. Schedules must be set up to minimize loss of production during the changeover, and the economic evaluation must include estimates of loss of production and delays that will measurably increase the costs for the modification. However, it is certain that in most plants

Figure 1. Escalation of equipment costs and imported fuel oil costs. (Construction and erection costs have escalated at a rate similar to that for equipment costs.)

	Imported fuel oil, \$/bbl	Equipment cost (Marshall and Swift index)
1973	3.29	344.1
1974	11.06	398.4
1975	11.49	444.3
1976	12.08	472.1

exceedingly high returns on both investment and evaluation expenses can be achieved since the investments required for modifying an existing plant are considerably lower than those for a complete replacement and the energy savings will be nearly the same.

The vast majority of new evaporators will be designed with the new energy costs taken into account. However, the replacement of older plants will account for a negligible amount of the total evaporation capacity. If any short-term gains are to be made in conserving energy and in reducing the operating costs for evaporation, the problem of upgrading existing evaporators must be addressed. By a reevaluation of all evaporators in the manufacturing facility, the energy consumption and savings can be determined, and the best upgrading method can be selected. The economic benefits are there for the engineer to achieve.

Unfortunately, there are no hard and fast rules that can be applied for these evaluations. On the basis of past evaporator design philosophies and the conditions normally found in many manufacturing processes, it is usually possible to outline the more promising areas of energy conservation and to provide the basic data required for the evaluation. With this understanding, the engineering time can be used most effectively in applying the latest information to the specific plant case.

RATIONALE BEHIND THE DESIGN OF MOST EXISTING EVAPORATORS

In the past the major criteria for the specification of the final evaporator design have been reliability of operation, minimization of capital costs, and the reduction of utility use, in that order. Because evaporation in most applications is considered to be a standard unit operation, often the final selection of the evaporator has been based on the lowest cost design from a group of vendors who based their bids on a specified evaporation duty. Since vendors were aware of the need to be competitive on the base price, it was common practice to bid a "bare bones" system that would meet the requirements of the request for bid at minimum capital cost and to offer energy-saving options as add-on options at increased prices. Although all the offered options had some positive return on investment and, in most cases, an acceptable rate of return even at the low energy costs, normally only the energy-saving options with outstanding economics found their way into the final plant.

There were sound reasons for managerial decisions that led to the design of the present evaporators, which now are lacking in energy-conservation technology.

- Normally the project manager was under strong pressure to keep the capital cost of the complete project close to the original project estimate. Because of inflation and the difficulty of accurately projecting total costs, the

final installed costs normally exceeded the project estimate and made it necessary to gain corporate approval for additional funds. Approving capital for any except the minimum hardware needed to complete the project was believed to increase the risks of both startup delays and budget overruns.

- The suggested options often appeared to add operating and maintenance complications. From the point of view of the startup team and production engineers, any changes that complicated the process or could decrease on-stream time and productive capacity were undesirable.

- When acceptable but not outstanding returns on investments were projected by the vendor for the add-on options, there was some skepticism about reaching those figures in actual practice.

- The easiest decision was to delay the study and approval of the options until after the plant was operating since the primary objective was to get the plant in operation at the earliest time.

In view of the energy costs of those times, the managerial reasoning behind this philosophy was not faulty. Building and operating an evaporator in a highly energy-efficient manner does add some complexities to the engineering, construction, and operation of the system. If the addition of hardware delays a plant startup, the possibility of decreased on-stream time and higher maintenance and labor costs could very quickly cut into the projected cost savings from energy reductions. The need for additional instrumentation and control systems, as well as increased piping, also tended to point toward operating problems and the possibility of more frequent downtimes. Given these possibilities, the usual tendency was to ensure maximum short-term profitability by putting available investment dollars in areas that would improve operating costs and on-stream times, e.g., in higher quality construction materials and spare pumps and parts.

The bulk of the highly energy-efficient evaporators installed in the past was therefore the result of special circumstances in which energy use was especially critical or costly. A plant with a limited amount of steam-generating capacity, for which an additional investment in energy-conservation hardware for the evaporator would defer the expansion and capital costs for large utility systems, had this special incentive. Also, very large evaporator systems, in which the economics for minimizing energy usage are usually more favorable, were designed.

This situation has now changed radically. Both fuel costs and investments for steam-generating and cooling water supply facilities to comply with air- and water-pollution guidelines have increased at a rate greater than that of price inflation. The economics of energy conservation are now sufficiently favorable to override much of the past design and cost reasoning. Although a greater economic incentive now exists, it is important to under-

SOURCES OF ENERGY LOSS

PROCESS AND UTILITY FLOWS

Figure 2. Sources of energy loss and process flows in the typical single-effect evaporator.

stand the limiting factors that led to past designs because many of these pressures are still present and can still restrict the number of alternatives or decrease the operating efficiencies of the evaporator.

PRESENT EVAPORATION SYSTEMS INSTALLED DURING PERIOD OF LOW ENERGY COSTS

Figure 2 shows a typical single-effect evaporator of the forced-circulation variety, illustrating past design and energy cost considerations. The major sources of energy loss are noted. In forced-circulation systems a pump maintains high velocity across the heating surfaces to minimize fouling and to provide controlled velocities through the evaporator tubes. The single-effect unit has no hardware to recover usable thermal energy. It is the simplest type of evaporator design and requires the lowest capital investment. A common application for this type of evaporator would be for low capacities or distinct physical-property problems of the liquor, such as foaming and heat sensitivity. The concentration of wet-process phosphoric acid has been a common application.

For intermediate-size evaporator applications, it was common to choose a multieffect system—usually one with two or three effects. In certain applications in which an

especially dilute material, such as waste stream or fruit juice, was being concentrated or in which the flow rate was very large, further effects were justified. For large installations, such as in the pulp and paper industry, six or seven effects sometimes were used.

A typical multieffect (three-effect) evaporator is shown in Fig. 3. This particular system is a forward-feed natural-circulation type. When operating properly, it should evaporate about 2.4 lb of water for every pound of steam used. In addition to the added effects and liquid-flow direction selected, the figure identifies other sources of energy loss that can be investigated in this typical evaporation system.

OPERATING COST ESCALATIONS OF EXISTING EVAPORATORS

The rapid escalation in fuel prices and therefore in steam costs has greatly affected the operating costs for existing evaporators. Although the costs of labor, maintenance, and material have increased, the significant factor is and will continue to be the cost of thermal energy—usually in the form of steam to evaporate water. Utility costs are no longer minor contributors to the total plant manufacturing costs.

Figure 4 shows the operating cost contribution attributable to thermal energy as a function of evaporator heat load

Figure 3. Sources of energy and process flows in the typical multieffect evaporator.

and energy costs. This graph can be used to estimate the operating costs attributed to thermal energy for evaporation, regardless of system design, by applying the appropriate plant fuel price or steam cost to the known evaporator heat load. Until the late 1960s, the cost of steam held reasonably steady within a range of 40 cents to 75 cents/1000 lb. Today the typical steam costs in most areas exceed \$3/1000 lb. Figure 4 shows that an evaporator requiring 20×10^6 Btu/hr (approximately 23,000 lb steam/hr) has experienced annual operating cost increases from approximately \$100,000 at 50 cents/1000 lb of steam to \$550,000 at \$3/1000 lb of steam.

The effect of the new energy costs on the total costs of evaporation can be very clearly shown by a simple analysis comparing the 1976 costs of an evaporator system with the costs under the conditions at the time it was built. Table 1 shows an example analysis for a double-effect evaporator built in 1967 with a total capacity of 50,000 lb water/hr. From 1967 to 1976 the operating costs for this evaporator rose 294%, of which 91% is attributed to higher steam costs. Less than 10% of the increase is attributed to all other expenses. This analysis clearly shows that the major

Figure 4. Operating costs attributable to evaporator heat load as a function of energy costs. Basis: 75% energy-generating efficiency; steam at 880 Btu/lb equivalent; 90% on-stream time.

Table 1. Operating Cost Escalation of an Existing Evaporator from 1967 to 1976

Item	1967	1976
Total installed investment in 1967: \$340,000		
Evaporation rate: 50,000 lb water/hr		
Manufacturing costs (annual)		
Operating labor (2 man-years)	\$ 15,800 (at \$3.80/hr)	\$ 28,300 (at \$6.80/hr)
Supervisor (0.5 man-years)	5,400 (at \$5.20/hr)	9,100 (at \$8.76/hr)
Maintenance costs (at 6% of investment)*	20,400	36,000
General plant overhead†	16,700	28,500
Raw materials		
Depreciation (at 10% of investment)	34,000	34,000
Taxes (at 1% of investment)	3,400	3,400
Utility costs (annual)		
Steam at 33,000 lb/hr	130,000 (at 50¢/1000 lb)	780,000 (at \$3/1000 lb)
Cooling water at 1500 gpm	14,200 (at 2¢/1000 gal)	28,400 (at 4¢/1000 gal)
Electricity at 70 kWh	3,300 (at 0.6¢/kWh)	11,000 (at 2¢/kWh)
Total operating costs excluding raw material costs	\$243,200	\$958,700
Percent increase in total costs		294
Percent increase attributed to steam cost		267
Percent increase attributed to total costs minus steam cost		27

*Maintenance labor equals 37% of maintenance costs.

†General plant overhead equals 58% of operating labor, supervision, and maintenance labor.

area of focus for efforts to lower operating costs is in decreasing the amount of thermal energy required to vaporize each pound of solvent.

AREAS FOR UPGRADING EXISTING EVAPORATORS

The capital investment required to implement reduced energy consumption in existing evaporators varies considerably. In most cases the energy cost savings will increase directly with the amount of capital invested. In some cases, such as improved procedures for operation and maintenance of an existing evaporator, sizable cost and energy reductions can be achieved at low capital costs with very little hardware modification.

The risks of lowering evaporator reliability and product quality by upgrading the system are minimal if the appropriate technical evaluation is made.

Basically the process engineer has three general types of projects for modifying existing evaporators.

- Fine tuning existing evaporator: These low-investment changes do not change the basic evaporator layout or operation. Examples: increased attention to preventing water leakage into the process, improved maintenance, or additional insulation.

- Modifying auxiliary hardware: These moderate-investment changes normally can be authorized at the plant level. Example: improved heat recovery from condensates and product streams.

- Major hardware modifications: These high-investment changes normally require approval at the corporate level. Examples: installing additional effects or adding a vapor-recompression system.

Any of these changes can show very high returns on investment and expended technical efforts.

Fine Tuning Existing Systems

Unless the performance of the existing system is known, the savings obtainable through modifications cannot be accurately evaluated. Studying the existing plant normally can lead to immediate short-term cost reductions and improved operating efficiencies.

The starting point for the upgrading analysis is an accurate determination of the theoretical and actual heat and mass balances, together with an assessment of the proper operation of all internal controls. A discussion of the major points to check during this analysis follows:

Venting Rates. Noncondensibles enter the system with the feed or by system leaks. These noncondensibles must be vented at a rate that is sufficient to maintain pressure control and heat transfer. If venting rates are too low, the loss of heat transfer must be recovered by additional steam at a higher pressure to increase the temperature differences for heat transfer. However, excessive venting of the noncondensibles does not improve operation; high venting rates will require additional heating steam to offset the losses from vapors vented to the atmosphere or condenser.

Water Leakage. When water from external sources enters the process side of the evaporator, thermal efficiency decreases since such leaks dilute the product and increase the amount of water to be evaporated. Possible sources of water leakage into the process are corroded heat-exchange surfaces, pump seal flushes, barometric condenser water flooding, condenser flushes, and leaking of the water valves used to clean out the evaporator for startup and shutdown.

Operating Pressures. The proper operating pressures within single-effect and multieffect evaporators must be maintained to enhance thermal efficiency and capacity. The pressures can be affected by loss of vacuum and line restrictions. As frictional pressure drops or vapor pressure within each effect increases, the boiling point of the liquor is higher and additional steam at a higher pressure is needed to maintain capacity and the achievable heat-transfer ΔT 's of each effect.

Poor steam quality* can reduce the performance of the steam jets used to vent the noncondensibles from the evaporator system. Also, steam quality can cause erroneous readings of steam rates during the evaluation of the system and can indicate poorer efficiencies than are actually experienced. Air leaks will overload the noncondensable venting system; this reduces the vacuum capability of both mechanical pumps and steam jets and increases the system pressures. In every case an increase in pressure will reduce the ΔT . The ΔT loss is overcome by increasing the steam pressure and rate to maintain capacity.

Fouling. Fouling of the evaporator heat-transfer surfaces on the process or condensing-vapor side decreases heat-transfer rates and increases pressure drops. Reduced heat-transfer rates cause loss of capacity that can be overcome only at the expense of higher steam rates at higher pressure. This increased pressure causes poorer steam economy by increasing condensate heat losses. Systems exhibiting a high fouling frequency also will require additional energy to "boil out" scale from the heat-transfer surface. A higher constant power requirement for auxiliary operations per ton of concentrated product also occurs as the capacity drops.

Separator Efficiencies. If vapor/liquid separation efficiency is poor, the product will be carried over into the condensate of the following effect. If solids are carried up into the wire mesh of demister-type separators, a high pressure drop and loss in ΔT result. Recovered-product yield and the capacity are then reduced, and the pollution load increases. The reduction in yield wastes all the energy previously expended to produce the lost product and increases the costs of raw material. If scaling materials, such as tars, salts, or oils, are entrained, fouling is accelerated.

Radiation and Convection Losses. Heat losses to the surroundings are a common source of reduced efficiency in

energy use. These losses can be decreased by further insulating the system or by repairing existing insulation. The effects of the losses show up as a reduction of intermediate vapor and liquor temperatures and separator or vapor-line reflux. In operation this problem directly affects steam consumption and is overcome by increasing steam rates and pressure in the first-effect evaporator.

Chapter 2 is devoted to a more detailed discussion of calculation and evaluation methods for locating and correcting these problem areas in existing evaporators.

Modifying Auxiliary Hardware

These hardware modifications normally can be applied without changing the basic flow scheme or operation of the existing evaporator, but they require some capital expenditures. The most important of these are

- Heat recovery.
- Condensate reuse.
- Improved instrumentation and control.

Heat Recovery. Figures 2 and 3 show the usual points of heat loss through existing vapors and liquids and the streams where low-level heat can be used to provide increased energy efficiency. An evaporation system normally loses some energy through the hot concentrated product, condensates from the evaporator steam chests, final-effect vapor, or steam-jet exhaust streams. The decision on whether to use heat recovery is usually made by a straightforward balancing of energy cost savings vs. the capital and increased operating costs for the heat-exchange equipment.

The arrangement of the optimum recovery scheme requires the investigation of several options. Most heat-recovery applications will result in higher feed temperature. The amount of thermal energy required in a single-effect or multieffect evaporator to raise the feed to its boiling point is therefore reduced. The material balance, vaporization rates, liquid concentrations, heat-transfer efficiencies, and temperature differentials remain unchanged. The steam rate to the first effect is decreased since less energy is required to bring the cold feed to its initial boiling point.

A good heat-recovery scheme will allow the recycling low-availability waste heat to replace a portion of valuable high-energy steam. Any areas in the existing system which reduce temperature or condense vapor with cooling water, such as barometric condensers and product coolers, have heat-recovery potentials. The proper arrangements of the feed-liquid sequence can sometimes enhance heat recovery with extra exchangers.

Condensate Reuse. Steam can be generated by flashing a high-pressure condensate to a lower pressure to preheat feed, to supply steam to other effects, or to be used in another section of the plant. Flash tanks are used for this

*Steam quality is a measure of the amount of water present in the steam.

Figure 5. Vapor recompression methods: mechanical recompression and thermal recompression.

purpose. The applicability of this approach depends primarily on the existing condensate return conditions to the steam boiler and on an economic comparison with heat-recovery methods described earlier.

Instrumentation and Control. Controllers reduce the problems associated with operational errors and maintain process conditions for optimum performance. An evaporator must be instrumented adequately to provide the data necessary for evaluating the system's performance and to provide indicators for identifying problems affecting thermal efficiencies. Pressure, temperature, and flow indicators and possibly alarms must be incorporated at critical areas, such as the first-stage condensate discharge line and evaporator vapor lines.

Chapter 3 discusses the economics, calculations, and hardware necessary to evaluate the application of these modifications to a specific system.

Major Hardware Modifications

The techniques discussed earlier for upgrading evaporators do not involve significant retrofitting or operating changes to the existing evaporator system. The temperatures and pressures within each effect remain nearly constant. For significant reductions in steam consumption, more extensive modifications usually must be made, and

these technologies will change the process conditions. The selection of the optimum approach for these methods becomes somewhat more complex. The options available for radically reducing the steam consumption involve either the recompression of the overhead vapors for reuse or the use of these vapors in additional effects. Often a combination of these two approaches becomes the best option to minimize capital costs. Frequently the method may be selected because the space in the plant available for the modification is limited or because retrofitting costs are much higher than the cost of the hardware. Therefore the selection of the major hardware modification must include an evaluation of the changes in the process conditions and a determination of both the retrofit costs and feasibility.

Vapor Recompression. The vapor exiting the evaporator contains significant heating value as latent heat of vaporization. This energy can be recovered by heat-recovery techniques or multieffect evaporation. The vapor can also be reused in the same unit if the temperature and pressure of the vapor stream are increased to the conditions existing at the time in the steam chests. The methods for reusing a fraction of this vapor are

- Thermal recompression.
- Mechanical recompression.

These methods are shown in Fig. 5.

• **Thermal Recompression.** Of the two vapor-recompression methods, thermal recompression requires less capital but yields lower heat recovery than does mechanical recompression. A steam-jet booster is used to recompress a fraction of the vapor leaving the evaporator so that the pressure and temperature are raised (Fig. 5). Thermal recompression is normally applied to the first effect on existing evaporators or where the conditions are right for the application in single-effect units.

The thermal energy savings depend on the available motive-steam pressure and the compression ratio.* From steam-jet performance curves, the ratio of suction-vapor to motive-steam flow rates is obtained and matched with the total steam requirements for the evaporator. The difference in the flow rates between the existing evaporator steam without thermal recompression and motive steam is the steam savings. The motive-steam rate is the new steam consumption. The condenser water rate is also reduced because the vapor rate to the condenser is decreased.

A steam-jet thermocompressor is employed when low-pressure steam is used for the evaporation and the available steam pressure is high. The jet compressor essentially replaces the existing pressure regulator or reducing valve that controls the steam pressure (and temperature) for evaporation and can be applied easily to single-effect and multieffect units under the right conditions.

The variability in energy savings and capital costs still depends on the existing evaporator conditions. For example, Table 2 shows the process conditions for two single-effect evaporators, both operating at atmospheric pressure. One evaporator uses 5-psig steam in the steam chest, whereas the other uses 12-psig steam. The energy savings realized by reducing 100-psig steam with a thermo-

compressor to the required chest pressure are 56.7% for 5 psig and 33.3% for 12 psig. Although the evaporation rates and process conditions are identical for each case, the need for different steam pressures to maintain the evaporation rate reduced the steam savings for the evaporator with the highest compression ratio. If the condensate had been returned to the steam boiler, excessive amounts of carryover from the evaporator could prevent the continuation of that practice and reduce the total energy savings since part of the process vapor is being returned to the steam chest.

For existing multieffect units in which the total amount of first-effect vapor is now used in the second effect, the choice and location of the steam-jet thermocompressor are more complicated because the heat balances and heat-transfer rates are affected for each evaporator section. The economics may still be favorable to justify the technical evaluation and modification. Since the limitations of steam-jet thermocompressors are a compression ratio below 1.8† and new heat-transfer rates, the first effect normally becomes the best location.

• **Mechanical Recompression.** In most mechanical recompression operations, all the vapor leaving the evaporator is compressed. The compressors include positive-displacement, axial-flow, or radial-flow (centrifugal) types that can be driven by an electric motor, steam turbine, or gas engine. Under the proper conditions the mechanical energy required by the compressor is low compared to the recoverable energy in the vapor. Compressing steam at 5 psig to 20 psig (a compression ratio of 1.8), for example, requires a theoretical isentropic energy input of 45 Btu/lb (approximately 65 Btu/lb when aerodynamic and mechanical efficiencies are taken into account) to recover 980 Btu/lb. Under optimum conditions the equivalent of up to 15 effects can be achieved by mechanical recompression. This high energy recovery must be balanced against the capital, maintenance, and electrical-energy costs for the compressor. The vapor leaving the compressor is superheated, and sometimes a vapor desuperheater is installed to maintain heat-transfer rates before the compressed vapor is fed into the heat exchanger.

If high-pressure steam is available and a demand exists for low-pressure exhaust steam, as is the normal case for multieffect units, a steam-turbine-driven compressor drastically reduces the operating costs of the compressor as compared to those for electrically driven units. In this case the turbine exhaust can be passed on to the next effect of an existing multieffect unit. If electrical costs are low, the choice based on operating costs would be an electrically driven compressor. The total energy consumption is nearly identical with either drive.

Table 2. Effects of Existing Evaporator Conditions on Upgrading by Thermal Recompression

Process conditions	Single-effect evaporators	
	1	2
Existing Evaporator Conditions		
Evaporator feed rate, lb/hr	15,000	15,000
Evaporator capacity, lb/hr	10,000	10,000
Evaporator pressure, psig	0	0
Steam-chest pressure, psig	5	12
Steam rates (approx.), lb/hr	12,000	12,000
Vapor rate to condenser, lb/hr	10,000	10,000
Thermocompression Application		
Compression ratio	1.3	1.8
Motive-steam rate at 100 psig, lb/hr	5,200	8,000
Vapor rate to condenser, lb/hr	3,200	6,000
Steam savings, lb/hr	6,800	4,000
Steam savings, %	56.7	33.3

*Compression ratio is the ratio of the absolute pressure of the steam chest to that of the vapor leaving the evaporator body.

†At compression ratios above 1.8, the jets reach a critical operation where steam-jet instability and pressure fluctuation occur with minor changes in conditions.

The use of mechanical recompression becomes more costly as higher compression ratios are required because both the capital costs and the operating costs increase rapidly. Since a vapor-recompression evaporator normally operates with a low ΔT between the condensing compressed vapor and the boiling liquid in the vapor body, a large heat-exchange surface is required. Most older evaporators, particularly single-effect units, do not have sufficient heat-transfer area to allow this modification unless the existing compression ratio (steam chest to vapor pressure) is below 1.5 to 2.0. The existing vapor capacity of the unit is also a prime factor to consider in evaluating the economic benefits of mechanical compressors.

The need for expensive construction materials also will limit the application of mechanical recompression. A large rise in boiling point, corrosive vapor, or a tendency by the liquor to foam also work against vapor recompression. For concentration of dilute streams, the initial preconcentration step is a likely candidate for mechanical recompression because boiling-point rise is usually low in this first stage. If boiling-point rise is not significant, a multieffect evaporator converted to parallel operation with mechanical recompression also becomes a possibility.

The thorough evaluation of vapor-recompression options is an important part of any upgrading program. In Chapter 4 the economics and applications of vapor recompression are discussed.

Installing Additional Effects. When most existing evaporation systems were selected, one of the major design evaluations was determining the optimum number of effects. Much of the data may still be available, and these can be updated and used. The initial choice was probably based on economics and capital availability, and, in some cases, the physical size of the systems also may have influenced the decision. The addition of effects sometimes can be used to increase production capacity as well as to reduce energy consumption, and thus there is a dual purpose for studying this option.

Figure 6 clearly shows the change in the energy cost incentive to install additional effects. The steam costs to evaporate 1000 gal of water are plotted against the number of effects for the older steam costs at \$1/1000 lb and for \$3/1000 lb. The amount of savings obtainable by adding additional effects is changed strikingly by the new energy costs.

The addition of effects also eventually runs into economic limitations, even at high energy prices. If the initial steam pressure and final condenser pressure are set by other processing considerations, the allowable ΔT between condensing vapor and boiling liquid decreases in proportion to the number of effects. As the average ΔT decreases, the area and therefore capital cost increase. This economic balance then must be met.

Figure 6. Energy costs for evaporation as a function of energy prices and number of effects.

The best location within the evaporation scheme for adding effects will depend on the capacity objectives, the existing heat-transfer areas, and the projected heat-transfer area required for the additional units. When the existing evaporator heat-transfer areas are equal, the location of the additional effects is still important. When the evaporator areas are not equal or when the boiling point and viscosities of solution are important factors in design, the projected heat-transfer coefficients for each effect in the revised system must be calculated according to the new concentration gradients and temperature differentials. Except in cases of heat-sensitive materials or physical-area restrictions, the additional stages can be designed into the system from the beginning to the end of the existing layout. The consideration of additional effects may require extensive calculations if done without the aid of a computer. Detailed guidance in these calculations and generalized curves that allow an early approximation of feasible conditions and preliminary economics is given in Chapter 4.

COMBINATIONS OF ADDITIONAL EFFECTS AND VAPOR RECOMPRESSION

A combination of additional effects and vapor recompression often will give an optimum configuration. Thermal recompression and one additional effect, for example, can be combined to achieve energy savings nearly equivalent to the addition of two effects but at a much lower capital cost. The usefulness of combination schemes depends on the properties of the liquid being concentrated, the expected energy prices, and the level of energy available at the plant site.

ECONOMIC EFFECTS OF EVAPORATOR IMPROVEMENTS

The actual energy savings and operating cost reductions will depend on the operating conditions of the existing evaporator, as well as on the upgrading techniques. Each upgrading technique also has a range of energy-saving potentials. Table 3 shows some typical energy savings that can normally be achieved.

Table 3. Typical Energy Savings

Method	Capital requirement	Achievable energy savings, %
Venting and thermal insulation	Low	5
Improved maintenance	Low	5
Heat-recovery exchangers	Low	10
Condensate reuse	Low	5
Thermal recompression	Medium	45
Mechanical recompression	High	70 to 90
Additional effects*	High	$\left[1 - \left(\frac{N}{N+n}\right)\right] 100\%$

*N = original number of effects; n = number of added effects.
Example: The savings for increasing a three-effect evaporator to four effects is equal to $\left[1 - \left(\frac{3}{3+1}\right)\right] 100\% = 25\%$ approximately.

The dependency of energy savings on the existing evaporator system can be shown by considering the installation of additional evaporator effects. Adding two effects to a plant with an existing two-effect evaporator evaporating 20,000 lb water/hr can reduce the energy requirements by approximately 50% for a savings of 6250 lb steam/hr. If two effects are added to an existing four-effect unit with the same evaporator duty, the energy savings will be only about 33%, or slightly over 2000 lb steam/hr.

Figure 7 illustrates the operating cost savings for a range of energy savings from 3% to 25% and for steam costs of

Figure 7. Annual savings in steam cost realized by reducing existing evaporator heat loads.

\$1.50 to \$3/1000 lb. A very low improvement of 3% for an evaporator with a heat load of 60×10^6 Btu/hr would achieve a savings of \$50,000/year with steam costs of \$3/1000 lb.

For improvements in operating costs and energy consumption exceeding 5%, some additional investment usually will be required. Table 4 shows the results of upgrading the two-effect evaporator, which was the basis for the costs shown in Table 1. For this example two cases are presented: installation of a heat-recovery exchanger alone and installation of a heat-recovery exchanger in combination with one additional effect. Since no additional capacity was required, the primary objective is to reduce the operating costs associated with steam use. The installation of the heat-recovery unit alone reduces the operating costs by \$59,500/year, and the return on that investment before taxes is 198%. When the revision includes both modifications, the total steam savings are 11,800 lb/hr, which reduces the operating costs by \$231,100 and gives a return on total installed investment of 79.7%.

The investment that can be justified to reach various levels of energy savings depends on the net amount of energy saved and the corporate requirements for return on invested capital. Figure 8 shows this relationship for returns on investment (ROI's) of 10, 30, and 50%. For a known energy savings, the primary factor in the economic evaluation is the projected energy costs. Given a constant reduction by an upgrading of 15×10^6 Btu/hr and a 30% ROI requirement, the justified investment at steam costs of \$1.50/1000 lb is \$400,000. At \$3/1000 lb of steam, \$800,000 can be justified. The cost of equipment has risen approximately 80% (according to the Marshall and Swift index), and construction costs have increased at a similar

Table 4. Operating Cost Reductions by Upgrading Double-Effect Evaporator

	Existing evaporator (1976 prices)	Upgraded evaporator (1976 prices)	
		With heat exchanger only	With heat exchanger and one effect
Total installed investment in 1967: \$340,000*			
Added installed investment		\$ 30,000	\$290,000
Evaporation rate: 50,000 lb water/hr			
Manufacturing costs (annual)			
Operating labor, 2 man-years at \$6.80/hr	\$ 28,300	\$ 28,300	\$ 28,300
Supervisor, 0.5 man-year at \$8.80/hr	9,100	9,100	9,100
Maintenance (at 6% of investment)†	36,000	37,800	53,400
General plant overhead‡	28,500	27,200	32,500
Raw materials			
Depreciation (at 10% of investment)	34,000	37,000	63,000
Taxes (at 1% of investment)	3,400	3,700	6,300
Utility costs (annual)			
Steam at \$3/1000 lb	780,000 at 33,000 lb/hr	716,700 at 30,300 lb/hr	500,300 at 21,200 lb/hr
Cooling water at 4¢/1000 gal	28,400 at 1500 gpm	28,400 at 1500 gpm	19,700 at 1040 gpm
Electricity at 2¢/kWh	11,000 at 70 kWh	11,000 at 70 kWh	15,000 at 95 kWh
Total operating costs excluding raw material costs	\$958,700	\$899,200	\$727,600
Percent decrease in steam rate		8.1	35.8
Percent decrease in total costs		6.0	24.1
Percent return on increased investment		198	79.7
Operating savings from existing evaporator		59,500/year	\$231,100/year

*Projected investment for double-effect evaporator in 1976, \$475,000.

†Maintenance labor equals 37% of maintenance costs; return on investment is before tax.

‡General plant overhead equals 58% of operating labor, supervision, and maintenance labor.

Figure 8. Justified investment as a function of energy savings and corporate requirements for return on invested capital. Basis: 75% energy-generating efficiency; steam at 880 Btu/lb equivalent; 90% on-stream time; return on investment before tax.

rate from 1967 to 1976. When compared to a 300% rise in energy costs, the additional capital investment does, in fact, achieve greatly improved energy reductions and operating costs at acceptable returns on investment.

SUMMARY

For each upgrading method the objective is to minimize the amount of energy required to concentrate the product and to accomplish the objective without decreasing product qualities or on-stream times. The major areas for the corrective actions have been introduced and the remaining chapters of this report will provide the basic tools that are required to evaluate the preliminary economics and then to select the best methods for upgrading the evaporators for a particular plant and application.

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Chapter 2

CORRECTABLE OPERATING FACTORS OF EXISTING EVAPORATORS

INTRODUCTION

Although an existing evaporator may be producing adequate product at an acceptable rate, its energy efficiency may still be much lower than necessary. Production rates are usually maintained by increasing the steam rates and pressure to overcome fouling or other problems at the expense of additional operating costs, which are mainly due to increased energy use. One of the first steps in improving the operating costs and thermal efficiency of an evaporator is to conduct a thorough analysis of the existing plant. A thorough plant analysis will (1) determine the best operating procedures and conditions for the existing plant; (2) identify the areas for which corrections are needed; (3) point out short-term savings via energy and yield improvements; and (4) develop an accurate data base for the future evaluation of hardware modifications to the existing plant to reduce energy requirements and possibly to increase capacity.

In evaluating the existing unit, the engineer is usually faced with working around production schedules and a lack of instrumentation to obtain all the necessary operating data. However, in the majority of cases, the benefits from this study will justify the technical effort and expense necessary to obtain the correct data base to upgrade the evaporator and to determine the optimum conditions. The goals of this analysis are

- Short-term energy savings and cost reductions.
- Improved operating performance.
- Improved system understanding.
- Firm economic and technical basis for planning future programs and for selecting system modifications.

In this chapter a general procedure for conducting an analysis of the existing evaporator is described. The major sources of energy inefficiencies in existing evaporators, the effects of these problems on the operating costs of the existing units, and suggestions for identifying the problems are presented.

There is no set procedure to follow in this evaluation. Each evaporator is different and must be evaluated independently. The ingenuity of the engineering staff will dictate the exact steps that should be followed. It is

critically important to remember that poor basic data can lead only to erroneous conclusions and the possibility of mistakes in the selection of upgrading techniques.

PERFORMANCE EVALUATIONS*

Preparation

The first part of the analysis requires that the engineer thoroughly evaluate the data presently available from the existing plant and determine what data are required to conduct his initial evaluation and calculations. Good starting points are the process and instrumentation diagrams, the general arrangement diagrams, the maintenance manuals and instrument specification sheets supplied by the vendor initially, and the original design material balances. In some cases the unit may be so old that this material is no longer available. If the material can be obtained, the first step is to compare the original design with the existing system. If the plant has been in operation for some time, the conditions and plant layout may be considerably different from those indicated on the documents. The locations and available points for installing new instruments will, however, be shown and, upon confirmation in the field, will enable the engineer to plan the installation of the instrumentation necessary to conduct the study. Newer units also may lack the information necessary for this analysis.

Any data on the operation of the evaporators should be collected, and a daily or weekly average of the major variables should be calculated. This would uncover trends relating to on-stream time, cleaning cycles (time between startup and cleaning out), and any variation in the process conditions which is associated with seasonal variations, flow rates, or composition changes. An estimate of the cleaning cycle time, frequency, and procedure should also be reviewed. This type of data analysis provides the engineer conducting the study with initial identification of some

*Supplementary reference: American Institute of Chemical Engineers, *Equipment Testing Procedure—Evaporators*, New York, 1961.

problem areas and the trends that occur when process conditions are changed. Such an analysis also indicates the timing between cleaning cycles and permits analysis of those losses. It is important to realize that the data may show real changes in operating conditions or may be the result of poor instrument readings, calibration, or accuracy.

Physical-property data also must be assembled. This information can be obtained from laboratory data or from literature references. The accuracy of the laboratory analysis for determining stream compositions and other values should be noted since this will affect the accuracy of the final calculations. The physical-property data should include specific heats, vapor-pressure curves, boiling-point curves, and, if required, heats of dilution and crystallization.

Accuracy of the existing instrument and physical specifications must be verified. The actual locations of instrumentation should be checked in the field to determine if any changes have been made to the original installation. The evaporator heat-transfer areas should be checked to determine if any modifications have been made, such as tubes which have been plugged. In most cases additional instrumentation will be needed to replace inaccurate or faulty instruments. It is usual to use one indicator for several measuring points to reduce the costs of adding instruments for the first evaluation. For instance, in pressure measurements, lines can be run to one manometer to obtain very accurate data, or thermocouples can be run to one indicator.

The engineer prepares for the first balance run by ensuring that enough data can be taken while at the same time developing an understanding of the existing evaporator. The operating conditions in the evaporator should be maintained constant as this preparation is being completed.

The data requirements will vary with the type of evaporator. In this discussion the system is assumed to be a continuous, single-effect or multieffect forward-feed evaporator. The calculation method will be developed to enable the analysis of evaporators both for cases in which the boiling-point rise is negligible and for those in which it is a factor.

In summary, preparation for the analysis requires the following support data and planning:

- Knowledge of system layout and design:
 1. Separator size and internal design.
 2. Heat-transfer areas per effect.
 3. Vapor piping size and equivalent lengths.
 4. Location of instruments for readings and a check of instrument accuracy.
 5. Location of all process and utility lines affecting evaporator performance or operation.
- Planning the operating time for balance runs.
- Physical-property data:
 1. Liquor enthalpy/concentration data (or specific-heat data).

2. Boiling-point curves.
3. Viscosities.
4. Steam tables.

Calculation of Performance

After collecting the background data, the engineer is ready to determine what the thermal economy of the unit should be with the present design. This first step is a determination of the steam requirements and process conditions under which the evaporator would be performing under ideal conditions without operating problems. Fortunately this can be done with the following minimum amount of data:

- Steam pressure in the first effect.
- Feed temperature, flow, and composition.
- Pressures in each effect (or liquor compositions and temperatures).
- Product temperature, flow, and composition.

After the steam requirements and process conditions are calculated, further measurements will help to identify the location and causes of energy losses as shown by differences with the ideal conditions.

The ideal conditions and complete heat and material balances are made on the basis of the following assumptions:

- Steam dry and saturated.
- No radiation—convection losses.
- Negligible vapor-pressure drop between effects.
- Negligible subcooling of condensate.
- No vaporization or entrainment losses from separators (bodies).
- Negligible vent losses.

Examples of such calculations are illustrated in the section "Calculations" at the end of Chapter 4.

Figure 9 shows the recommended data inputs as they are located for a triple-effect evaporator with feed/product heat recovery, vapor bleeds off the first two effects, and condensate flash tank.

The data are collected from locations that exert influence on the steam consumption. Therefore it is best to delete any auxiliary hardware from the initial calculation of the evaporator steam consumption. An example of this approach is illustrated in Fig. 9 with the calculation boundary. The data input and methods for measuring the data are summarized in Table 5.

By isolating the major hardware that has a more direct effect on the steam consumption in this manner, the engineer can calculate the theoretical steam and water requirements more easily. This system is fairly complex since it has flash tanks, vapor bleeds, and a heat-recovery exchanger. Although the bleed rates and vapors from the flash tank are not inside the calculation boundary, they

directly affect the steam consumption within the boundary, and thus these values must be approximated. The flash-tank steam rates are approximated from Fig. 41 in Chapter 3. Bleed rates are more difficult to estimate and would require a direct measurement of the condensate at the point where the vapors are used. Since the feed temperature entering the first effect from the heat-recovery exchanger affects the

steam rate, measurements of the feed temperature as it enters the first effect would eliminate an evaluation of the exchanger.

Many different approaches can be taken to complete the calculation of the heat and material balances. In each approach a set of equations must be used for each effect, as shown in Fig. 10.

Table 5. Required Data Input, Measurement Methods, and Accuracy for Calculating Evaporator Steam Requirements

Data Input	Examples of measurement methods	Normal accuracy range, ± %
Feed rate, lb/hr	Storage tank levels, flow meters, weigh tanks	0.2 to 0.5
Feed composition, % solids	Laboratory methods	0.5 to 1.0
Product composition, % solids	Laboratory methods	0.5 to 1.0
Pressure, psig or in. Hg vacuum	Pressure gauges, manometers, recorders and indicating controllers	0.5 to 1.0% of full scale
Temperature, °F	Indicating thermometers, recording thermometers (thermocouples, resistance)	0.5 to 1.0
Heat-transfer area, sq ft	Number of tubes, size and length; vendor data	1 to 3
Flash tank rates,* lb/hr	Flash = $\frac{(\text{condensate in})(\text{temp. in} - \text{temp. out})}{\text{latent heat of the vapor}}$	5

*See also Fig. 41 in Chapter 3.

Figure 9. Data requirements and locations for determining plant thermal efficiencies.

Figure 10. Heat and material balances per effect.

HEAT BALANCES

Enthalpy Basis (most accurate)

Basic Equation

1. $Q_{V_1} = V_1 (H_V - h_C)$
2. $Q_{L_1} = L_1 (h_1)$
3. $Q_{L_2} = L_2 (h_2)$
4. $Q_{V_2} = V_2 (H_{V_2}) = (V_2 + B_2) (H_{V_2})$
5. $Q_{V_1} + Q_{L_1} = \text{total heat in}$
6. $Q_{V_2} + Q_{L_2} = \text{total heat out}$
7. $Q_{V_1} + Q_{L_1} = Q_{V_2} + Q_{L_2}$

Description

H = enthalpy of vapors
h = enthalpy of liquids (condensate)

1. Heat in with vapors into steam chest (steam tables at P_1).
2. Heat in with feed to second effect; (enthalpy of liquor at concentration X_1 and T_1) $\times$ rate.
3. Heat out with liquor to third effect; (enthalpy of liquor at concentration X_2 and T_2) $\times$ rate.
4. Heat out with vapor (enthalpy of vapor at P_2 and T_2 , steam tables)

Note: Flash tank adds to heat in equal to the flash rate according to Eq. 1 at P_1 .

Specific Heat Basis (approximate)

Basic Equation

1. $Q_{V_1} = V_1 \lambda$
2. $Q_{L_2} = L_1 (T_1 - T_2) c_{p2}$
3. $Q_{V_2} = V_2 \lambda = (B_2 + V_2) \lambda$
4. $Q_{V_1} + Q_{L_1} = Q_{V_2}$

Description

1. Q_V is heat into steam chest (Btu/hr), V_1 is vapor from first effect, and λ is latent heat of vaporization at P_1 (steam tables).
2. Q_{L_1} is heat with liquor, liquor rate $\times$ specific heat $\times$ temperature differences.
3. Q_{V_2} is heat out with vapors; λ obtained from steam tables at P_2 .
4. Total heat-balance equation.

Mass Balances

Basic Equation

1. $L_1 - L_2 = V_2 + B_2 = V_2'$
2. $X_1 L_1 = X_2 L_2$ (no solids vaporized)
3. $V_2 + B_2 = L_1 - (X_1/X_2) L_1$
4. $L_1 = F - V_1$

Description

1. Difference in liquor = total evaporation.
2. Liquor solids in = liquor solids out.
3. Determine evaporation rate knowing X_1, X_2, L_1 .
4. F is known and L_1 is calculated from V_1 and previous heat balance on preceding effect.

Figure 11. Heat and material balance with a thermocompressor. The basic equations in Fig. 10 still apply.

Enthalpy Balance Changes

$$1. Q_S = (B_1 + M)(H_S - h_C) = S(H_S - h_C)$$

Additional Data Input

Supply steam pressure (P_0) to thermocompressor

Material Balance Changes

1. $B_1 + M = S$ = total steam flow into steam chest
2. Capacity ratio curves give ratio $(R) = B/M$ at P_S/P_1
3. $S = B(1 + 1/R)$ or $S = M(1 + R)$

The two basic methods often used to calculate the heat and material balances over an evaporation system are (1) solution by simultaneous algebraic equations and (2) trial-and-error solution. If the liquor has a negligible boiling-point rise or if the unit has one or two effects, use of simultaneous equations is preferable. If the boiling-point rise is significant, the data input is limited, or the arrangement is complex, a trial-and-error procedure is preferred in which a steam rate is assumed and the material and heat balances are calculated sequentially for each effect until the steam rate that gives the actual product composition is found.

The only hardware that complicates the calculations any further is the thermal recompressor. Although the use of thermocompressors is described in detail in Chapter 4, the significance of the calculation is briefly presented at this time. The calculational procedures will not change significantly. The thermocompressor is also treated as being

outside the imaginary calculation boundary, and the performance characteristics are obtained from vendor specifications. In this case the compression ratio is known (pressure of steam chest/suction pressure), and the motive-steam rate can be obtained from the jet capacity ratio curves which specify the rates of flow between the suction vapor and motive steam.

Figure 11 shows the effects of a thermocompressor on the heat and material balance equations. The basic formulas previously shown in Fig. 10 are used, and trial-and-error calculations are made by assuming variable steam rates until the product concentration is finally reached.

Once the steam rate and calculated process conditions for the existing evaporator under ideal conditions have been determined, the next step is to compare the results with actual conditions. This comparison may show that the theoretical steam rate is considerably less than that of the present unit and thus indicate a need for corrective action.

An evaporator should approach the calculated steam-consumption value plus approximately 2% to 3% to account for normal radiation losses. If the percentages are higher; the engineer can, by identifying the problems, achieve steam savings equal to (measured steam rate) - (calculated ideal steam rate) - (radiation loss as steam equivalent). These steam savings can be achieved without committing significant investment.

However, the measurement of the steam flow is critical in this analysis, since any differential above $\pm 2\%$ would be assumed to indicate some operational problem, and further investigations would be started. The steam rates can be measured by installing differential pressure meters on the steam line entering the system (accuracy $\pm 1\%$) or by measuring the rate of the cooled condensate. Another method would be to calculate the steam rate by taking accurate flow, temperature, and composition measurements for each effect.

Precautions

As discussed later in the section on cleaning cycles, the heat-transfer rate also affects the energy consumption for the evaporator. If the surfaces are fouled at the time of

the initial calculation, as would be indicated by low heat-transfer coefficients, the ultimate savings would be higher than those calculated. The heat-transfer coefficients for each effect are obtained by using the following standard formula:

$$U = \frac{Q}{A \Delta T}$$

where U = heat-transfer coefficient, Btu/(hr) ($^{\circ}\text{F}$) (sq ft)

Q = total heat transferred, Btu/hr

A = heat-transfer area, sq ft

ΔT = log mean temperature difference

For natural-circulation evaporators the ΔT , as shown in Fig. 12, equals steam temperature minus boiling liquor temperature in the body. The ΔT for forced-circulation units is the log mean temperature differences between the steam and the liquor entering and leaving the tubes.

These heat-transfer coefficients and the existing calculated process conditions should be compared with the original design values, which can be supplied by the evaporator vendor. If the heat-transfer coefficients are lower than those originally designed or if the process conditions are different, the coefficients indicate that the

Figure 12. Calculating the heat-transfer coefficient (U) from measured or calculated values.

Natural-Circulation Unit

Calculation for overall heat-transfer coefficient (U)

Q = total heat transferred per hour

$\Delta T = T_S - T_L$

$$U = \frac{Q}{A(T_S - T_L)}$$

Forced-Circulation Unit

Calculation for temperature difference

Q = total heat transferred per hour

$Q = UA \Delta T_{LMTD}$

$$\Delta T = \frac{(T_S - T_{L in}) - (T_S - T_{L out})}{\ln [(T_S - T_{L in}) / (T_S - T_{L out})]}$$

Figure 13. Heat and material balance for double-effect evaporator.

operation could be improved by revising cleaning cycles for the system. In any case the study to identify problem areas can continue without optimizing the cleaning cycle since the calculated thermal efficiencies are based on the present state of the heat-transfer surfaces.

Data Accuracy

The accuracy of the data used in the upgrading program is critically important. An excellent balance between the calculated steam flows and the measured rates would be $\pm 2\%$. Five percent is the maximum acceptable error. If the values fall within this range, the evaporator is running as well as can be expected with its existing design, and the study can continue to the next phase of the program.

Collecting data from an operating plant is never an easy task, and several attempts may be required to obtain accurate data. The importance of obtaining accurate data is probably best illustrated by showing the effects of incorrect data on the calculated value for steam rates. Figure 13 shows the heat and material balance results for a double-effect evaporator rated at 14,000-lb/hr evaporation, when the actual steam flow rate is 6590 lb/hr. The error contribution on the calculated steam rate based on various measurement errors of $\pm 10\%$ input is shown in Table 6. For example, a -10% error in the feed concentration gives a calculated steam rate of 8481 lb/hr, which is 1891 lb/hr higher than the actual rate. This type of measurement error,

either singly or combined with others, would lead to incorrect conclusions about the present performance. The investigator could erroneously assume that the system was operating satisfactorily and did not require further study when in reality the situation was the reverse. Therefore data collection by taking several points at a balanced condition and the repetition of several balances is almost always necessary.

Common sources of errors in steam-measurement rates are poor calibration of the instrument, faulty steam traps, condensate in the impulse lines to the instrument, and wet steam.

The quality of the steam and proper trapping of the condensate are particularly important for measuring actual steam rates being used to concentrate the product. A comparison of measured condensate rates and measured steam rates is a good way to learn whether steam quality is a problem. Also, if the traps are bypassing steam, the measured steam flow to the evaporator will be higher than the rates that were calculated. If the evaporator chest contains a level gauge and the volume of the steam chest is known, the condensate rates can be measured by drawing the level down and measuring the time to return. A low rate, as checked against the measured steam flow, would indicate vapor bypassing in the condensate line. Any water entering the steam chest because of low-quality steam will show up as a high condensate rate compared to the measured steam-flow rate.

Table 6. Effect of a $\pm 10\%$ Measurement Error on the Calculation of Steam Flow

Actual conditions	Potential errors in measurement	Calculated steam rate resulting from error in measurement			
		+10% Error		-10% Error	
		Rate, lb/hr	Error in rate, lb/hr	Rate, lb/hr	Error in rate, lb/hr
Feed: 50,000 lb/hr	Feed	7282	+692	5957	-633
Feed temperature: 182°F	Feed concentration	4732	-1858	8481	+1891
Feed concentration: 50%	Product concentration	8579	+1989	4395	-2195
Product concentration: 70%	Steam pressure*	6617	+27	6572	-18
Steam pressure: 19 psia	First-effect body pressure	6749	+159	6466	-124
First-effect body: 8 psia	Second-effect body pressure	6692	+102	6514	-76
Second-effect body: 2 psia					
Actual steam rate: 6590 lb/hr					

*Causes greater error in determination of heat-transfer coefficients since the saturated temperature at the measured pressure is used to calculate the log mean temperature difference.

Steam quality can also be measured by the throttling calorimeter procedure illustrated in Fig. 14. The steam quality* is obtained by measuring the exhaust temperature of the steam at atmospheric pressure and by using a Mollier chart to determine the quality by taking a constant enthalpy at atmospheric pressure of the steam pressure in the line.

CAUSES OF REDUCED ENERGY EFFICIENCY

The data obtained from the calculations outlined in the previous sections can be used to assist in troubleshooting the unit.

The major sources of energy losses due to operating factors are

- Venting rates.
- Radiation and convection losses.
- Poor vacuum system performance.
- Air leakage.
- Water leakage.
- Fouling—improper cleaning cycles (lack of maintenance of optimum heat-transfer coefficient).
- Separator efficiency.

In most cases the loss of efficiency is caused by a combination of factors. However, by a process of elimination and by understanding the effects of each problem on the operation of the plant, each problem can be separately analyzed and corrected. Generally, since the only control

the operator has is the heating supply, the steam pressure is raised to maintain capacity when deficiencies occur, and the production rate is regained at the loss of energy efficiency.

As each problem is corrected, the operating conditions of the unit may change. If this occurs, the balance and comparison of the steam rates should be repeated until the desired degree of accuracy is obtained.

Venting Rates

Noncondensibles enter the evaporator from inerts in the feed, decomposition reactions during the evaporation step or in the feed, and air leaks through piping networks. For atmospheric single-effect units without vapor recompression, the air leaks will have little effect on the heat-transfer rates. However, the heat-transfer and steam-consumption rates will be affected if the vacuum system is overloaded in the vacuum single-effect evaporators.

For multieffect units the last-effect pressure is controlled by the vacuum system. In the last effect, loss of vacuum will have a similar result on performance as in the single-effect evaporator. The noncondensibles from earlier effects, however, will enter the steam chest of the next effect. If the venting of the steam chest is inadequate, the noncondensibles accumulate and cause heat-transfer rates to drop. Additional steam at a higher pressure is then required to overcome the loss of heat transfer and pressure.† The concentration of noncondensibles necessary to reduce the overall heat-transfer rate can be quite low,

*Low-quality steam will also be reflected in poor steam-jet performance. If the steam quality is below 98%, steam purifiers should be installed in the lines to the jets.

†The loss of vacuum in each effect increases the boiling point of the liquor and will decrease the ΔT for heat transfer.

Figure 14. Method of determining steam quality. Example: If steam is supplied at 40 psia (PI), determine the moisture content of the steam. After a portion is passed through a throttling calorimeter, then exhausted to the atmosphere, the temperature (TI) is 220°F.

Locate 14.7 psia and 220°F on the Mollier diagram.

Draw a horizontal line from this point to intersect the pressure curve of the steam, 40 psia. (During the throttling process, the steam expands at constant enthalpy.)

Read the moisture content of the steam as approximately 2%.

less than 2%, particularly in evaporators in which the design is based on high overall coefficients or low ΔT 's.

If the venting rate in the steam chest is increased to take care of the higher amount of noncondensibles, thermal efficiency is further decreased. Since the noncondensibles are commingled with the steam, additional steam is lost to the vents. Any more venting than that required is also wasted since excessive venting does not increase heat-transfer rates, and the difference between adequate and excessive venting is not discernible. Since the concentration of inerts is difficult to measure, it is common to be on the safe side by adjusting the venting rates far above the amount that is necessary. This results in the loss of appreciable amounts of steam. The best plant check to prevent this is to reduce the venting rate until the capacity at a constant steam pressure begins to decrease. Then increase the rate slightly. The balance between capacity and energy losses is thus achieved by closer control without investing capital.

Figure 15a shows a typical venting system for a triple-effect evaporator. Since the cost of steam was low, in the past it was common practice to vent to the atmosphere

Figure 15. (a) Conventional venting to the atmosphere. (b) Stage venting protecting against excess venting in one effect.

when the body pressure exceeded the atmospheric pressure or to vent to the steam-ejector system in vacuum operations. The system normally included vent valves on which the operator could make periodic checks, and, when necessary, as the inert concentration rose, the pressure in the steam chest could be reduced by bleeding at a higher rate. This practice normally led to losses due to valve leakage or to the valves themselves being left open by mistake.

Series staging of the vent, from one effect to the next, can be installed to overcome this problem and to reduce the overall steam losses by limiting them to the venting area. In this manner the noncondensibles can be finally vented off the last effect, where the greatest amount of air is present and control is limited to one location. Therefore operator control of vent loss is easier because only one location must be controlled. Figure 15b shows one recommended venting procedure as the venting system in Fig. 15a is converted to series venting.

The importance of the venting rate to operating costs and to steam consumption is shown in Fig. 16 for a 100,000-lb/hr triple-effect evaporator. Under the assumed conditions, an excess venting rate of 150% over that necessary for correct control would result in an additional steam loss of 680 lb/hr, or the equivalent steam cost per year of \$16,000. At the old steam costs of 75 cents/1000 lb, the cost was \$4000/year; therefore at current steam costs the importance of closer control would lead to much greater savings without additional operating costs and with a minimum capital expense.

Figure 16. Effects of excess venting. Basis: Triple-effect evaporator; feed rate, 100,000 lb/hr at 100°F; dissolved air in feed, 30 ppm; leakage, 25 lb air/hr/effect; air in vented vapors, 5%. Base case = 0%. Curves valid for specific example only.

Figure 17. Venting steam rates at various steam-chest pressures and pressure drops per 10 ft of 1-in. and

$\frac{1}{2}$ -in. pipe. Note: The steam rate at pipe lengths other than 10 ft can be estimated by $\frac{S \text{ at } 10 \text{ ft}}{\sqrt{\text{desired length}/10 \text{ ft}}}$.

— — —, see discussion in text.

The maximum steam loss through the vent lines is a function of the pressure drop, as shown in Fig. 17. This plot can be used to estimate approximate steam savings for any plant for which the pipe sizes and lengths and pressures are known. When series venting is not used, the pressure drops from the steam chest to the vacuum source are higher. For example, with a 1-in. vent line and a steam-chest pressure of 12 psia, the steam flow to the vacuum line with a pressure of 2 psia can approach 675 lb/hr. However, staging of the vent system not only utilizes some of the steam in the next effect, but it also decreases the pressure drop and maximum steam flow that can be lost to the vacuum system. If the last-effect steam-chest pressure is 5 psia, the maximum loss for the entire system would be 220 lb/hr. For multieffect units with more vent lines, the losses increase. The series venting loss is always the last-effect line to the vacuum source. As the number of stages or the pressure drop increases, the savings cor-

respondingly increase. Correct sizing of the vent lines and the use of series vents will eliminate the possibility of energy loss and will minimize vent control problems.

When series venting is applied, any excess noncondensable flow causes some pressure buildup and loss of capacity. Therefore it is advisable to have an auxiliary line or to size the final vent line with an orifice that will give the desired pressure drop and to include a bypass valve to handle startups and emergency conditions when the inert level increases to the point that the capacity is reduced.

Radiation and Convection Losses

Radiation and convection heat losses are probably the most publicized factors that, if minimized, can improve energy efficiency. These losses are a function of the temperature (process and ambient), surface area, and insulation type and thickness. The major sources of heat

Figure 18. Heat losses from bare and insulated surfaces as a function of process temperature and ambient air temperature. — — —, see discussion in text.

loss which are easily identified will be bare pipes or wet insulation. The economics of insulation has been well publicized in vendor literature, although the applications to evaporators have not.

The effects of high radiation or convection losses are indicated by low temperature readings of the condensing vapors and liquor lines between each effect. Figure 18 shows heat losses from bare surface areas over a range of process temperatures that are typical for evaporation systems. The additional steam rate that would be required to offset heat losses for a single-effect evaporator would be the summation of the losses according to the temperature profile and the exposed surface area. These values can be estimated for the existing plant, and the steam rate required to account for the radiation losses can be estimated. In multieffect units, however, the total heat loss cannot be directly related to increased steam consumption. This additional heat input from steam is approximately 70% of the summation of the heat losses.*

*A good approximation of the additional steam required to make up radiation heat losses is to take 70% of the steam savings that can be estimated by the following formula:

$$\frac{\text{Total heat losses (Btu/hr) from Fig. 18}}{\text{Latent heat of vaporization of steam, Btu/lb}}$$

$$= \text{steam rate increase (lb/hr)}$$

As seen in Fig. 18, 1 in. of insulation for 100 sq ft of an existing exposed surface at 200°F and an ambient air temperature of 70°F would reduce the heat loss by 49,400 Btu/hr and would save approximately 420,000 lb of steam/year, which is a cost savings of \$1260/year at \$3/1000 lb steam for a single-effect unit.

The radiation and convection heat losses can be approximated from Fig. 18 for a multieffect unit by taking the sum of the losses from each effect. For a triple-effect evaporator with an assumed temperature profile and exposed area of 100 sq ft at 250°F in the first effect, 200 sq ft at 200°F in the second effect, and 100 sq ft at 140°F in the third effect, the total additional steam is 178 lb/hr at 10 psia, for first-effect chest pressure, at a cost of \$4300/year at a steam cost of \$3/1000 lb.

The operating cost savings per year as a function of heat loss and insulation thickness is shown in Fig. 19. An inch of insulation provides a significant cost savings; since the evaporators are usually operating at a low temperature, more insulation is usually not economical or required.

The cost savings derived by adequately insulating the system are considerably higher than those indicated when the insulation is wet or the ambient temperature is considerably lower. The important point is that 1 in. of weatherproofed insulation provides a significant reduction to the energy costs, and further insulation would not be economical.

Figure 19. Annual steam savings per hundred square feet of area with 1- and 3-in. insulation over a range of process temperatures for single-effect evaporator. Basis: Values from Fig. 18; ambient air temperature, 70°F; 7880 operating hr/year.

Vacuum System Performance

The vacuum system, whether it is a vacuum pump or a steam-ejector system, is important to the thermal efficiency of the operation and is itself an energy user. The vacuum generated by the system can be affected by poor condensing rates in the barometric condenser (direct contact) or surface condenser (promoted by high water temperature or low condensing water rate) and by the amount of air leakage. Both these factors affect the load to the vacuum system and cause either a rise or a fluctuation in the suction pressure. For systems using steam ejectors, the performance is also affected by the steam quality,* condition of the nozzles, and condenser flooding. All these factors can reduce the suction load capability of the jet system. In addition, mechanical wear on vacuum pumps can cause a reduction in suction load.

If the vacuum is reduced (evaporator pressure increases), the loss of ΔT is made up by raising the steam pressure (temperature rises), and the amount of steam required to maintain capacity is increased. The vacuum source is the primary control on the last effect and on the pressure profile for a multieffect unit. If there is not sufficient vacuum, the liquor temperature rises, and steam temperature must be increased to maintain the evaporation rate. This sensible heat rise and the loss of recoverable heat due to flashing require both a higher steam rate and a higher pressure to maintain capacity.

The operation of the vacuum pumps and steam ejectors can be evaluated by running at shut-off conditions and comparing the suction pressure with vendor performance

curves. If the pressure is higher than it was designed to be, the units should be inspected. If the exit temperature from the condenser cooling water is higher than it was designed to be, the amount of water vapor exiting the condenser to the vacuum system will be too high and exceed the design load. If all these factors are checked and found to be satisfactory, the problem is probably excessive air leakage, and the system should be checked for leaks.

The effect of loss of vacuum is illustrated in Fig. 20 for a single- and triple-effect evaporator. The pressure in the last effect for these examples is normally maintained at 25 in. Hg vacuum. For a combination of reasons, a 10% loss in vacuum (equivalent to 22.5 in. Hg vacuum pressure) would increase the steam rate by 2% for a single-effect evaporator and 5% for a triple-effect evaporator. This increase in pressure from a loss of vacuum is shown in Fig. 21 for a triple-effect unit rated at 30,000 lb/hr. A 50% increase in absolute condenser pressure would increase the operating steam rate by 5.6×10^6 lb/year, at an additional operating cost of \$16,800/year for \$3/1000 lb steam.

In many cases a different ejector system can usually be used in the wintertime, when much colder water is available for the condensers. Since the water temperature drops, the amount of vapor entering the air ejectors decreases. The ejectors, which have been designed to handle specific noncondensable loads at the maximum cooling water temperature in the summer, will perform better and pull a greater vacuum. At the same noncondensable load, the motive steam can be decreased and maintain the same vacuum as a function of the lower water vapor rate. Consultation with the equipment supplier after data on annual plant cooling water temperatures are collected may show how steam rates could be reduced to the degree where

*Quality should be above 98%.

Figure 20. Increase in steam consumption necessary to overcome loss in vacuum.

Figure 21. Increase in steam consumption to maintain evaporator capacity with increase in last-effect pressure.

Basis: Triple-effect 30,000-lb/hr evaporator with no heat recovery for product or condensates; evaporator feed at 180°. — — —, see discussion in text.

the operating savings would justify investment for modifying the ejector system for situations when colder water is available.

Water Leakage

The amount of water removed from the feed is called the net evaporation rate.* It is often discovered in controlled tests that additional water is being removed or

*The net evaporation rate (V) equals the feed rate minus the product rate. It can be calculated with the measured feed rate and the feed and product composition according to this formula:

$$\text{Feed rate} - \frac{(\text{concentration solids})(\text{feed rate})}{\text{product solids}} = \text{net evaporation rate}$$

Figure 22. Effects of water leakage on evaporator steam rates.

that the product concentration does not match the expected concentration from the ideal balance. The amount exceeding the net rate is often due to water leakage that dilutes the product. The water can enter the system from flush lines or condenser flooding or by the excessive rinsing of the system to prevent the buildup of scale or deposits in instrument lines, traps, or sightglasses. Water leakage is one of the most severe problems affecting energy consumption which can be corrected by closer operator attention and maintenance.

The effects of water leakage on the energy consumption are shown in Fig. 22, which is based on a flow rate per 10,000 lb of vapor being removed. For a triple-effect evaporator, a leak averaging 1 gpm per 10,000 lb evaporation can result in increased steam consumption of 2%, an increase which is necessary to remove the additional water added to the liquor. For a 50,000-lb/hr unit, this rate is 400 lb/hr, or an additional cost of \$9600/year at a steam cost of \$3/1000 lb. Since a portion of this vapor is recovered in the next effect of the multieffect evaporator, the additional steam required is lower than that for a single-effect unit for the same leakage rate.

The potential leakage rate can be very high, as illustrated in Fig. 23. Multiplying these rates by the number of potential leak sources results in an enormous potential for reduced energy economy. For example, a 1/2-in. valve inadvertently left open with 40-psig water back pressure can leak 72,000 gal/day into the body of an evaporator operating at 10 psia.

Finding the source of water leakage is difficult. The heat and material balances derived during the analysis previously described will help. If the heat-transfer coefficient profile is irregular, e.g., if the coefficient of one effect is too low, leakage may be indicated in that effect. Water leakage through a pump seal (which can approach 5 gpm) can be identified by analyzing the concentration of liquor from each effect and comparing it with the calculated value. Likewise, the heat balance for the condenser based on the calculated vapor rate, cooling water temperature, and cooling water ΔT may show that the water rate should be lower than measured, and this would indicate leakage into that effect or dilution from condenser flush lines, flooding, or leakage. Similar analyses and calculations can be made to eliminate all potential sources.

$$\text{Third-effect vapor rate (V}_3\text{)} = \frac{(\text{water flow to condenser, lb/hr}) (\text{specific heat of water}) (T_0 - T_1)}{(\text{enthalpy of steam at } P_3 T_3) + 32 - T_0}$$

Figure 25. Instrumentation to monitor rinse water.

The cost of thermal energy attributed to water leakage from excessive rinsing cannot be measured by these types of balance runs since the rinsing occurs at periodic intervals. When the balance is made to discover leaks at other sources, the effects of excessive rinsing or boilout are not occurring. Therefore a longer study with appropriate instrumentation must be made. The instrumentation to monitor the rinse water during the operating cycle, which would also detect leakage, is shown in Fig. 25. These water meters would measure the water rate consumed by rinsing. The meter could be installed at the main inlet line to each rinse line valve. The meter would also help detect water leakage at times other than during rinsing. The leakage at the pump seals would be observed by installing inexpensive purge-type rotameters. Any difference in flow rate would indicate failure of the seal and water leakage. If rinse cycles can be reduced and leakage points identified, energy savings can be obtained.

In summary, the only good method for reducing losses associated with water leakage is measurement. The leakage occurring at certain times cannot be detected until complete heat and material balances are made on the system with the described flow measurements. Likewise, the losses attributed to excessive rinsing cannot be calculated and will be reduced only by the combined efforts of the operators and the technical staff to measure rinse and to reduce it wherever possible.

Cleaning Cycles

Thermal losses caused by fouling of the heat-transfer surfaces occur because the loss in heat-transfer rate must be overcome by increasing the ΔT for heat transfer, an increase which uses higher pressure steam at a higher rate. The total

energy input to the evaporator to concentrate the material does not change; however, the amount of heat loss in the condensate does. If the condensate is being recycled to the steam generator or is being used elsewhere, the effects of using higher pressure steam are minimized. Reducing the energy losses due to fouling to the lowest possible degree becomes a matter of selecting the best cleaning frequency to minimize energy losses.

The steam rate, as shown in Fig. 26 for a specific design steam pressure of 10 psig, increases as the steam pressure is

Figure 26. Additional steam required to maintain constant evaporator heat load (10^6 Btu/hr) as steam-chest pressure increases. Basis: Supply steam, 10 psig; no condensate heat recovery. — — —, see discussion in text.

raised to maintain the constant evaporator heat load. If the absolute pressure to the steam chest is increased by 40%, e.g., from 25 psia to approximately 35 psia, the steam rate increases by 1.45% for each 10^6 Btu/hr heat load requirement. If the present heat load is 10×10^6 Btu/hr (equivalent to 10,500 lb steam/hr), the steam rate would increase by 152 lb/hr and the operating cost would be \$3590/year higher at a steam price of \$3/1000 lb.

Another example of the effect of the heat-transfer rate on the steam consumption is shown in Fig. 27. For a 30,000-lb/hr rate of evaporation in a triple-effect evaporator, the steam rate increases from 10,000 lb/hr to 11,500 lb/hr for a reduction in heat transfer of 30%, and the steam cost rises the equivalent of \$36,000/year for \$3/1000 lb steam.

The change in heat-transfer rates is a function of many dynamic variables within the system. The heat and material balances initially discussed to identify problem areas were made under a particular situation with a certain degree of fouling or scaling already present. Thermal savings by maintaining a higher heat-transfer rate could be achieved with increased maintenance and cleaning of fouled surfaces. One or two balances will not provide this information. A long-term study over one cleaning cycle with periodic evaluations of the heat-transfer rates during that cycle is required, and the effect of other process variables that affect heat transfer also must be minimized. The economics requires a balancing of steam savings against the additional production loss required to obtain more frequent cleanups, as shown in Fig. 28. The plant operating savings at a generally improved heat-transfer condition is the difference between the steam savings and the lost production capacity due to downtime plus the added labor and maintenance for more frequent cleaning.

Figure 27. Increases in steam consumption as heat-transfer coefficient drops and capacity is maintained by increasing steam pressure to first effect. Basis: Triple-effect 30,000-lb/hr evaporator with no heat recovery for product or condensates; evaporator feed at 180°F. — — —, see discussion in text.

Fouling is a function of time (see Fig. 28). The trends for two evaporators with different rates of fouling are illustrated. When the tubes are initially cleaned, the heat-transfer coefficient (U) is highest. When the fouling begins after startup, this coefficient reduces at a rapid rate to the design coefficient, which by experience was initially selected to match the process demands. The steam pressure is raised, and its consumption rises if extensive heat-recovery equipment is not present. Eventually, capacity

Figure 28. Effects of fouling on heat-transfer coefficients for two evaporators with different rates of fouling.

cannot be maintained, and the unit is cleaned, at which point the coefficients go back to their original values. The economical frequency of cleaning depends on the fouling rate and the additional steam required and can be calculated on that basis. After cleaning, the rate of fouling may build up to a somewhat constant value if scale is removed, owing to turbulence, at a rate similar to the rate of formation, as shown in curve A of Fig. 29. However, some evaporators can continue to foul at the same rate or a higher rate (curve B) as the unit operates.

Applying the data on Fig. 27 to a time sequence as in Fig. 30 illustrates the potential for additional cleaning cycles. It is assumed that the unit is run for 6 months at capacity until the heat-transfer coefficient reaches 50% of the clean coefficient. The assumed decreases in the heat-transfer coefficient for a 6-month interval are shown in Fig. 30a, and the new profile for cleaning the unit every 3 months is shown in Fig. 30b. By doubling the cleaning frequency, the resultant steam flow, as a function of time and U for each cleaning frequency, also is shown. The total steam uses for a 6-month operation are 8,000,000 lb for the single 6-month cycle and 4,400,000 lb for doubling the cleaning frequency. The operating cost savings at steam costs of \$3/1000 lb are \$21,600. At old steam costs of 75 cents/1000 lb, the energy savings were only \$5400. If the increased cost of the cleaning cycle (production loss plus cleaning costs) is considerably less than \$21,600, more frequent cleaning will be preferred. In most instances the production rate through the evaporator also can be slightly increased after the initial cleaning since the average rate of

Figure 29. Buildup of fouling resistance on evaporator heat-transfer surfaces.

heat transfer is higher for the time intervals of operation. This increase would help to compensate for the loss in production if the cleaning frequency is increased. Figure 31 indicates the total steam savings that can be achieved on the basis of this example.

Total steam consumption for six months (shaded area): ~8,000,000 lb steam

Total steam consumption for six months (shaded area): 4,400,000 lb steam

Total steam savings = 8,000,000 - 4,400,000 = 3,600,000 lb steam
Total steam savings per year at \$3/1000 lb = \$21,600
Total steam savings per year at 75 cents/1000 lb = \$5,400

Figure 30. Potential steam savings by doubling the cleaning frequency.

Figure 31. Steam savings that can be achieved with more frequent cleaning cycles.

For some plants an analysis of the potential energy savings by increased cleaning frequency does not apply since daily or shift cleaning and "boil outs" are required. The investigation should, then, turn to the amount of energy required to shut down, clean, and bring the system up to temperature for startup. Increasing the temperature of the cleaning water would increase the solubility of the scale and minimize "cool down" of the evaporator shell and tubes. Reducing the amount of cleaning water and the time for cleaning will also reduce the steam requirements.

Separator Efficiency

Cyclone separators separate the liquor and vapor by directing the two-phase vapor and liquor stream together around the body of the separator. As the velocity decreases, the vapor separates from the two-phase stream and enters the center of the cyclone where it rises to the vapor outlet. The liquor against the separator wall spins down and out the drop leg. The efficiency of the cyclone separator depends primarily on the mass velocity entering the separator (higher velocity usually improves separation efficiency), average mass velocity in the cyclone body, and the liquor droplet size and viscosities. Without a high inlet velocity, the liquor would not be spun against the wall of the separator, and the efficiency is lost. Cyclones seldom have any internal elements other than vanes or baffles. The pressure drop in the cyclone separator is due primarily to an acceleration pressure drop in the inlet—which occurs as the vapor expands and the liquor flashes to the pressure of the body—and also to frictional pressure drop.

A second type of unit is the coalescing separator. This design is based on a low superficial vapor velocity through the body, which reduces the chance of liquor entrainment, and a demister pad that removes any entrained liquid droplets being carried up with the vapor stream. The pressure drops through the demister pad are normally designed to be $\frac{1}{2}$ to 1 in. of water.

Decreased efficiency of the separator can promote entrainment of the product into the vapor line, which reduces yields and may cause coating or corrosion of the vapor piping. If the product or liquor contains tars, salts, or oils, a decrease in the heat-transfer coefficient is accelerated as these materials form a film on the vapor side of the tubes in the next effect. Pressure drops and losses in heat-transfer rate also are increased if any damage occurs to the inlet guide of the cyclone or if a film of solids, rust, or product coats the demister pad. The effect of the additional pressure drop on the thermal efficiency is similar to that resulting from an increase in the overall pressure through air leakage or through other problems, such as poor vacuum system performance.

Entrainment losses can be attributed to the following factors: (1) damaged inlet guides for the cyclone which permit the liquor to flow away from the body of the separator into the exit vapor stream, (2) higher vapor velocities than originally designed, (3) holes in the demister pads, (4) high liquid levels, (5) air leakage in drop legs, (6) splashing and vacuum fluctuations that cause surges in the vapor velocity, and (7) boiling and foaming of the liquor in the body of the separator.

Troubleshooting of the entrainment separators is normally made by inspection after measurements of liquor temperatures and pressure drops are taken. The troubleshooting procedure begins by comparing these measurements with the theoretical balance. If the liquor temperature exiting the body in a coalescing-type separator is higher than originally calculated, this will indicate that the pad is partially plugged. (The pressure drop over the pad can best be measured with a manometer.) Also, a temperature differential between the temperature of the liquor entering the cyclone and that at the bottom would denote a high pressure drop in a cyclone separator.

In the absence of a high vapor pressure liquor, entrainment losses should not be high enough to cause the solids concentration in the condensate to exceed approximately 100 ppm. However, in some applications 10 ppm may be too much entrainment to accept.

The benefits of improving existing separator performance are, then, enhanced product yields and improved thermal efficiency. The thermal efficiency improvement is a combination of reduction in fouling and pressure drop. Normally, the separator performance can be greatly improved by eliminating air leaks, maintaining proper vacuum and level control, replacing old demister pads or baffles, and repairing existing separator internals. The present vapor velocities should also be compared with the vendor design criteria. To predict theoretically the separator performance is almost impossible, and most separator designs have been made on the basis of existing commercial experience or testing. Since in most cases the separators cannot be modified to increase separation efficiency without extensive technical evaluation and retrofit costs, replacing the

existing separators is justified only when the entrainment is greatly affecting on-stream times or thermal efficiency. The best approach is to maintain the performance of the existing units by keeping the separators clean and making necessary maintenance repairs and to prevent operational problems, such as sudden changes in pressures which affect separation of the liquor and the vapor.

Optimum Pressure Profile

The major consequences of each problem area were changes in the pressure profile across individual effects. The objective for correction of these problems was to prevent an increase in the pressure within each effect. The higher pressure, in turn, reduces the available ΔT for heat transfer by raising the boiling point of the liquor and increases the amount of thermal energy required to maintain capacity, unless extensive heat-recovery systems (condensate flash tanks, feed preheating, etc.) are already installed. The factors affecting pressure drops, which have been previously discussed, include poor vacuum system performance, fouling, air leakage, poor venting rates, high water leakage, and poor vapor/liquor separation. The remaining problem areas that can affect this profile are restrictions in vapor piping, inadequate vapor/liquid separation, and improper maintenance of liquor levels.

Still another phenomenon that reduces the Net ΔT available in the evaporator is the hydrostatic elevation of the boiling point. Since heating surfaces must be submerged under the boiling liquid, it is necessary that certain depths of liquor be maintained. The resulting hydrostatic pressure gradient will cause a temperature gradient from the top to the bottom of the heating surface. The effective temperature for heat transfer will lie somewhere within the gradient. This effective temperature must be higher than that at the surface of the boiling liquid and therefore represents another potential loss to the Net ΔT available. Maintaining unnecessarily high liquid levels above the heating surface will shift the temperature gradient to higher values and result in further losses. Higher pressure steam must then be used to maintain capacity, and lower energy utilization occurs.

If the optimum pressure profile is being maintained, the pressure within the vapor space of each effect and the condensing pressure in the steam chest of the next effect are identical except for the vapor-line pressure drop. Normally this pressure drop is low to maintain maximum ΔT for heat transfer. If the line becomes plugged by entrained solids or corroded surfaces that reduce the cross-sectional area, the pressure drop increases. This requires an increase in the steam rate and pressure to maintain capacity. From the initial calculations made on the system and the measured body pressure, the condensing temperature in the steam chest can be estimated. From

Figure 32. Vapor pressure drop per hundred feet of pipe in the evaporator vapor lines. Curves are based on 12-in.-diameter pipe. Note: To find the ΔP for pipes with diameters other than 12 in., multiply the ΔP from Fig. 32 by $12^5/d^5$, where d is the new diameter.

Example: If the body pressure is 10 psia and the vapor rate is 13,000 lb/hr, the ΔP should be 0.25 psia, and the steam-chest pressure in the next effect will be 10 minus 0.25, or 9.75 psia. A lower pressure in the next effect would indicate a restriction in the vapor lines.

Fig. 32 the pressure drop as a function of vapor rate and the body pressure can be obtained. The condensate temperature in the steam chest should be within a few degrees of the saturated temperature at the calculated pressure for the steam chest ($P_{\text{body}} - \text{vapor line } \Delta P = P_{\text{steam chest}}$). For example, a first-effect body pressure of 10 psia at a vapor rate of 13,000 lb/hr should have a pressure drop of 0.25 psia (Fig. 32). This would give a steam-chest pressure of 9.75 psia and a saturated steam temperature of 192°F. A lower condensing temperature would indicate a restriction in the vapor line or high radiation losses.

Figure 33 shows the increase in steam rate required to overcome increased pressure drop through the vapor piping of a single- and a triple-effect evaporator. The effects on capacity and thermal efficiency also will become particularly acute for evaporators with low design temperature differences for heat transfer.

In every area that was evaluated for correcting the operation of the existing unit, the importance of maintaining the optimum pressure distribution was discussed. Whatever the reason, an increase in the operating pressure reduces the ΔT for heat transfer, and this requires that the

steam pressure (temperature) must be raised. At higher steam pressures without adequate condensate heat recovery, increased energy is required for the same rate of evaporation.

Figure 33. Effect of higher vapor-line pressure drop on steam rate.

Maintenance of the Optimum Operation

Once the existing evaporator has been evaluated and the corrections made to the existing unit, it is important to maintain those conditions. Without appropriate routine evaluations any operating cost reductions achieved by reaching the best thermal efficiency can be lost immediately.

The easiest way to maintain the newly derived optimum performance level is to instrument the unit, maintain an operating log, and routinely check the performance.

Figure 34 shows the minimum and maximum amount of instrumentation that would be required. The minimum level of instrumentation would generate data from which routine calculation analysis steps identical to those previously described can be made. This would require allocation of engineering time for completing the heat and material balances since the performance is not directly measured.

If the maximum amount of instrumentation is installed, the measurements would provide the basis for completing a

Figure 34. Required maximum and minimum instrumentation to check the performance of a multi-effect evaporator.

number of simple calculations to check performance. In this case the steam rate is known and can be easily compared to the net evaporation rate to monitor energy efficiency continually.

The use of computer programs greatly reduces the technical cost for continued surveillance. These programs would be valuable for any plant with multieffect systems or numerous evaporator installations.

The economic benefits of good instrumentation are difficult to predict. Generally the costs for installing pressure and temperature sensors and for running periodic heat and material balances become insignificant in comparison to reducing operational problems that significantly

increase the operating costs. However, it has been shown that the economics of no instrumentation or evaluations can be extremely poor.

The analysis of the existing plant now forms the basis for further study of the upgrading methods that usually have a higher energy-saving return through the addition of hardware. The heat and material balance that is derived by eliminating the effects of the present problems in the existing unit becomes the basis for these studies. The following two chapters describe the economics, calculation methods, and types of hardware which will further reduce the operating costs and steam usage for evaporators.

Chapter 3

IMPROVING ENERGY USE THROUGH MINOR CAPITAL INVESTMENTS: HEAT RECOVERY

INTRODUCTION

Chapter 2 dealt with the evaluation of existing plants and the identification of the major causes of reduced energy efficiency. Use of the plant evaluation as a basis for further upgrading analysis and the economics of optimizing the existing plant were also presented. This chapter discusses the technology and the economics of installing heat-recovery equipment to recover further the energy from such sources as the steam condensate, product, and feed. In addition to improving energy consumption, heat-recovery systems will also greatly decrease the sensitivity of evaporators to changes in operating conditions, which were shown to affect the energy usage of most existing evaporators.

For single-effect evaporators the selection of a heat-recovery scheme is a straightforward design procedure. The savings in operating costs for the few available options easily can be evaluated and weighed against the required investment. For the more complex multieffect systems, the heat-recovery options increase dramatically, and, although the basic engineering remains straightforward, the complexity of analyzing the actual savings in operating costs increases.

The sources of heat loss and thus the major points of potential heat recovery for both single-effect and multi-effect evaporators were shown in Figs. 2 and 3 of Chapter 1. The potential for and magnitude of the recovery of energy in these streams are functions of their temperatures and flow rates. For heat to be recovered, there must be a user at a lower temperature. The greater the temperature difference between the receiving stream and the heat source, the more favorable are the economics, since the equipment cost is less.

Generally, heat-recovery methods will not affect the operating conditions, although the steam rate at the same pressure and temperature will be reduced. Heat-recovery options obviously depend on the existing conditions; therefore it is critically important that the existing unit has been optimized and any problems affecting performance

have been corrected. In most cases the major heat-recovery options will include preheating the feed with heat recovered from the condensate and condensate-product flashing.

The usual heat sources available for preheating the feed are condensate and product streams. The recipients of the vapors from condensate and product flashing are the effects that are operating at lower temperatures and pressures.

FEED PREHEATING

A cold feed stream is one of the most attractive recipients for low-level heat recovered from the condensate and product. The energy required to bring the feed to its boiling point in the first effect is supplied directly by steam. If the temperature of the feed entering the evaporator is increased, the steam rate decreases. The steam saved by preheating a feed by recovered heat is plotted as a function of the increase in feed temperature over the original temperature in Fig. 35. The energy savings can be significant if the feed temperature is very low. In a single-effect unit using 30,000 lb/hr steam, increasing the feed temperature from 100 to 180°F, a temperature increase of 80°F, would achieve a steam savings of 30%, or 9000 lb/hr. Since the steam necessary to bring the liquor to its boiling point does not generate vapor, this steam adds directly to the total energy requirement. Cold feed to a multieffect evaporator would consume the same additional energy as that to a single-effect evaporator, provided both have the same final product concentration and pressure. The first step in any heat-recovery system should be an examination of the feed conditions and the available heat sources.

Figure 36 shows some possible ways to preheat the feed in forward-feed evaporators, where the liquor flow and vapor flow are in the same direction. As the feed enters the first effect, the temperature of liquor and pressure are higher than in any subsequent effect. Steam must be added to bring the feed to this boiling point, and any increase in the feed temperature reduces the steam rate. Since the

Figure 35. Steam savings as a result of increasing feed temperatures. Basis: Feed rate, 100,000 lb/hr; evaporation rate, 27,000 lb/hr; maximum feed temperature, 180°F; no boiling-point rise.

temperature of the steam in the first effect must be above the liquor boiling point, it is common practice to use this stream to preheat the feed. In addition, the condensates from the other effects offer possibilities although the temperature differentials for heat transfer will be lower. The heat exchange between feed and product is also an option although the application is somewhat more complex because of physical properties of the product.

In backward-feed evaporators, the feed enters the last effect first. Since this effect operates at the lowest pressure, the feed entering it does not require as great a temperature rise to reach the boiling point. The product usually exits the first effect at a higher temperature, and therefore the basic problem is to find streams that can use heat from the hot product. Although it could be used to preheat the feed entering the last effect, there would be greater economy in preheating the liquor between the first and last effect, where additional flashing of the liquor can be put to use.

There is a practical limit to the amount of preheating that can be economically accepted by the feed. This limit is based on the economics of purchasing additional heat-exchange hardware, and the economics for a specific evaporator is affected by the physical properties and temperatures of the heat-exchange process streams. As the temperature of the product decreases and further levels of heat recovery are achieved, both the heat-transfer rates and ΔT for heat transfer are reduced, and the heat-transfer area

and investment escalate for only marginal increases in heat recovery. Figure 37 plots this relationship and shows that the best return on investment is at a ΔT of 20°F. After this point, the investment begins to escalate for marginal heat-recovery increases. In an analysis of the economics for liquid/liquid heat exchange, this pattern of diminishing returns will always occur.

Viscosity and/or the crystallization point of the product stream is also important since the pressure drops through the hardware can be restrictive. If the product stream becomes viscous at lower temperatures, the heat-transfer rates decrease, the required heat-transfer area begins to rise exponentially, and the pumping and handling costs also increase rapidly. If the product begins to crystallize, then additional handling and fouling problems

Figure 36. Various schemes for preheating feed in forward-feed evaporators.

Figure 37. Steam savings over a range of ΔT 's between the feed and product for various installed heat exchanger costs. Basis: Triple-effect evaporator; feed rate, 100,000 lb/hr at 100°F to heat exchanger; steam consumption, 16,000 lb/hr (unheated feed); steam at 27 psia; product flow, 73,600 lb/hr at 145°F to heat exchanger. Curves valid for specific example only.

Time required for steam savings to equal investment, years		
ΔT , °F	With steam at \$3/1000 lb	With steam at 75 cents/1000 lb
40	3.2	8
30	1.3	5.1
20	1.1	4.4
10	1.2	5.1

occur. An estimate of the product film temperature under the most unfavorable conditions must be made and checked against the crystallization temperature.

In the design for a feed/product heat-exchange system, the consequences of a temporary interruption must be evaluated. If, in the event of a product-flow stoppage, the cold feed, which is still being pumped, will further cool the product to the solidification or crystallization temperature, consideration should be given to using another stream (such as a condensate) or an auxiliary steam line to preheat the feed to above the product freezing point before the feed enters the heat exchanger. If such preheating is not possible, some arrangements must be made for sensing a product-flow stoppage and bypassing the feed in such an event.

The feasibility of preheating the feed with condensates from each effect depends on the current use of the condensate. Since the first-effect condensate has not contacted the process stream, it can be returned to the boiler for reuse, and its heat is recovered there. If the

evaporator system is so remote from the boiler that the bulk of the heat would be lost, then it may be economical to exchange even the first-stage condensate with the feed and to return cold condensate to the boiler.

Since the condensates from other effects are slightly contaminated from entrainment or vaporization of the product, they may not be suitable for return to the boiler. In most cases these condensates are discharged into the plant waste-treatment system or used as makeup water elsewhere in the plant. The temperature of these condensates is usually much lower than that of the first effect. If the feed temperature is lower than that of the condensate, a series of heat-recovery exchangers can use all the condensates to preheat the feed.

The use of steam condensates and the hot product for a liquid/liquid exchange must be balanced against their possible use in flash tanks. Although flash tanks are usually less expensive than heat-recovery exchangers, the total amount of recoverable heat is lower. Basically, if the feed stream can use the sensible heat of the condensates and the condensates are not returned for boiler makeup, condensate heat recovery would be the best option for reducing steam consumption.

Using the product to preheat the feed is similar to using the condensates except that in some cases the condensate will be pumped from a vacuum. Also, the condensates usually are nonfouling and are easily handled, which makes the heat exchange with the feed and the calculation of heat-transfer rates much less critical than in the case of some products.

LIQUID/LIQUID HEAT-EXCHANGE HARDWARE

Most liquid/liquid heat exchange can be made with conventional shell-and-tube heat exchangers. This type of exchanger usually is the most economical to install per unit of heat-transfer area provided only mild steel is required. However, as is discussed later, the plate type of exchanger has several features that make it attractive for heat recovery in many evaporation systems.

Shell-and-Tube Exchangers

Since there are many excellent publications related to the design and selection of shell-and-tube exchangers (see the Bibliography), this section will address only the features of special interest to an engineer who is upgrading an evaporation system.

Figure 38 shows the approximate costs for various shell-and-tube exchangers. Many vendors now are using computerized design techniques that easily determine the

Figure 38. Approximate purchase cost of various shell-and-tube heat exchangers. Basis: Tubes, $\frac{3}{4}$ -in. O.D.; carbon steel shells; design pressure, 300 psig. [From J. P. Fanaritis and J. W. Bevevino, *Chemical Engineering*, 83(14): 70 (1976).]

size of the exchanger. The shell-and-tube exchanger is the workhorse of heat exchange. It is especially applicable to nonviscous fluids, with which standard off-the-shelf carbon steel equipment can be used. As a special precaution in the exchange of product heat with feed, the fouling fluids should be kept on the tube side, where cleaning is easier. The feed, or condensate in the case of condensate heat exchange, should be kept on the shell side. In retrofitting heat-recovery systems, be sure to allow ample room to pull the tubes and remove the nozzles for cleaning.

Plate Heat Exchangers

There are many instances in which the compactness, short contact time, and versatility of possible flow arrangements make the plate exchanger the best choice for liquid/liquid heat exchange. If alloy construction is required, the plate heat exchanger should receive special attention. In addition, higher heat-transfer rates can be

achieved (sometimes at the expense of higher pressure drops) to increase the percentage of heat recovered.

A plate exchanger can squeeze a large amount of heat-transfer area into a small space. This can minimize piping requirements and thus lower the retrofitting cost. Of greater importance is the versatility of the flow interchange patterns possible with plate exchangers. By the use of appropriate plates, two or more heated streams (such as condensates from each effect) can be exchanged with the same feed stream.

Figure 39 shows a configuration in which first a condensate stream and then a product stream are interchanged with the feed. In this case the condensate could be cooled to a low temperature while the product stream is protected against excessive cooling.

The plate exchanger is easy to clean. The plate package can be disassembled so that all heat-transfer surfaces can be reached. Although the costs per square foot for plate exchangers are usually higher than those for shell-and-tube exchangers, the total installed costs can be lower. Plate exchangers generally achieve much higher heat-transfer coefficients for the same function, and less heat-transfer area is required. The narrow spacing between the heat-exchange plates makes these exchangers more suitable for heating or cooling heat-sensitive or viscous materials. However, the narrow spacing and the support clips present a possible plugging problem if the feed or product contains suspended solids. Also, the compatibility of the gaskets and the fluid that is being processed must be checked.

Since most suppliers of heat exchangers use computerized design of their equipment, it is relatively simple to investigate a wide range of cases. Therefore, in requesting cost and design, the process engineer should feel free to have the total expected range of conditions investigated. Changes in viscosity will be especially important to the design.

Figure 39. Plate heat exchanger configuration to exchange both condensate and product streams with the feed.

Condensate Flash Tanks

As discussed earlier in this chapter, the condensate from each of the evaporator effects usually has available energy. One method to recover this energy is to exchange hot condensate under pressure with the cold feed or with another cold stream. If there is no cold stream, the use of condensate flash tanks can be economical. For backward-feed evaporators, product flash tanks can be used in the same manner.

Flash systems for condensate and product are shown in Fig. 40. For these flash tanks, the condensate from each effect or the hot product is allowed to flash into the vapor line of the following effect. The steam generated by this flashing is then used for further evaporation. The amount of flash vapor created is a function of the pressure differential (condensate pressure and the flash-tank pressure). Figure 41 can be used to estimate the amount of additional vapor that would be generated by flashing condensates between the indicated pressures.

The flashing of additional vapor from the flash tanks into the following effect will cause some changes in the operating pressures of the evaporation system. This pressure variation occurs because the temperature difference between the steam chest and the vaporizing liquid will automatically be redistributed so that the heat contained in the additional vapor can be transferred. Thus the total vapor generated by the flash tank itself does not necessarily mean the same reduction in steam usage. For example, in Fig. 42, which indicates conditions for a system using two flash tanks, the total flash rate of 840 lb/hr from the condensate achieved only a 410 lb/hr reduction in the steam rate. However, this steam reduction decreases the thermal energy cost by \$9890/year at \$3/1000 lb of steam.

For the first-effect condensate being returned to the boiler under its own pressure, a condensate pump may be

needed to deliver the condensate to the condensate receiver at the steam boiler. Thus the installation of flash tanks does require some specific analysis of each plant.

In addition to decreasing steam consumption, flash tanks will reduce the sensitivity of the evaporator to the loss of thermal efficiency when higher pressure steam is needed in the first effect. Thus, for the situations discussed in Chapter 2 in which, to maintain capacity, the operator introduced new energy inefficiencies into the system by raising steam pressure as the heat-transfer coefficient decreased, a much smaller effect would be seen if flash tanks were used. This is shown in Fig. 43 in which the energy consumption of a double-effect evaporator is analyzed with and without the use of flash tanks. The installation of flash tanks reduces the steam consumption by 395 lb/hr under normal operation with design heat-transfer coefficients. When fouling progresses to the stage where the coefficients are reduced by 20%, the increase in steam rate to maintain capacity is 690 lb/hr without flash tanks. With flash tanks the increase in steam rate is 620 lb/hr for the same decrease in heat transfer. The same decrease in sensitivity can be obtained by any scheme in which the sensible heat in the condensates and product is returned to the evaporator system.

The incorporation of heat-recovery units does not vary the conditions of the evaporator to any significant degree. The investments are moderate. The risks associated with the installations are low since the operational procedures remain unchanged. The next chapter describes the technologies that require higher investment and will change the operating conditions of the unit. Although those technologies are more complicated to evaluate, the energy reductions are usually much higher than those with heat-recovery methods or optimization and maintenance of the optimum conditions that the existing evaporator can reach.

Figure 40. (a) Condensate flash system for forward-feed evaporators. (b) Multistage product flash for backward-feed multieffect evaporators.

Figure 41. Percent of condensate flash in condensate flash tanks. Approximately 10 lb of vapor is generated for every 10°F drop per 100 lb of condensate.

Figure 42. Energy savings with condensate flash tanks.

Steam consumption without condensate flash = 15,500 lb/hr
 Steam savings with condensate flash = 410 lb/hr
 Total steam savings at \$3/1000 lb/hr = \$9890
 Total steam savings at 75 cents/1000 lb/hr = \$2470

Figure 43. Material balance for a double-effect evaporator with and without flash tank over a range of heat-transfer efficiencies.

MATERIAL BALANCE

Without Flash Tank at Design Conditions

Steam

Flow rate = 16,375 lb/hr

Pressure = 26.5 psia at 243°F

Vapor

$V_1 = 11,920$ lb/hr at 192°F

$V_2 = 15,180$ lb/hr at 134°F

Flash = 0

Condensate

$C_1 = 16,375$ lb/hr at 243°F

$C_2 = 11,920$ lb/hr at 192°F

At 80% of Design Heat-Transfer Coefficient

Steam

Flow rate = 17,065 lb/hr

Pressure = 38.5 psia at 265°F

Vapor

$V_1 = 11,595$ lb/hr at 202°F

$V_2 = 15,505$ lb/hr at 134°F

Flash = 0

Condensate

$C_1 = 17,065$ lb/hr at 265°F

$C_2 = 11,595$ lb/hr at 202°F

With Flash Tank at Design Conditions

Steam

Flow rate = 15,980 lb/hr

Pressure = 26.5 psia at 243°F

Vapor

$V_1 = 11,435$ lb/hr at 193°F

$V_2 = 15,660$ lb/hr at 134°F

Flash = 860 lb/hr

Condensate

$C_1 = 15,120$ lb/hr at 193°F

$C_2 = 12,295$ lb/hr at 193°F

At 80% of Design Heat-Transfer Coefficient

Steam

Flow rate = 16,600 lb/hr

Pressure = 38.5 psia at 265°F

Vapor

$V_1 = 10,970$ lb/hr at 204°F

$V_2 = 16,130$ lb/hr at 134°F

Flash = 1100 lb/hr

Condensate

$C_1 = 15,500$ lb/hr at 204°F

$C_2 = 12,070$ lb/hr at 204°F

Chapter 4

UPGRADING EXISTING EVAPORATORS THROUGH MAJOR CAPITAL INVESTMENTS

INTRODUCTION

The preceding two chapters discussed how to reduce operating costs in an existing evaporator by correcting any problems and by adding auxiliary hardware to recover waste heat. These methods do not change the initial process conditions of the unit or the basic flow within the boundary of the evaporator sections. Also, the changes in the heat and material balances are small, and the capital investment necessary to reduce energy consumption is minimal. Significantly higher reductions in steam consumption can be achieved with techniques that do alter the existing evaporator layout and change the process conditions. However, these techniques will require higher capital investments and installation costs.

Since the basic heat and material balances will be changed, the study of these options must be made in conjunction with a study of the heat-recovery methods since they will directly affect the economics and application of the options. In fact, application of the major options can change the conditions to the point where the lesser heat-recovery methods no longer apply. However, in each situation the data developed in the initial study, as described in Chapter 2, always will be the basis for the evaluation.

The techniques that can significantly reduce evaporator heat load and, if desired, simultaneously increase capacity include

- Thermal recompression.
- Mechanical recompression.
- Addition of effects.

The selection of one of these major modifications is complicated by the fact that the process conditions and the heat-transfer rates (coefficients) vary according to the existing process conditions and the desired degree of recovery. Therefore the best option will depend on the economics of energy savings and the investment and on whether the objective for upgrading is solely to improve thermal efficiency at the present capacity or to increase the

capacity as well. Often a combination of major modifications can minimize capital costs and still obtain the objective.

The procedure for determining the applicability of recompression techniques is started by identifying the performance characteristics and limitations of the hardware and auxiliary equipment and matching them to the process conditions of the evaporator. The limitations include the compression ratios,* liquor boiling-point rise (BPR), and available steam pressure. The addition of effects becomes a question of the economic size for the evaporator, the physical-property restrictions of the product, and the resulting changes in the evaporator process conditions.

Figure 44 shows how each technique or combination of techniques generally is applied in evaporation. Applying these techniques to existing systems is more difficult than applying them to new systems since some limitations already have been set by the previous design criteria, for example, heat-transfer areas and design steam pressure.

The evaporator process situations for which the major modifications normally will be most applicable and which will result in significant reductions in evaporator heat loads also are summarized in Fig. 44.

All recompression methods use the vapors from the existing plant and recycle them to the heating side of the evaporator. Thermal recompression uses a steam jet to compress a fraction of the overhead vapors with high-pressure steam. Mechanical recompression uses a compressor driven by a mechanical drive (electric motor or steam turbine) to compress all the vapors. Both vapor-recompression applications are limited both by the compression ratio and by the existing heat-transfer area and resulting heat-transfer rates.

For the preceding reasons, thermal recompression generally will be applicable when the existing first-effect heat-transfer ΔT is below 30°F and high-pressure steam is

*Ratio of absolute pressures of the vapor in the steam chest to that within the evaporator body.

ADDING THERMOCOMPRESSOR TO SINGLE-EFFECT EVAPORATOR

Best existing conditions

- Compression ratio of $P_2/P_1 \leq 1.8$
- Near atmospheric pressure at P_1
- Low liquor BPR
- No condensate heat recovery
- Low-pressure steam for evaporation (P_2)
- Clean vapor without corrosive or fouling products
- Usually feasible when adding effects

ADDING MECHANICAL COMPRESSOR TO SINGLE-EFFECT EVAPORATOR

Best existing conditions

- Low compression ratios with maximum of $2/1$ (P_2/P_1)
- Low liquor BPR
- Near atmospheric pressure at P_1
- Cheap electricity for drives
- High-pressure steam available for turbine drive
- High feed temperature
- Condensate heat recovery possible
- Low corrosive vapor carry-over
- Single-effect evaporator
- Also feasible as preconcentrator for multieffect evaporators

Figure 44. Major-modification schemes for existing evaporators.

(Figure continued next page)

ADDING NEW EFFECTS

Best existing conditions

Low BPR

Low number of existing effects

Existing evaporator pressure is vacuum

High Net ΔT available for heat transfer

Figure 44 (Continued).

used to supply the thermal energy. Thermal recompression is also used in conjunction with adding effects when the loss in heat-transfer rates can be overcome by adding more area to the new effect.

Mechanical recompression normally will be applicable for evaporators with a low ΔT for heat transfer, for single-effect evaporators, or as a preconcentrator along with another evaporator section if the BPR is low ($\sim$ less than 3°F). Although a liquor may have a much higher BPR, a preconcentrator using mechanical recompression can be designed to concentrate the feed to the point where BPR becomes prohibitive.

Since mechanical recompression recycles essentially all the vapor back to the steam chest, its application for existing multieffect evaporators normally will be as a preconcentrator. The combination of preconcentrator and existing multieffect evaporator usually is made to expand the capacity of the existing plant and to simultaneously achieve energy savings.

Adding an effect to an existing evaporator offers more flexibility for upgrading and generally can be applied to any existing evaporator. Although some restrictions still apply, the limitation is one of economics rather than process unless the BPR is very high. The reduction in heat-transfer

efficiency (ΔT and the heat-transfer rates or coefficient) of each existing effect can be compensated for by adding more area to the new effect. The new process conditions also may play a major part in determining the feasibility of adding effects. For example, the new process conditions might increase scale formation (fouling) or heat degradation of the product, both of which reduce the on-stream times and the ΔT for heat transfer. In most applications the optimum combination of thermal recompression and adding an effect will result in the most favorable reduction of heat load with minimum capital investment.

The difficulty in applying these techniques to existing multieffect evaporators is the problem of accurately predicting the change in heat-transfer coefficients since the modification will alter the process conditions. Nothing remains as it once was. Therefore the engineer is faced with a trial-and-error procedure to develop heat and material balances and to determine the resulting heat-transfer rates to see if the capacity can be maintained. In some applications, particularly in single-effect units, the existing conditions may match the optimum recompression requirements. In some cases the conditions of the existing evaporator can be adjusted by increasing the available ΔT in the evaporator with larger noncondensable vacuum systems. The total analysis becomes one of reducing each effect's heat load (sensible heat) faster than the loss in heat-transfer rates or the rise in the boiling point. Therefore the application of each method depends on the type of evaporator system and the physical properties of the solutions.

The following sections will deal with the limitations, calculation methods, and economics for applying the major modifications. Again, it should be remembered that each evaporator must be evaluated independently and whatever modification is selected must be based on accurate data from the existing unit, its operational trends, and confirmation of the initial results by consulting hardware suppliers. The first step is to calculate the various heat and material balances within the evaporator and then to determine whether the heat-transfer rates will accommodate the change and still maintain capacity. In any case the potential for significantly reducing operating costs justifies the expense for conducting initial studies.

THERMAL RECOMPRESSION

Thermal recompression is the term used to describe the application of a steam ejector or thermocompressor to the evaporator for the purpose of reducing the heat load. A steam jet achieves this by taking a fraction of the overhead vapors to the suction of the steam jet and compressing those vapors to a higher pressure and temperature with a high-pressure source of steam. The steam entering the thermocompressor is termed the motive steam. The applica-

tion of thermal recompression depends primarily on the process and operating conditions of the existing evaporator, the operational limitations of the ejectors, and the availability and pressure of the motive steam. The steam requirements of the evaporator are therefore met by mixing the low-pressure vapor off the evaporator with the high-pressure motive steam. If the first-effect vapors contain noncondensibles, recycling this stream to the steam chest may reduce heat-transfer rates, as previously described in Chapter 2.

Thermal recompression is especially applicable to single-effect evaporators and to the first effect of a multieffect system. If the present steam supply to the evaporator is reduced through a pressure regulator from pressures exceeding 100 psig to pressures approaching 10 psig in the first-effect steam chest, thermal recompression should be studied. Basically thermal recompression is the replacement of the pressure regulator with a steam ejector to permit heat recovery. The new steam consumption becomes the amount of motive steam required.

Hardware Description

The thermocompressor design is very similar to that of steam ejectors commonly used for vacuum generation. The basic ejector design or control loops are modified to compensate for the dynamics of the evaporator system. The ever-changing nature of most evaporators, e.g., different flow rates or variable pressure drops due to fouling, makes a study of this application difficult.

The standard types of ejectors used for thermal recompression are shown in Fig. 45. Each ejector consists of three basic parts: nozzle, mixing chamber, and diffuser (venturi throat). The higher pressure motive steam enters the nozzle section and expands through the nozzle into the mixing chamber. As the vapor expands, it produces a suction that entrains the vapors from the suction side. The suction and motive steam are recompressed through the diffuser and are discharged against the back pressure, which in this case is the operating pressure in the steam chest for the evaporator. If the motive steam is dry and saturated, the vapors will be slightly superheated but less so than the steam that is presently being used since lower temperature vapors are being mixed.

The two types of ejectors are of standard steam-ejector design, with fixed multiple nozzles that have no control over the motive-steam flow and with a regulating spindle that has some control over the volume of flow through the nozzle by varying the size of the nozzle orifice with a valve stem. Fixed-nozzle thermocompressors are more efficient and are used when the thermocompressor can be operated under steady-state conditions. Thermocompressors with manual or automatic operated spindles are designed for use when performance approaches critical conditions and

Figure 45. Standard types of ejectors used for thermal recompression.

changes in suction load, suction, and discharge pressures are expected.

Thermocompressors are basically single point designed, i.e., the design and operation are best at the one specified design condition of suction pressure and load, motive-steam pressure, and vapor rates. The ejector design may be either critical or noncritical. Critical design means that the vapor velocity in the diffuser is sonic and occurs at compression ratios above 1.8 (ratio of absolute pressures of the discharge and the suction). Since ejectors designed for higher compression ratios in which the performance is critical cannot

be controlled by varying the motive-steam flow, any change in suction pressure or discharge pressure reaching the critical compression ratio will cause the thermocompressor to "break"; that is, the variation in the suction and discharge pressures will cause loss of capacity, pump cavitation, and liquor flashing in the body from unstable pressures within the evaporator. For these reasons thermocompressors are designed for noncritical operation.

In steam ejectors of noncritical design, the changes in suction pressure and vapor rates from the evaporator can be adjusted by varying the motive-steam pressure or flow. The design of the thermocompressor for the evaporator is therefore based on the performance curve, such as the general one shown in Fig. 46. The design point is chosen on the basis of the most typical evaporator operating conditions. If a constant suction load is to be maintained and the discharge pressure increases, the suction pressure (pressure in the evaporator body) will increase to maintain the design compression ratio. A reduction in the suction pressure (when the vapor rate decreases) will reduce the discharge pressure. An increase in the discharge pressure decreases entrainment in the thermocompressor and thus reduces the suction load and heat recovery. If the discharge pressure is further increased, unstable operation will occur, and the unit breaks. If the operating pressure and therefore motive-steam flow are increased, the higher discharge pressure can be maintained at the same suction load. However, the amount of steam flow entering the steam chest is increased, and the evaporator must be balanced to keep capacity constant. Control by increasing the motive-steam pressure

Figure 46. Typical thermocompressor performance curve.

or inlet pressure has a limited range of application, depending on the hardware design. In addition, under a critical flow any reduction in motive-steam pressure below the design pressure causes the system to break.

Thermocompressors, if applied correctly, offer, for a much lower installation cost, energy savings equivalent to the addition of another effect. Other advantages of thermocompressors are their simplicity of construction, reliability of mechanical performance, and low maintenance costs. Therefore under the right circumstances thermocompressors should be considered.

Applying Thermocompressors

The application of thermocompressors to existing evaporators is based on the existing operating conditions for the present system, design motive-steam pressure, and the effects of thermocompression on the operating conditions and heat-transfer rates. The capacity ratio* and sizing chart for the hardware are used as guides to predict the heat recovery and economics. The basis of the calculation begins with the detailed heat and material balance previously done for fine tuning the existing plant. The maximum compression ratio to evaluate is set at 1.8, and the first effect will normally be chosen as the location for the thermocompressor. If this compression ratio must be exceeded or if the operating conditions cannot be adjusted to maintain capacity in the evaporator, the thermocompressor by itself is not applicable.

The design motive-steam pressure for the thermocompressor is normally taken to be 15 psig below the existing inlet pressure to the pressure regulator. For example, at a measured supply pressure of 140 psig to the pressure regulator, the thermocompressor motive-steam pressure for sizing the unit and selecting the capacity ratio would be 125 psig. This pressure drop and expansion help dry out the steam to the thermocompressor.

Figures 47 and 48 show typical capacity ratio and sizing curves for fixed multiple nozzle and regulating spindle thermocompressors. These curves should be used to approximate thermocompressor size and performance. The actual efficiency can vary with the size of the thermocompressor, the control variability, the system, and the supplier. It is not uncommon to obtain a variance in design efficiency exceeding 10% from different suppliers.

For a suction pressure at atmospheric pressure and a discharge pressure of 25 psia (compression ratio of 1.7) with 100-psig steam, Fig. 47 gives a capacity ratio of 1.1 for multiple nozzle units and 0.6 for regulating spindle units.

This capacity ratio for an evaporator means that, for the multiple nozzle thermocompressor, 1.1 lb of vapor from the evaporator would be boosted from atmospheric pressure in the evaporator body by using 1 lb of motive steam, for a total flow to the steam chest of 2.1 lb at a pressure of 25 psia. An increase in motive-steam pressure to 200 psig for the multiple nozzle thermocompressor would raise the capacity ratio from 1.1 to 1.3, which uses less motive steam to meet the total steam flow into the evaporator steam chest. Once the total steam rate is calculated for the existing evaporator with a thermocompressor, the unit can be selected from the sizing chart on the basis of the discharge pressure and total steam flow. It is interesting to note that although thermocompressor size is consistent with loading capacity, boosting capability can vary considerably.

Figure 49 shows operating conditions for a single-effect evaporator both with and without a thermocompressor. Generally, sizing and predicting the performance of thermal recompression for the single-effect evaporator is a straightforward calculation, provided the existing unit has a compression ratio below 1.8. The only change in the evaporator itself is in the vapor rate to the condenser. The heat-transfer coefficient, total evaporation rate, and total steam rate to the steam chest remain constant. The amount of steam savings is obtained from the capacity ratio curves as a function of the compression ratio and design motive-steam pressure. In the example in Fig. 49, the compression ratio is 1.7. The discharge from the compressor is 15,800 lb/hr, and steam savings become 8400 lb/hr. The new steam rate is 7400 lb/hr, which is the motive-steam flow to the compressor. As seen in Fig. 48, with this discharge flow an 8-in. thermocompressor would be required.

The calculation becomes more difficult for single-effect evaporators with existing compression ratios above critical design and for multieffect evaporators in which the recycle of vapor for the thermocompressor affects the heat balance of the next effects. Consider first a single-effect evaporator similar to the one illustrated in Fig. 49. If the steam pressure for the existing steam chest was 40 psig instead of 10 psig, the compression ratio would be 3.7 (54.7/14.7) and would exceed the critical design limit of 1.8. Direct application of thermocompression is not possible; the steam pressure must be reduced to 12 psig to meet the allowable compression ratio. This reduction in steam pressure lowers the steam temperature in the steam chest and thus reduces the temperature difference for heat transfer. Since the total amount of heat transfer is a function of $UA \Delta T$, the evaporative capacity would drop. This loss in heat-transfer rate would be the ratio of the $U \Delta T$'s since the area remains constant. Instead of reducing the steam pressure to 12 psig, another possibility would be to decrease the evaporator heat load. This would permit use of the lower pressure steam and reach the compression ratio for the thermocompressor. As discussed in Chapter 3, the heat load could

*Capacity ratio is the ratio of the suction-vapor flow rate to the motive-steam flow rate. The total steam required for the evaporator is the motive steam. The total steam rate entering the steam chest is the sum of the motive and suction flows.

Figure 47. Capacity ratio curves of steam ejectors for thermal recompression, — → —, see discussion in text.

be decreased by increasing the feed temperature through heat recovery or by reducing the operating pressure in the evaporator with a vacuum system. A third possibility would be to increase the heat-transfer area in the existing single-effect unit or to add another effect. The greatest steam savings would be reached by adding an effect with thermocompression. Alternate methods would require addi-

tional investment for heat-exchange surfaces, and the evaluation depends primarily on economics.

The calculation for determining the application of the thermocompressor for multieffect evaporators is even more difficult because the vapor from the first-effect evaporator is presently used to concentrate further the liquor in the next effect. In addition, any changes in the operating

Figure 48. Sizing chart for thermocompressors.
 — → —, see discussion in text.

conditions will affect all effects. However, multieffect units normally are designed at low ΔT 's in the first effect and offer good prospects for the application of thermocompressors. Figure 50 shows a triple-effect example. Application of the thermocompressor would reduce the vapor rate to the next effect. The first-effect evaporation rate must be increased to permit enough vapor to enter the next effect. As for the single-effect unit, the application of the thermocompressor depends on the existing evaporator process conditions and the resulting changes in heat-transfer rates and conditions after the thermocompressor is added. It becomes a question of balancing the decrease in the

pressure profile and boiling point of the liquor against the drop in the vapor rates to the next effects and changes in heat-transfer coefficients. As shown in Fig. 50, the application of the thermocompressor for multieffect units will change the operating conditions in these general ways:

- First-effect total evaporation rate must increase to accommodate recycle of vapors to the steam jet. For this to occur, the vapor pressure must decrease so that liquor boiling point is reduced to the point where the heat-transfer rate can be maintained at a higher ΔT .

- Pressure decreases in all effects except in the last effect, which is controlled by the vacuum system. This pressure reduces because of the decrease in heat-transfer rates.

- Except for the last effect, heat-transfer coefficients decrease per effect because of the lower liquor temperature and higher liquor concentrations and viscosities in each effect. The last effect remains the same since the liquor concentration, pressures, and temperatures remain constant.

- The ΔT will increase in the first effect at a constant steam-chest pressure as the body pressure reduces until BPR becomes significant.

- Evaporation rates reduce in each effect after the first one since more of the evaporation takes place in the first effect and the vapor temperatures in the next effects have been reduced.

- Condensate return to boiler from first effect may have to stop if entrainment occurs.

Therefore the application of thermocompressors is more promising when the existing evaporator conditions are such that the sensible heat load can be reduced at a rate equivalent to the drop in heat-transfer coefficients. This situation develops in cases with high feed temperatures, low BPR, low percentage of evaporated liquor, low concentration profiles, high first-effect pressures, and low-pressure steam used for evaporation.

The calculations for determining the application of a thermocompressor can take various forms. Each method requires a trial-and-error solution (see "Calculations," this chapter) starting with these basic assumptions:

- The new overall heat-transfer coefficient for each effect for the new operating conditions can be assumed to be equivalent to the original coefficients at the new concentrations. The heat-transfer coefficients from the initial evaluation, discussed in Chapter 2, are plotted as a function of concentration and are used to estimate new coefficients.

- The suction load to the thermocompressor is treated as a bleed stream or vapor drawoff from the effect selected for locating the thermocompressor.

- The initial bleed rate is estimated to be equivalent to the reduction in steam rate that can be achieved by adding another effect.

Figure 49. Single-effect unit with and without a thermocompressor.

SYSTEM WITHOUT THERMOCOMPRESSOR

Steam rate = 15,800 lb/hr
 Feed rate = 50,000 lb/hr
 Product rate = 40,000 lb/hr
 Cooling water = 1,950 gpm
 Vapor rate = 10,000 lb/hr

SYSTEM WITH THERMOCOMPRESSOR

Compression ratio = $24.7/14.7 = 1.7$
 Capacity ratio = 1.13 (see Fig. 47)
 Motive-steam rate = 7,400 lb/hr
 Feed rate = 50,000 lb/hr
 Product rate = 40,000 lb/hr
 Cooling water = 310 gpm
 Vapor rates
 To thermocompressor = 8,400 lb/hr
 To condenser = 1,600 lb/hr
 Total steam rate into evaporator steam chest = 15,800 lb/hr
 Steam savings = 8400 lb/hr

- The steam pressure and temperature are assumed to be those of the existing pressure if the ratio of the existing pressure to the first-effect vapor pressure is less than 1.8. Otherwise, the steam pressure is assumed to be 1.8 times the present evaporator body pressure for the initial calculation.

- The first estimate of the steam flow to the first-effect steam chest is assumed to be the bleed rate plus the existing first-effect vapor rate. The calculation procedure based on the preceding assumptions is basically identical to the trial-and-error procedure of calculating the evaporator steam rate until the product composition was matched (described in Chapter 2 and further illustrated by the calculations at the end of this chapter). The pressure profile will change, and the calculation procedure (excluding the

last effect) must be modified since the new vapor pressures are unknown.

The calculation for an evaporator similar to that in Fig. 50 will start with the following general steps on the basis of the assumptions previously described.

1. The amount of heat transferred in the first effect is equal to the assumed steam rate multiplied by the enthalpy differences between the vapor and condensate. This steam-flow rate is assumed to be the total amount of vapor that is generated in the first effect or the sum of the vapor bleed to the thermocompressor (B) plus the original vapor rate (V).

2. Then the product weight fraction of solids of the liquor from the first effect is estimated according to the formula $X_{Pr_1} = X_F F / [F - (B + V)]$, where X_{Pr_1} is the

Figure 50. Applying a thermocompressor to a three-effect evaporator.

Conditions*	FIRST EFFECT		SECOND EFFECT		THIRD EFFECT	
	Without thermo-compressor	With thermo-compressor	Without thermo-compressor	With thermo-compressor	Without thermo-compressor	With thermo-compressor
Steam (S), lb/hr	4580	3270				
Vapor recycled (B), lb/hr	0	3700				
Total evaporation (B + V)	3020	5610	3410	2140	3570	2250
Vapor pressure, psia	18.5	14.7	11.0	10.0	7.8	8.0
Liquor temperature, °F	225	213	198	195	185	185
Concentration, %	18	21	22	25	30	30
U, Btu/(hr)(sq ft)(°F)	350	298	275	250	210	210
Steam savings = 1,310 lb/hr or \$31,000/year at \$3/1000 lb						
*S = steam rate; B = vapor rate to thermocompressor; V = vapor rate to next effect.						

product weight fraction of solids, X_F is the feed weight fraction of solids, and F is the feed rate (lb/hr).

3. The first-effect heat-transfer coefficient is estimated at the concentration calculated from step 2 and the plot of U and concentration obtained from the existing unit. The values that are calculated at this point are the heat transferred in the first effect (Q_1), the heat-transfer coefficient (U), the steam temperature, and the first-effect material balance.

4. The temperature of the liquor in the first effect is then estimated from $T_1 = T_S - (Q/UA)$, changed from basic heat-transfer formula $Q = UA \Delta T$.

5. The vapor pressure of the first effect is obtained from boiling-point curves at assumed concentrations and liquor temperatures.

6. The preceding steps are repeated for each of the remaining effects, and the evaporator steam rate is adjusted until the calculated last-effect vapor pressure matches the actual pressure. At this point the calculated concentration is checked against the desired one.

7. If the desired composition is not met, the assumed bleed rate is reduced, and the calculation is repeated until the last-effect pressure and concentration match the existing conditions. (Note: If the existing ratio of steam

pressure and vapor pressure in the first effect exceeds 1.8, it may be impossible to calculate a matching condition.)

8. Once the concentration and pressure in the last effect match, then the compression ratio is checked. If it is below 1.8, refer to the capacity ratio and sizing curves, and select the thermocompressor. This represents one solution to the application.

9. The first satisfactory solution should be reviewed, and solutions at higher and lower motive-steam pressures and steam-chest pressures to the first effect should be evaluated. In addition, other changes to the existing conditions, such as a decrease in the last-effect pressure to obtain a more favorable pressure profile, should be investigated as possible options.

Once again it should be said that each evaporator must be evaluated independently. Since the heat-transfer areas are fixed, the capacity of the system will depend on the heat-transfer coefficients and ΔT 's for the new process conditions with the thermocompressor. In most cases the thermocompressor cannot be retrofitted into an existing plant unless excess heat-transfer area exists or auxiliary hardware is added to reduce the total existing heat load and thus give the same effects as excess area. (Note that most evaporators with a thermocompressor would have higher total heat-transfer areas than systems without one.) Therefore the available heat-transfer area is normally the limiting factor for upgrading the present unit with a thermocompressor alone.

Economics

The economics for installing thermocompressors is based on the steam savings and the cost of the thermocompressors. The number of effects in the present evaporator, the available steam pressure, and the capacity ratios (ratio of suction steam to motive steam) determine the steam savings. The discharge pressure for the thermocompressor, which is the steam-chest pressure, and the total flow rate of steam (motive steam plus suction vapors) which is required to maintain evaporator capacity will determine the size of the thermocompressor and the investment. The capacity ratio curves previously discussed are used to calculate the steam savings. Figure 51 shows the capital costs for the fixed multiple nozzle and automatically controlled spindle thermocompressors.

Figure 52 shows the steam savings vs. the capacity for single-effect, double-effect, and triple-effect evaporators in which conditions are suitable for thermocompression. For each evaporator the liquor has a small BPR. The savings for each evaporator are shown for steam-chest pressures of 10 psig and no heat recovery. The examples in Table 7 show return on investment as discounted cash flow and payback for three different cases.

Figure 51. Purchase costs for thermocompressors. Basis: Material of construction, cast iron-carbon steel bodies; nozzles, 304 stainless steel. No auxiliaries.

Figure 52. Steam savings from installing a thermocompressor to single-effect, double-effect, and triple-effect evaporators. Basis: Steam costs at \$3/1000 lb; operating time, 7880 hr/year; motive steam at 100 psig; suction pressure, 14.7 psia; discharge pressure, 10 psig.

Summary for Thermal Recompression

Thermal recompression is an effective method for achieving significant reductions in the steam consumption for existing evaporators by recycling a portion of the overhead vapors to the steam chests. If the conditions in the evaporator can accommodate the thermocompressor, it is the least expensive of the techniques for improving energy consumption by making major modifications to the system. For existing evaporators that use the steam at high pressures, the heat-transfer area may be a limitation. Therefore the application will normally be feasible for single-effect units and multieffect evaporators for which the existing pressure ratios are below 1.8 or the ΔT for heat

Table 7. Return on Investment for Conversion of Single-, Double- and Triple-Effect Evaporators to Thermal Recompression

Basis					
Compression ratio: 1.7					
Feed rate: 2 lb/lb of evaporation at 150° F					
Installed cost: 2.5 x purchased cost					
Steam: \$3/1000 lb for first year					
Cost escalation: 10% for every year after first year					
Return on investment: discounted cash flow (DCF), straight-line depreciation at 10-year life					
Fixed costs: maintenance cost is 6% of investment; maintenance labor, 37% of maintenance cost; taxes at 48%					
Compressor size, in.	Evaporator	Evaporation rate, lb/hr	Percent interest return on investment (DCF)	Payback, year	Steam savings (Fig. 52)
6	Single effect	5,000	680	0.2	\$71,500
8	Double effect	10,000	350	0.3	\$56,600
10	Triple effect	25,000	380	0.3	\$77,000

transfer can be increased by reducing the vapor pressure within the evaporator body at a faster rate than the heat-transfer coefficients decrease. In all instances the effect where the thermocompressor is located will be vaporizing more water from the liquor. Thermocompressors also can be widely applied in conjunction with modifications to auxiliary hardware, such as feed preheaters (heat recovery) and additional effects.

The application of thermocompressors is limited when the BPR is high, the heat-transfer coefficients decrease rapidly with liquor concentration, the existing evaporator presently uses high-pressure steam, a low-pressure source of steam supplies the heating load, or the steam condensate is being returned to the boiler for makeup.

MECHANICAL RECOMPRESSION

Mechanical recompression is the second type of vapor recompression scheme used to reduce evaporator energy consumption. Mechanical recompressors have heat-transfer and compression-ratio limitations similar to those of thermocompressors, and the calculational approach also is basically the same. However, fundamental differences between the two require individual analysis for each evaporator.

Mechanical recompression achieves the highest possible energy savings since the vapors are completely recycled, compressed, and discharged into the evaporator steam chest. These energy savings are reached at the expense of much higher investments and increased electrical power consumption. Since there is usually total vapor recycle, the mechanical recompressor must be installed on existing single-effect evaporators or included in a new preconcentrator for existing units, as shown in Fig. 53.

The major factors to consider in selecting a compressor are the flow rate, compression ratios, variability of evaporator flow rates and operating conditions, vapor temperature, expected maintenance, and installed costs. Generally, the costs for all mechanical recompressors limit their application to situations in which the existing compression ratio is below 2 : 1, since multistage compressors or a series of single-stage compressors would be required at the higher compression ratios.

Compressors applied to evaporation systems include positive-displacement models and radial-flow and axial-flow centrifugal types.

The positive-displacement compressors, such as rotary-screw and sliding-vane types, have one or more rotating elements in a casing. The positive-displacement machines displace a constant volume of vapor, and therefore the mass flow rate depends on the inlet gas density. The total efficiency of the positive-displacement compressor is approximately 70 to 74%. The maximum capacity for a single-stage rotary is approximately 10,000 inlet cubic feet per minute (inlet cfm) of water vapor, which is equivalent to 20,000 lb/hr of water vapor at atmospheric conditions. These compressors can be used at compression ratios below 2.0. Therefore positive-displacement units are used for low-flow applications and slightly higher compression ratios than are centrifugal types.

Axial-flow centrifugal compressors direct the flow of gas parallel to the compressor shaft. Their capacity range is much higher than that of radial-flow compressors and approaches a maximum at 400,000 inlet cfm, or approximately 800,000 lb/hr of water vapor at atmospheric conditions. The total efficiency is 85%, the highest for any compressor.

Figure 53. Mechanical recompression for a single-effect evaporator or as a preconcentrator for existing double-effect evaporator.

Above compression ratios of approximately 1.5, a multi-stage unit must be used. Axial-flow units are more expensive than radial-flow compressors, and, except for situations in which the capacity is needed or the efficiency justifies the added expenditure, a radial-flow centrifugal compressor normally is used.

Radial-flow centrifugal compressors are the most common compressors for large evaporation applications where the rate of evaporation exceeds approximately 40,000 lb/hr. The gas changes direction radially at the inlet in the compressor. For most duties their design simplicity and commercial flow ranges bracket those of the axial-flow and positive-displacement types. Their capacity ranges are 3000 inlet cfm (6000 lb/hr of water vapor) to 150,000 inlet cfm (300,000 lb/hr of water vapor) for atmospheric suction conditions. The normal total mechanical efficiency is 75%. The single-stage compression ratio is limited to approximately 1.5. The basic advantage for radial-flow units as compared to axial-flow units is a greater range of performance to match standard methods of pressure and temperature control, as well as their capability for turndowns of 50% and a lower sensitivity to changes in suction and discharge pressures.

When the proper application for it exists, the thermal economy for mechanical recompression is 10 to 15 lb of water evaporated per pound of steam (or equivalent) energy. This exceptionally high energy utilization is equiva-

lent to a fifteen-effect evaporator. With the compressor's mechanical and frictional inefficiencies included, 93 to 95% heat recovery usually is obtained.

The selection of the type and size of mechanical compressor is best made by the vendors who are specialists in the application of the hardware. However, the project engineer can conduct a preliminary evaluation of the size, design conditions, and economics. Since most compressor suppliers offer each type of compressor, the best one for the job can be supplied if the design conditions are properly developed. The objectives for this section therefore are to identify the situations in which mechanical compressors are applicable for evaporation and to determine the economics involved so that a comparison can be made with the other upgrading techniques. Detailed design methods and equipment descriptions can be found in the publications in the Bibliography and the vendor literature.

Compressor Performance and Auxiliary Equipment

Compressors are sized to handle the required volume of vapor under design conditions of compression ratios and flow rates. However, the evaporator is never a stable operation. Discharge pressures will vary as the surfaces foul and evaporation rates change. The compressor selected should be the one that best accommodates any of the

known variations in the operation without reaching an inoperable condition. The characteristic curves* of the compressor and the addition of the recommended instrumentation and auxiliary equipment for the installation will help in obtaining satisfactory long-term operation.

The relationship between the inlet volume and the discharge pressure of a radial-flow centrifugal compressor is shown in Fig. 54. This is termed the characteristic curve.

Figure 54. General characteristic curve for radial-flow centrifugal compressor.

Axial-flow compressors have similar characteristic curves, although the change in pressure rise is much greater with equivalent changes in compressor rotational speed. These curves show the change in pressure head developed (head is the difference in absolute discharge pressure to suction pressure) and the percent of volume being compressed as a function of compressor speed. The application of a compressor to evaporators is made by selecting the design point that allows changes in the compressor operation to match the operation of the evaporator. The compressor is used to adjust the evaporator body pressure and the steam pressure flow into the steam chest. The designer will have four basic choices for control. Given in the order of power efficiency, they are

- Control of compressor speed by using a variable-speed drive. (The most energy-efficient choice.)
- Control of suction pressure by throttling inlet vanes.

- Control of suction pressure by throttling an inlet valve.

- Control of discharge pressure by throttling a valve. (No savings.)

These methods of control must be studied in relation to the evaporator operating conditions and the compressor performance shown in the characteristic curve. The compressor performance must be changed to accommodate any changes in the flow rates and heat-transfer conditions (ΔT over the evaporator heat surface). For example, if the feed rate is reduced, lower pressure steam is required to maintain the product concentration. From the characteristic curve (Fig. 54), it can be seen that reducing the compressor speed from point A to point B would reduce the steam pressure to the steam chest while adjusting the heat-transfer ΔT required for the reduced flow.

The change in compressor speed to achieve equilibrium depends on the evaporator design and the liquid being concentrated. The equilibrium point could be points A to C, which would maintain a constant vapor volume by further reducing the pressure increase between suction and discharge pressure. If the design point on Fig. 54 was point D and a higher ΔT was required to maintain capacity, an increase in compressor speed to point E would allow the operational change.

As just discussed, the volume of vapor that can be compressed can be held constant by varying the speed of the compressor drive. If the discharge pressure must change, the capacity (volume of gas compressed) automatically adjusts to meet the change in pressure. At a constant speed the flow will decrease as the pressure increases. If the inlet pressure varies, the discharge pressure also changes. In other words, the compressor maintains a total head in relation to the difference between the discharge and the suction pressures. In addition, the power consumption is almost directly proportional to discharge pressures and the volume compressed, which in turn is subject to changes in the inlet pressure and temperature flow rates.

This automatic balancing process has limitations. There is a maximum vapor capacity that can be compressed with a given compressor despite a significant drop in the discharge pressure. Perhaps of more significance for an evaporator, there is a maximum discharge pressure and minimum vapor rate at which the operation becomes unstable. This point of instability is called "surge."

At surge conditions the inlet and discharge pressures and the flow rates rapidly oscillate as the compressor tries to find a condition to satisfy its stable-performance characteristics. The surge occurs because the process requires a higher compression ratio than the compressor can achieve. The instability caused by surge will cause flashing in the evaporator body, loss of capacity, and variable product compositions. If surging continues for some time, the hardware will be damaged.

*The characteristic curves show the performance of the compressor as a function of head developed, drive speed, horsepower requirements, and volume of gas being compressed.

For evaporator applications in which conditions will vary, provisions can be designed into the installation to prevent damage and to ensure a trouble-free operation. Figure 55 shows such an installation and methods of control. A blowoff valve will permit venting of excess vapors and will allow the evaporator to be adjusted for reduced feed rate. A variable-speed drive, such as a steam turbine with a governor to control steam flow, is the most efficient method to control flow rate and adjust discharge rates. As indicated on the characteristic curve, changing the speed permits a wide range of control. Alarms for high pressure or low flow should also be considered.

thereby reduces the volume of steam entering the compressor and affects the total head (pressure) that can be developed. The butterfly-type valve, in contrast, directly imposes an artificial pressure drop in the suction line, which lowers the inlet pressure. The valve or vanes control the discharge pressure and/or inlet volume. The new discharge pressure is equal to the new inlet pressure times the fixed pressure ratio if the butterfly valve is used and the pressure ratio remains fixed. In addition, the horsepower requirement decreases in direct proportion to the inlet pressure when a constant volume is being compressed. The mass flow is varied by changing the gas density. The inlet vanes

Figure 55. Various control loops for mechanical re-compressor for single-effect evaporator.

When constant-speed drives are used, which is usually the situation when steam turbine drives are not employed, the suction pressure to the compressor can be throttled (automatically or manually) by using inlet-guide vanes or butterfly-type valves. This control adjusts performance by varying the volume. The inlet pressure to the compressor controls the discharge pressure and volume. As the suction pressure is reduced by increasing the line pressure drop between the evaporator and the compressor inlet, the compressor operates farther down on its characteristic curve (points D to F, Fig. 54) and thus the discharge pressure is reduced. If the pressure drop is decreased, the discharge pressure increases.

The inlet-guide vanes achieve these results by changing the angle of incoming vapor in relation to the rotating impeller. This shifts the best operating point and

are more power efficient, although they are more expensive and complex than is valve control.

Another method to avoid surge is to control the discharge pressure by throttling the discharge flow. The suction pressure (evaporator pressure) and compressor loads vary directly with any change in the discharge pressure. This means of control fixes the power consumption and should be used for very small installations whose power efficiency can be neglected. In addition, any throttling of the valve that increases the discharge pressure above the compression capability would cause surge. Throttling at discharge of the compressor is therefore avoided owing to the flat pressure-volume performance characteristics of most compressors.

If the mechanical compressor is to be installed on an existing unit or as a preconcentrator for a new installation,

the compressor is selected on the basis of the most reliable performance over the expected range of evaporator operating conditions. With proper installation and startup of the mechanical compressor, maintenance frequency can be extended by 1 to 2 years. Naturally the proper material of construction should be chosen, and entrainment of water or corrosive vapor into compressors should be avoided to maintain optimum performance and minimum maintenance costs.

Applying Mechanical Recompression

Mechanical recompressors take all the vapor from the existing evaporator and compress it to a higher pressure and temperature. The total amount of recoverable energy depends on the following factors.

- Compression ratio.
- Discharge pressure required for the evaporator.
- Liquor boiling-point rise.
- Selection of compressor drives.
- Condensate heat recovery.

Calculating Procedure

Steam compressed to a higher pressure becomes superheated. Theoretically the amount of energy required to compress the vapor is equal to the difference between the enthalpies at the suction and discharge points, at a constant vapor entropy. The compressor drive and internal efficiencies must be taken into account in predicting the actual operating conditions. These data can be obtained from vendor specifications. Heat input due to friction will further increase the vapor temperature beyond that calculated on the basis of constant entropy. Figure 56 shows a single-stage evaporator and gives the procedure to size the compressor and to estimate power requirements.

The amount of superheat is the sum of that predicted from isentropic compression and that generated by the heat of compression of the compressor. This superheat is recovered as the steam condenses in the steam chest, since the net heat input to the steam chest is the difference between the enthalpies of the superheated steam and the condensate.

Vapor desuperheating to maintain the existing heat transfer must be considered. If superheated vapors enter the steam chest, the average heat-transfer coefficient is lowered because a portion of the heat-transfer surface is not operating with a condensing coefficient. The liquor side of the tubes is essentially acting as a desuperheater, and this area is not being used effectively for evaporation. Therefore, if this heat-transfer area is required for evaporation, a desuperheater will be necessary. The desuperheater uses a water spray to control the temperature in the steam chest.

Since the water in the spray is vaporized to additional steam, the net heat input is still equal to the difference between the enthalpies of the superheated steam at the compressor outlet and the condensate. The maximum condensing coefficient is maintained in the evaporator since the steam entering the steam chest is saturated.

The preceding examples were based on the condition that the total heat recovered by the compressor equaled the evaporator heat load; makeup steam was not required. Unfortunately this is not always the case. There are several variables that will affect the energy balance for existing single-effect evaporators, e.g., radiation losses, feed temperature, boiling-point rise, and compression ratio (ratio of absolute pressures of the vapor in the steam chest to that in the vapor space). The situation can arise in which the total heat input from the compressor exceeds the evaporator heat load and vapor must be bled off the body of the evaporator. If the net heat input is not sufficient, live steam must be fed into the steam chest.

Boiling-point rise affects the potential applications of recompression in evaporators since the pressure in the vapor (space) at a given temperature is less than that for the saturated case. More heat-transfer area may be required to provide for the lower ΔT 's which favor vapor recompression. In existing evaporators BPR has been accounted for in the design; the only question is whether the design engineer has used enough area to overcome BPR at the compression ratio if the unit is operating at a ratio above the capability of the compressor.

If the existing evaporator has a ratio of pressure in the steam chest to pressure in the vapor space which is above 1.5, the chances are that the existing heat-transfer area will not be sufficient to maintain capacity with mechanical recompression where single-stage compressors can be applied. However, there may be methods, such as preheating feed, which will allow this capacity to be regained. In almost all cases compressor costs limit the application of mechanical recompression to existing single-effect evaporators with a pressure ratio of less than 1.5. Therefore this discussion will continue on that basis.

The effect on net energy savings over a range of compression ratios from 1.1 to 2.0 for evaporation systems with different liquor BPR's is shown in Fig. 57. It is assumed that the total input by the compressor equals the heat required to preheat the feed and compensate for radiation and other losses. The effect of BPR of 0 to 10°F above the atmospheric boiling point of water also is shown. The increase in the energy required to compress the water vapor associated at different BPR's is directly proportional to the amount of sensible heat required to reach the boiling point above that of pure water at the same pressure. This heat leaves with the liquor and is not recovered in the vapors. Since the drive energy to account for the internal

Text continues on page 60

CONDITIONS

V_1 = vapor flow of 25,000 lb/hr of steam

$T_L = 213^\circ\text{F}$

$P_1 = 15.0$ psia

$P_2 = 19.5$ psia

$T_1 = 213^\circ\text{F}$

T_2 (saturated) = 226°F

Temperature of the discharge vapors (T'_2) = 266°F

Steam (S) = Makeup steam, 230°F

Feed rate (F) = 75,000 lb/hr at 150°F

Product rate (Pr) = 50,000 lb/hr at 213°F

Figure 56. The procedure to size a mechanical recompressor and to estimate power requirements for a single-effect evaporator.

PROCEDURE

Change in enthalpy for isentropic compression can be found from the Mollier diagram (see Fig. 14) or from steam tables.

$$H_2 - H_1 = 1171 - 1151 = 20 \text{ Btu/lb}$$

Assume compression efficiency = 78%

Actual energy required = $20/0.78 = 26$ Btu/lb

(For the superheated temperature of the compressed vapor, add this quantity to the enthalpy of the vapors before compression. The temperature can be obtained from the Mollier diagram or steam tables. In this case total enthalpy of compressed vapor = $1151 + 26 = 1177$ Btu/lb. At this enthalpy and 19.5 psia, the superheated temperature of the discharge vapors is 266°F .)

Compression work = $26 \text{ Btu/lb} \times 25,000 \text{ lb/hr} = 650,000$ Btu/hr

An alternate way of computing the compression work is as follows: the volumetric inlet flow (from Fig. 61) for 25,000 lb/hr evaporation is 11,000 actual cubic feet per minute (actual cfm). The horsepower required for this flow (from Fig. 60) at 19.5 psia is approximately 225 hp. Compression work = $255 \text{ hp} \times 2546 \text{ Btu/hp-hr} = 650,000$ Btu/hr.

Overall heat balance:

$$h_F F + H_S S + wV = h_{Pr} Pr + h_C C$$

(but $h_C C = h_C V + h_C S$). Therefore

$$(H_S - h_C)S = h_{Pr} Pr + h_C V - h_F F - wV$$

where h_F = enthalpy of the feed = 118 Btu/lb ($cp = 1.0$)

h_{Pr} = enthalpy of the product = 181 Btu/lb ($cp = 1.0$)

H_S = enthalpy of the makeup steam = 1157 Btu/lb

h_C = enthalpy of the condensate = 194 Btu/lb

wV = compression work = 650,000 Btu/hr

Substituting and solving the equation above:

$$\begin{aligned} (1157 - 194)S &= 181 \times 50,000 + 194 \times 25,000 \\ &\quad - 118 \times 75,000 - 650,000 \\ S &= 4570 \text{ lb/hr} \end{aligned}$$

This indicates that makeup steam is required.

If S is negative, there is excess steam that must be bled out of the evaporator. Usually this steam is not bled from the compressor, since this means additional compression work. Unless the excess steam can be used elsewhere, it is best to bleed it before the compression. Under these circumstances, the preceding formula is not applicable. The following formulas are to be used when steam is bled out before recompression:

$$(H_V - h_C + w)B = (h_F - h_C + w)F - (h_{Pr} - h_C + w)Pr$$

where H_V = enthalpy of the vapor before compression

w = compression work per pound of steam

B = vapors bled from the evaporator before compression

The quantity of vapors recompressed can be calculated from

$$V = F - Pr - B \quad (B \text{ in diagram is } 0)$$

Figure 57. Net energy savings and brake horsepower for mechanical recompression over a range of compression ratios. Basis: 50% evaporation ratio (vapor/feed); suction pressure, 14.7 psia.

efficiency of the compressor is transferred to the vapor as superheat, the energy loss is associated with the motor drive itself. This power requirement is dependent on the electrical characteristics and the motor and compressor efficiencies. Therefore the net energy savings* are slightly decreased at the higher compression ratios if the mechanical and internal efficiencies are considered to be equivalent. The cost to drive the motor and the compressor investment also become considerably higher at the higher compression ratios, and, once the compression ratio reaches 1.5, multistage compressors must be used.

When the heat required for feed preheating and for radiation losses exceeds the net heat input from the compressor, then makeup steam is required. This amount of steam is shown in Fig. 58. At low feed temperatures the makeup steam can become significant, and the evaporator can be equipped with a heat exchanger to preheat the feed by using condensate from the steam chest. The heat exchanger can introduce an added pressure drop and can add to the existing steam-chest pressure. This may require a condensate pump or an increase in the compression ratio.

The situation can exist in which the present feed temperature is already above the boiling point of the liquor in the evaporator. Some vapors would then have to be bled off to control the vapor pressure and evaporation rate. Otherwise, the suction and discharge pressures would continue to rise and surging would occur, since the compressor is recycling larger mass flows of vapor while the actual volumetric flow rate is decreasing. These vapors could be discharged to the existing condenser or used elsewhere in the plant.

*Net energy savings is the total heat recovered in the evaporator minus the power requirement expressed as Btu/hr.

Figure 58. Makeup steam for mechanical recompression evaporators for various feed temperatures and compression ratios. Basis: Feed rate, 50,000 lb/hr; evaporation rate, 25,000 lb/hr; inlet pressure, 15 psia to steam chest; makeup steam at compressor discharge conditions; no BPR in liquor.

Mechanical Recompression for Preconcentration

Mechanical recompressors also can be combined with an evaporator section as an additional preconcentrating step for existing single-effect and multieffect systems to simultaneously increase capacity and/or improve energy consumption per ton of product, as in Fig. 53. The BPR plays a predominant role in sizing this preconcentrator, as for any new evaporator, since the heat-transfer area will be determined primarily by heat-transfer ΔT . If the BPR is significant, the size of the evaporator must be larger than otherwise would be necessary, and the extent to which this system can be used is limited. Use of the preconcentrator in the first stage is favored because of lower BPR. The effects of BPR on sizing evaporators are discussed in greater detail in the next section. A balance must be reached between the compressor size, investment, and operating costs and the evaporator investment and energy savings.

Since BPR is a function of concentration and the liquor concentration, the application of preconcentrators is particularly well suited to liquors with negligible BPR's or to dilute solutions in which the BPR can be neglected.

The calculation procedure is basically the same as the one previously discussed to determine the applicability of a thermocompressor. In this case, however, the resulting changes to the existing evaporator conditions, such as the heat-transfer coefficients and the heat and material balances, depend on the changes in feed concentration and temperature which will occur under the conditions selected for the new preconcentrator. A trial-and-error solution is

used to size the system to meet the desired capacity increase or to obtain the maximum steam savings at a minimum investment. The new steam consumption is the sum of the preconcentration rate and the new rate for the existing evaporator, which is now receiving a more concentrated feed.

Economics

Mechanical recompression can be applied to existing single-effect evaporators that meet the criteria of the compression ratio or as a new evaporator section that will preconcentrate the feed for an existing single-effect or multieffect unit. The economics primarily depends on the investment and maintenance costs since the energy savings (reduced steam consumption) are significant.

The costs for a mechanical recompressor are primarily based on the size of the drive and on the vapor rate and volume. The capital costs for a single-stage centrifugal compressor and single-stage rotary compressor as a function of vapor volume (dependent on evaporator body pressure and rate) are shown in Fig. 59. The brake horsepower requirements, as a function of discharge pressure and inlet volume, are indicated in Fig. 60 along with approximate drive costs. Multistage centrifugal investment costs are at least 200% greater than single-stage units.

The volume of steam in cubic feet per minute as a function of evaporation rate is shown in Fig. 61. The vapor volume at atmospheric pressure for 10,000 lb/hr of steam is 4900 actual cfm, as compared to 6500 actual cfm at 10 psia. Since the vapor volume is greater at the lower pressure, it is common practice to design new mechanical recompression evaporators at a minimum pressure of 14.7 psia, to reduce the size and cost of the compressor.

The total economic benefit for mechanical recompression is also decreased if the condensate in the existing unit is being recycled back to the boiler as makeup or the vapor purity off the evaporator does not allow its recycle. If the separator can achieve good separation, it may be practical to return condensate from mechanical recompressors to boilers. However, the level of impurities from vapor entrainment, due to separator carry-over caused by operating error, poor design, or the presence of distillable impurities, makes the practice unwise for most older evaporators and should be thoroughly investigated before recycling the condensate from the evaporator.

If high-pressure steam is available and a demand exists for low-pressure exhaust steam, a compressor driven by steam turbines reduces the energy usage of the compressor, compared to that of electrically driven units. If electrical costs are low in comparison to steam, however, electrically driven compressors are favored. With either drive the total energy consumption is nearly identical, and the investment for the turbine or electric motor is considered to be equal.

Figure 59. Estimated price of centrifugal and rotary compressors over a range of vapor volumes for a compression ratio of 1.3 to 1.5. Notes: (1) Compressor prepared for steam service. (2) Materials of construction: casing, cast iron; impeller, 410 stainless steel. (3) Price includes lube system, coupling, and baseplate ($Q \leq 85$) or soleplate ($Q \geq 85$). (4) Operating conditions: $P_1 = 15.0$ psia, $P_2 = 19.5$ to 22.5 psia, $T_1 = 218^\circ\text{F}$. (5) Price does not include the drive.

The use of vapor recompression becomes uneconomical as high compression ratios are required, because both the capital costs and the operating costs increase rapidly. A breakpoint occurs at a compression ratio of approximately 1.5, when a multistage centrifugal compressor must be used. Figure 62 shows the energy savings as a function of compression ratio for one evaporator, along with the return on investment.

Owing to the limits of the compression ratio, a new vapor-recompression evaporator is designed with a low temperature difference (ΔT) between the condensing compressed vapor and the boiling liquid in the vapor body. Therefore a larger heat-exchange surface is required than would be needed if live steam were used. The economic balance becomes one of the investment costs for the

Figure 60. Compressor horsepower requirements over a range of inlet volumes at various discharge pressures with suction at 14.7 psia. Compressor drive prices range from \$20 to \$35/hp, according to type of drive, insulation class, and electrical or steam characteristics.

Figure 61. Actual cubic feet per minute over a range of evaporation rates for water vapor at saturated conditions.

Figure 62. Energy costs savings and average return on investment of mechanical recompression as a function of compression ratio. Average return on investment is after taxes (48%) for a 10-year lifetime. Basis: Feed rate, 50,000 lb/hr at 150°F; evaporation rate, 25,000 lb/hr; inlet pressure to compressor, 15 psia; centrifugal compressors with drives.

compressor, the energy savings, and the heat-transfer area. In cases where high compression ratio would require multistage compressors, it may be beneficial to replace the existing heat exchanger with a larger one so that a single-stage compressor can be used.

Summary for Mechanical Recompression

Mechanical recompression offers the highest energy-savings potential of any upgrading technique. The energy recovery is above 90%. The limitations of compression ratios and investment costs now are being reduced by higher operating costs. With proper application of the mechanical recompressor and routine maintenance discipline, trouble-free operation can be achieved. Mechanical recompression preconcentrators are an excellent method for increasing the capacity of existing installations, and the application can be used for the majority of concentration processes. In addition, the preconcentrator alone can be economical in reducing the evaporator heating loads even without the added incentive of increasing the capacity.

ADDING EFFECTS

Figure 63 shows two typical multieffect evaporation schemes, forward feed and backward feed. Each effect in the evaporator consists of one or more bodies, with vapor lines connected in parallel. The feed can be introduced into the first effect to follow a path concurrent to the vapors. This configuration is known as forward-feed evaporation. The feed also can be introduced into the last effect to follow a path that is countercurrent to the vapors. This is called a backward-feed evaporator. In addition, feed can be

Figure 63. Typical evaporator process flows, with data for the heat and material balances. (a) Forward-feed evaporator. (b) Backward-feed evaporator.

Figure 64. Annual steam cost per 1000 lb of evaporation as related to ranges of steam costs per 1000 lb. Basis: 1 lb of steam evaporates 0.9N lb of water, where N = number of effects. — — —, see discussion in text.

introduced into an intermediate effect. It is possible to use any of these combinations in the same evaporator. The best configuration is a function of the steam and feed temperature, the properties of the process stream, and the product specifications. Normally, however, evaporators are either forward or backward feed.

When most existing evaporation systems were selected, one of the major design evaluations was a determination of the optimum number of effects. The initial choice probably was based on economics and the availability of capital. Since energy costs have escalated faster than equipment

costs, a new study may show excellent returns on investment for the capital costs required for additional effects. The addition of effects can be used to increase production capacity as well as to reduce energy consumption per ton of product. This will provide two incentives for upgrading—energy and capacity—and greatly improve economics.

Figure 64 shows the annual costs of steam per 1000 lb of evaporation at steam costs varying from \$1 to \$5/1000 lb and ranging from a single-effect to a ten-effect evaporator. This figure clearly illustrates the incentive to install additional effects because of new energy costs. The savings

by adding additional effects are strikingly changed by the new energy costs. The operator of a single-effect evaporator, who once had energy costs of \$1/1000 lb and presently has costs of \$3/1000 lb, can return to his earlier energy costs if he replaces his single-effect evaporator with a three-effect system.

However, adding effects is expensive, and this approach eventually runs into economic limitations even at high energy prices. If the initial feed and final product conditions are set by other processing considerations, the allowable temperature difference between condensing vapor and boiling liquor per effect decreases in proportion to the number of effects. As the average ΔT , and therefore the ΔT per effect, decreases, the capital cost increases rapidly. As more effects are considered, the steam reduction per effect added is decreased. As the solution is concentrated, BPR can further decrease the number of effects that otherwise would be economical. The economic balance is one that matches the maximum energy cost reduction with an acceptable return on the capital investment.

This section gives the process engineer guidance in evaluating the potential economics of installing additional effects to an existing system.

The Concept of Available ΔT

Since heat is supplied by the vapors of one effect to the boiling liquor in the steam chest of the following effect, a multieffect evaporator can be considered as a series of heat resistances. The driving force across each resistance is the difference in temperature between the steam chest and the boiling liquid, or the ΔT . The overall temperature difference between the steam to the first effect and the saturated temperature of the vapors at the condenser pressure (vacuum pressure) is defined as the Total ΔT available. For process fluids with a significant BPR, the Total ΔT is reduced by the sum of the BPR within each effect. This new ΔT is known as the Net ΔT available.*

To develop the concept and importance of ΔT , first consider a theoretical evaporator with constant (and equal) heat-transfer coefficients (U), no pressure drop in the vapor lines, and negligible feed heating and flash loads. In this idealized case the temperature of the vapor above the boiling liquid is the same as the condensing vapor in the steam chest of the next effect, and there are no pressure drop in vapor lines and no BPR. Therefore the individual ΔT 's in each effect add up to the Total ΔT available, and the Total ΔT is equal to the Net ΔT . Furthermore, because of equal heat-transfer coefficients (and equal areas), the resistances to heat transfer will be equally divided throughout the evaporator, and the Total ΔT available for heat

transfer will be shared equally among all effects. For N effects, the ΔT for each individual effect is then proportional to $1/N$. Since the rate of evaporation is proportional to ΔT (see Eq. 1),† the evaporation rate in each effect is also proportional to $1/N$ of the total evaporation. If one more effect is added to the evaporator, the heat-transfer area is increased from N to $N + 1$, while the ΔT and the evaporation rate in each effect decrease from $1/N$ to $1/(N + 1)$. Under these circumstances the total evaporation rate remains constant because $N + 1$ effects are evaporating $1/(N + 1)$ of the total load. Not only does the evaporation per effect decrease, but for the same reason the steam that is supplied to the first effect also is proportionately reduced. Therefore the steam required to evaporate a given quantity of water is approximately equal to $1/N$ of that required for single evaporation. This rule often is modified to 0.8 to 0.9 lb of evaporation per pound of steam to compensate for deviations from ideal conditions. Notice, however, that increased steam economy means additional capital investment. A twofold reduction in steam requires two effects of approximately the same area as a single effect (twice the capital cost) and so on for a higher number of effects. As described later, BPR and other factors set a practical limit to the number of effects that can be used in a given application.

There are many reasons for an evaporator to depart from the ideal behavior just described. One important reason is that because of the temperature differences between consecutive effects the liquid will either flash into vapor or require heating. For the evaporator to reach steady-state conditions, the individual ΔT 's must be rearranged to meet different heat loads in each effect. Total ΔT , however, is fixed by the steam temperature and the pressure at the condenser.‡

Figure 65 shows the temperature difference required in each effect for an evaporator with no BPR and an evaporation capacity of 19,400 lb/hr, starting with a single effect and considering up to five effects. Notice that as the number of effects increases the ΔT 's across each effect become smaller.

The temperature of the liquid in the first effect is easily calculated as the temperature of the steam minus the ΔT in the first effect. In the case of three effects, the ΔT in the first effect is 35°F, and, since the steam temperature is 225°F, the temperature of the liquor in the first effect is 190°F. Therefore, since ΔT decreases in the first effect, the pressure profile will be such that the temperature of the liquid in the first effect will increase as the number of

†In the section "Influence of Heat-Transfer Coefficients," this chapter.

‡Total ΔT = temperature of steam entering evaporator minus the temperature of saturated water vapor at the condenser at the pressure of the condenser.

*Net ΔT available = Total ΔT — sum of BPR's in each effect.

Figure 65a. Comparison of ΔT distribution for multi-effect evaporation with and without BPR (maximum BPR of 13°F occurs for single effect). Basis: Steam temperature, 225°F ; condenser pressure, 1.5 psia (114°F); feed rate, 50,000 lb/hr at 182°F ; cp, 1.0; area, approximately 500 sq ft/effect; heat-transfer coefficient, 300 Btu/(hr) (sq ft)($^{\circ}\text{F}$).

Figure 65b. Boiling-point rise for percent of solids. Applicable to Fig. 65a and the calculations at the end of the chapter.

effects is increased. For example, with a single effect the maximum liquid temperature is 114°F, but in a triple-effect evaporator this maximum temperature climbs to 190°F. Since the feed temperature is at 182°F, flashing will occur in the single-effect and in the first effect of the double-effect evaporator. However, in a triple-effect evaporator the feed is cold relative to the temperature in the first effect. Therefore some of the steam consumed by the triple-effect is used to heat the feed. As the number of effects is increased, an even higher portion of the steam consumption is needed to heat the feed. This heat will be stored as sensible heat to be released in latter stages, with lower than maximum economic gain. For this reason, a four-effect forward-feed evaporator will not have approximately four times the steam economy of one with a single effect. The delayed flashing, however, may have a detrimental effect on the efficiency of the evaporator, especially those with high feed rates, combined with low evaporation rates, and cold feeds ("Feed Preheating," Chapter 3).

A pattern is reflected in Fig. 65, starting with the triple-effect evaporator. The effect's ΔT is at a minimum in the second effect of the evaporator because of the increased demand for steam to heat the feed in the first effect. Because of the increased heat load, the first effect needs a larger temperature difference than does the second effect. On the other hand, the hot liquid flashes in the second effect, and this flash adds to the evaporation obtained. Therefore, since the third effect also must absorb the energy from a larger vapor load than the second effect, a larger ΔT is required there. The same trend also is observed in the four- and five-effect evaporators.

This illustrates one of the first departures from the ideal multieffect evaporation; the natural temperature gradients in multieffect evaporators cause heating and flashing vapor loads which result in unequal vapor splits (vapor flow distribution in each effect).

Because of increasing liquor temperatures, backward-feed evaporators usually experience flash when the feed enters the evaporator. This requires heating in the other effects and results in more equal vapor splits. Some energy in the product can be converted to additional evaporation by providing a flash tank, as previously described. Usually the required vacuum for the flash tank is obtained by introducing the flash vapors into one or more of the steam chests of the evaporator. Although the latter scheme is superior for steam economy, it must be investigated carefully since the heat loads and temperatures in each effect are altered.

Detrimental Effect of Boiling-Point Rise

As a liquid solution is concentrated in a multieffect evaporator, the content of dissolved solids increases. The

increase in the molar concentration of the solids causes the boiling point to rise above that of pure water. Boiling point rise (BPR), as this phenomenon is known, is defined as the difference between the boiling point of the specific solution and the boiling point of water at the same pressure. For example, a boiling solution of 14 wt.% CaCl_2 at 15.6 psia will show a boiling temperature of 219°F. The corresponding saturated temperature for water at this pressure is 215°F. Therefore the BPR for this solution is 4°F.

For a multieffect evaporator the vapors above this boiling solution would be at 219°F since they must be at thermal equilibrium with the boiling liquor. However, as the vapors are drawn to the next effect, they must condense to the saturation temperature corresponding to 15.6 psia, or 215°F. Therefore 4°F has been lost from the Total ΔT available to the BPR. This loss is detrimental to evaporative capacity and economy.

The loss in capacity from BPR can be seen in Fig. 65a. This figure graphically shows the loss in effect ΔT 's and the loss in evaporation rate brought about by BPR, as compared to the case with negligible BPR. Identical evaporators under equal conditions except for BPR were assumed in this case.

Furthermore, with BPR the rate of evaporation falls off progressively as the number of effects is increased; this was not the case for negligible BPR. Again, this represents a departure from the ideal model for multieffect evaporation. The capacity of the evaporator is not independent of the number of effects, even when all the heat-transfer coefficients are the same. The reason for this progressive loss in capacity can be easily explained by tabulating the BPR in each effect (Table 8).

Table 8 Tabulation of Boiling-Point Rise in Each Effect

Number of effects	Boiling-point rise, °F					Total
	First effect	Second effect	Third effect	Fourth effect	Fifth effect	
1	13					13
2	6	12				18
3	5	6.5	11.5			23
4	4.5	5	7	11		27.5
5	4	4.5	5.5	7	10	31.0

The Total ΔT available is the same for all cases, including the negligible BPR case, which is simply the difference between the steam temperature (225°F) and the saturated temperature of the vapor at the condenser (114°F). Subtracting the sum of the BPR from the Total ΔT available gives the Net ΔT , which is the net overall temperature difference available for heat transfer. In the

following table, the Net ΔT is shown as a percentage of the Total ΔT available:

Number of Effects	1	2	3	4	5
Net ΔT as percent of Total ΔT	88%	84%	79%	75%	72%

Figure 65a also shows that the drop in Net ΔT closely corresponds to the loss in evaporation capacity that can be recovered at the expense of energy usage. However, it should be noted that these curves apply only to the specific example in question. Indeed, the BPR exhibited by the liquid solution in this example is only a moderate one. Many substances, such as caustic soda, have a BPR several times higher than this example. For this reason caustic evaporators have been limited to double-effect or triple-effect units in the past. With the present price structures and energy costs, triple-effect evaporation is becoming the standard, with some quadruple effects also making their appearance in the caustic industry.

Detrimental Effects of Pressure Drop

The effects of loss of frictional pressure in the vapor piping are similar to those of BPR. The pressure loss from the vapor chamber in one effect to the steam chest in the next also represents a loss in ΔT . The loss in ΔT can be readily calculated and is equal to the difference between the saturated temperature of the vapor at the higher pressure and the saturated temperature at the lower pressure in the steam chest. For example, steam at 14.7 psia has a saturated temperature of 212°F. If the pressure loss in the vapor piping (and the entrainment separator) amounts to 1 psia, the saturated temperature of the steam in the steam chest of the following effect is 209°F (13.7 psia), a further loss of 3°F from the Net ΔT available for evaporation. For this example the temperature loss would represent a 2 to 4% loss in capacity, since the Net ΔT ranges from 80 to 111°F.

Influence of Heat-Transfer Coefficients

The heat transferred through each individual surface of the evaporator can be expressed as

$$Q = UA \Delta T \quad (1)$$

where Q = rate of heat transferred, Btu/hr

U = overall heat-transfer coefficient, Btu/(hr)(sq ft)(°F)

A = heat-transfer surface, sq ft

ΔT = temperature difference between the saturated vapor from the preceding effect and the liquid temperature in the i th effect, °F, or in some cases LMTD (see Fig. 12)

Occasionally ΔT 's are calculated by using the temperature difference between the vapor in the steam chest and the saturated temperature corresponding to the vapor pressure of the boiling liquid in that effect. The temperature difference obtained by this method is called the apparent temperature difference, and it is somewhat larger than the ΔT defined for Eq. 1 since it does not take BPR into consideration. Care must be taken that the U used in Eq. 1 is consistent with the ΔT . Heat-transfer coefficients obtained from an apparent temperature difference are called apparent heat-transfer coefficients and should be used with apparent temperature differences. Corrected heat-transfer coefficients, as defined in Eq. 1, are used throughout this chapter since they give better results under variable process conditions.

The heat-transfer coefficient is strongly affected by temperature, concentration, and viscosity. Other factors, such as liquor velocity, heat capacity, and thermal conductivity, also affect this variable. For this reason, it is important to know the conditions under which heat-transfer coefficients were derived. Experience and judgment are required for selecting these variables when the evaporator conditions are changed radically. A method for obtaining heat-transfer coefficients in existing evaporators has been previously discussed. If conditions affecting heat-transfer coefficients do not change radically, the same coefficients often are used to calculate heat balances under slightly different conditions.

In forward-feed evaporators heat-transfer coefficients and liquor temperatures usually fall very rapidly from the first to the last effect. This drop of the heat-transfer coefficient occurs because the viscosity increases as the concentration rises and the temperature drops. Backward-feed evaporators, on the other hand, do not show such a rapid change in viscosity since the higher temperatures at higher concentrations tend to compensate for the increasing solids. This is, in fact, an important factor in choosing the backward-feed configuration.

As was discussed previously, even with equal heat-transfer surfaces and equal heat-transfer coefficients in each effect, an evaporator cannot maintain equal ΔT 's. For similar reasons unequal heat-transfer coefficients will result in unequal ΔT 's unless the areas have been specifically adjusted for the changes in the original design.

In contrast to the case with BPR, where additional equal effects adversely affect capacity, multiple effects may have a beneficial effect on heat-transfer coefficients. With known values for heat-transfer coefficients given, it would be a simple matter to discuss the effect of this variable on capacity. For example, a simple relationship was assumed for heat transfer as a function of temperature and concentration, and the areas required to obtain equal evaporation were calculated for evaporators ranging from a single effect to five effects.

Figure 66. Distribution of heat-transfer coefficients in a multi-effect evaporator with from one to five effects. Basis: Heat-transfer coefficients are a function of temperature and concentration only; feed rate, 50,000 lb/hr at 150°F; evaporation rate, 36,000 lb/hr.

The results of this type of analysis are given in Fig. 66, showing the heat-transfer coefficients as a function of the number of effects. The total heat-transfer area required for each case also is indicated. The heat-transfer area required for a single effect is 3500 sq ft, and that for two effects is 5100 sq ft, or 2550 sq ft/effect. This appears to contradict the findings previously discussed concerning Fig. 65a, where, because of BPR, there was a loss in capacity even when the total area was doubled from one to two effects. The results in Fig. 65a were calculated by holding the heat-transfer coefficients constant. However, in the case shown in Fig. 66, allowances are made for the variations of the heat-transfer coefficients by using a correlation that accounted for changing temperature and concentration profiles within the evaporator. Under these conditions a different and real situation develops which shows the improved heat-transfer conditions for the first effect when the single-effect evaporator is converted to two effects. The heat-transfer coefficient for the single effect is 100 Btu/(hr)(sq ft)(°F), at a temperature of 142°F and a concentration of 65%. In the double-effect evaporator, the concentration and temperature in the last effect are the same as those for the single effect; therefore the heat-transfer coefficient usually is the same. However, in the first effect of this evaporator, the temperature climbs to 216°F and the concentration drops to 28%. Under these conditions (of lower viscosity), the heat-transfer coefficient is 250 Btu/(hr)(sq ft)(°F). Obviously the average heat-transfer coefficients have improved so that there is a smaller area than that predicted by the original idealized model. As the number of effects increases still further, the improvements in heat-transfer coefficients become less significant.

At the same time the detrimental effects of BPR and other losses become more pronounced. The extent to which these trends affect each other depends on such properties as the process fluid, the operating conditions, and the design of the evaporator, all of which play an important role in the economics of upgrading an existing evaporator.

These results will vary because of the many factors affecting heat-transfer coefficients, among the most important of which are the different types of evaporators available. Natural-circulation evaporators rely on temperature differences to promote circulation, and heat transfer is therefore very sensitive to reductions in ΔT . On the other hand, forced-circulation evaporators will be less sensitive to ΔT , and their performance is more closely related to boiling temperatures and viscosity, as was the assumption in the preceding example.

Temperature Sensitivity of the Product

Very often the temperature of the process stream is not allowed to exceed a certain maximum value. Usually this limit is imposed to avoid thermal decomposition of one or more of the constituents in the process stream. Substances exhibiting inverse solubility curves are also subject to temperature limitations since high temperatures can cause accelerated scaling of the heating surfaces.

It already has been shown that increasing the number of effects will result in higher temperatures in the first effect of a multi-effect evaporator. Increasing the vacuum at the condenser and reducing the ΔT of the evaporator is one way to overcome this problem. This adjustment is not easily achieved since the evaporator probably was originally designed for the lowest vacuum possible for the process. Some of the limitations to achieving higher vacuum are decreased solubility of the product at lower temperatures, insufficient flow rate, and temperature of the cooling water. Another disadvantage in increasing the number of effects for heat-sensitive substances is the increased residence time in the additional effects, which will result in increased losses from thermal decomposition. However, careful selection of the evaporator can minimize this residence time. Long-tube vertical evaporators, for example, have very short contact times.

Therefore, upgrading evaporators for heat-sensitive materials requires careful study and detailed calculations to predict the new operating conditions and to determine whether they are within acceptable limits.

Gaining Additional Economy under Conditions of High Boiling-Point Rise

For a given steam and condenser pressure, i.e., fixed Total ΔT , BPR limits the number of effects that can be

Figure 67. Energy conservation schemes for liquids with high BPR with comparison of energy consumption.

used in multieffect evaporators. Substances with very high BPR's at high concentrations may be limited to single-effect evaporation. For instance, in Fig. 67a a liquor is concentrated from 80% to 95% solids in a single-effect evaporator, and the Total ΔT available is 168°F. At 95% concentration the BPR is 119°F (Fig. 67e), and therefore the Net ΔT for heat transfer is 49°F. If a standard double-effect evaporator were to be used, the BPR in the second effect would remain at 119°F (Fig. 67b), since the same product concentration is required. For the first effect of the evaporator, the BPR would be at least 55°F, which corresponds to the concentration of 83% solids. Therefore the sum of the BPR's for

this evaporator would be at least 174°F, and, since the Total ΔT would still remain at 168°F, the Net ΔT would become negative. Obviously a standard double-effect evaporator would be incapable of concentrating the process stream to the required 95% concentration.

It might be possible to obtain the desired product concentration by increasing the Total ΔT available. This can be achieved by increasing the steam-chest temperature or the vacuum at the condenser.

If considerably higher steam pressure is available and product quality permits, one very effective way to attain high steam economy is to use the single-effect evaporator as

a steam-reducing station. The reducing-station evaporator replaces steam-reducing valves in plants where high-pressure steam is being reduced by control valves. For example, steam at 200 psig can be used to supply the evaporator, with the pressure in the vapor body kept at, say, 20 psig. The vapors then can be discharged into a low-pressure (20 psig) steam header line. Thus, instead of wastefully reducing the steam pressure with valves, a reducing-station evaporator can be used to achieve the same result and increase the thermal economy of the plant as well. Because of the high product temperatures and pressures resulting from this scheme, further evaporation can be attained by providing flash tanks. In this way the product can be cooled in the flash tanks to the desired temperature, and at the same time the sensible heat in the product is recovered.

There are limitations to the reducing-station approach for existing evaporators. The rated pressure of the evaporator body may be less than the required steam pressure; the product may be heat sensitive; the existing boiler may also be rated below this pressure. Because entrainment of liquid into the vapors will cause contamination of the steam from the evaporator, applications requiring clean steam are impractical. Plants equipped with turbogenerators and turbine drives would have no use for this type of evaporator since their high pressure steam is reduced by turbines.

If it is assumed that the single-effect evaporator is limited to the existing pressure (88 psig), there are other options to modifying the double-effect evaporator to operate within the limitations imposed by BPR.

Figure 67b shows a scheme in which the idea of a reducing-station evaporator has been retained with certain modifications. The existing single-effect evaporator is used as a reducing-station evaporator to reduce 88-psig steam to 20 psig. Because the vapor pressure has been increased from 10 in. Hg (vacuum pressure) to 20 psig, the Total ΔT available for heat transfer has been reduced from 168°F to 70°F. Under these conditions the evaporator is incapable of sustaining the same rate of evaporation, and the concentration drops to 83%. A new single-effect evaporator is required to bring the product concentration up to the required 95%. In order to overcome BPR, both evaporators are supplied with live steam. This results in higher steam consumption at the evaporator station, but, because live steam is replaced by vapors somewhere else in the plant, the equivalent of an 18% steam reduction is achieved.

In Fig. 67c, the double-effect evaporator is used to concentrate the process stream to a concentration of 90% solids. At this intermediate product concentration, the BPR is lowered to where it becomes possible to use a standard double-effect evaporator. The additional evaporation required is achieved with a single-effect evaporator as the third step. By achieving at least a portion of the total evaporation required in a double-effect evaporator, a 20% reduction in steam is obtained. However, a heavy price is

paid, since now, two additional evaporator sections are required instead of the single effect. A summary of the steam savings and additional heating surfaces required for each scheme is given in Fig. 67d; the single effect has been used as a base case.

These examples are meant to stimulate the interest of the reader and serve as guides. There are many possible variations, each having some merits and some limitations. Since no two plants are alike, the best approach will be that which is best suited to the particular plant conditions.

Determining the Economic Feasibility of Additional Effects

The first step in the investigation of adding additional effects to an evaporator is to conduct a heat and material balance on the existing evaporator. A heat and material balance is essential to understanding the capabilities and limitations of the evaporator, and it is a useful tool for checking the performance of the evaporator under normal and abnormal circumstances regardless of whether it will be upgraded.

The curves in Fig. 64 can be used on a preliminary basis to estimate return on investment and paybacks. For example, consider a two-effect evaporator with a 30,000-lb/hr evaporation rate and steam costs at \$3/1000 lb of steam. From that figure the annual steam costs are \$14,800/year for 1000 lb/hr of evaporation, or \$444,000 per year* at the rated capacity. However, if this two-effect evaporator is modified to four effects, the annual steam costs will be in the order of \$7200/year/1000 lb evaporated, or a total annual steam cost of \$216,000. Therefore, the potential steam savings are \$228,000 per year. Figure 68 shows the approximate purchase price of evaporators and the installation costs vs. the heat-transfer area. The additional area required can be estimated by calculating the average area per effect of the existing evaporator and using this as the required new area. In this example, the double-effect evaporator consists of a total area of 5000 sq ft, or 2500 sq ft/effect. Thus, an estimate of the additional area required is 2500 sq ft for each new effect under consideration. If a natural-circulation evaporator with stainless steel tubes and a stainless steel shell is assumed, the installed cost is as follows:

$$\text{Purchase cost} = (\text{base price}) \times \text{FM}$$

where FM is the material of construction multiplier.

$$\text{Base price} = \$100,000 \text{ (2500 sq ft)}$$

$$\text{FM} = 1.32$$

$$\text{Purchase cost} = \$132,000$$

$$* \left[\frac{\$14,800/\text{year}}{(1000 \text{ lb evaporation})/(\text{hr})} \right] \left(\frac{30,000 \text{ lb}}{\text{hr}} \right) = \$444,000/\text{year}.$$

Figure 68. Purchase cost of evaporators. For approximate installed cost, multiply by 1.8. Basis for approximate evaporator costs: 304 stainless steel for wetted parts (does not include installation, controls, or condensing equipment). For other materials of construction, multiply as follows: 1.0 for carbon steel—stainless steel; 0.32 for carbon steel—carbon steel; 1.32 for carbon steel—Inconel; and 1.47 for Monel—Inconel.

Installation cost (est. at 80%
of purchase price) = \$105,600 (2500 sq ft)
Total installed cost = \$237,600/effect
Total cost to convert
from 2 to 4 effects = \$475,200

The increased manufacturing costs must be deducted from the savings to obtain the payback. Operating labor and electricity may not change. Cooling water costs reduce slightly. For this example, increased maintenance is taken as 6% of the initial installed investment, and maintenance labor, as 37% of the maintenance costs. Depreciation is taken as 10% of the investment. The increased operating costs would be as follows:

Maintenance =	\$28,512
Maintenance labor =	10,549
Depreciation =	47,520
Total increase =	\$86,581
Steam savings =	\$228,000
Increased manufacturing costs =	86,581
Net savings =	\$141,419
Savings after tax =	\$ 73,538 (at 48% tax rate)
Payback =	6.5 years
Return on investment	
(after taxes) =	15%

This example was chosen to show the more general case in which more than one effect is added. The case of converting a double-effect evaporator to three effects should be considered first. In fact, the payback for converting to a triple-effect evaporator is 4.4 years with a 23% return on investment, which is more attractive than conversion to a quadruple-effect evaporator.

Since, in this case, the preliminary results indicate that conversion from a double-effect to a triple-effect evaporator is attractive, the project is continued. Before such a major commitment is made, however, detailed heat and material balances must be conducted to determine the exact heat-transfer area required and the actual attainable savings. At this point or earlier, evaporator vendors should be contacted for budget prices.

As an example, a detailed and thorough technical and economic analysis has been conducted for a specific case. Figure 69 shows the rate of return (discounted cash flow) over a range of evaporation rates for increasing the number of effects by one. The lower portion of each curve represents fixed steam costs at \$3/1000 lb; the upper limit of each curve represents steam at \$3/1000 lb for the first year and a 10% escalation for every year thereafter. Because of the variable steam cost, interest rate of return, based on the discounted cash flow method, was chosen over a simpler ROI analysis. These curves clearly show that the best returns are obtained when converting a single-effect to a double-effect evaporator. The rate of return increases at higher rates of evaporation. The economics for converting a four-effect to five-effect is marginal.

Several publications (A.I.Ch.E. Equipment Testing Procedures, Evaporators) and textbooks discuss heat and material balances in multieffect evaporators. In the next section, the following calculations are outlined in detail to illustrate previous discussions.

1. Determining heat and material balances for a forward-feed evaporator from a pressure profile.

2. Determining heat and material balances from heat-transfer coefficients for a forward-feed triple-effect evaporator.

3. Determining heat and material balances for a triple-effect backward-feed evaporator.

4. Determining additional area requirements for a new effect. By using these methods, an accurate estimate of the performance of the evaporator under the new conditions can be made.

CALCULATIONS

Multieffect evaporator calculations require a trial-and-error procedure. The calculations can be approached from a known pressure profile or from a heat-transfer coefficient (U) profile. This section describes the approaches in detail

Figure 69. After-tax interest return on investment (discounted cash flow) over a range of evaporation rates for adding effects. Basis: Natural-circulation evaporator with thermosiphon; evaporator shell, carbon steel construction; wetted parts, 304 stainless steel; straight-line depreciation method; 10-year lifetime; no salvage value; heat-transfer coefficients are a function of temperature and concentration only; feed at 150°F; 72% evaporation ratio (vapor/feed).

and gives several examples that will be helpful in applications for existing evaporators.

To approach the calculations from a pressure profile essentially means that the pressure in each effect of the evaporator is known. Figure 70a is a logic diagram that shows the method of calculation. A U profile means that the heat-transfer coefficients are known whereas the pressures are not. The logic diagram for this procedure is shown in Fig. 70b.

Determining the Heat and Material Balance for a Forward-Feed Evaporator from a Pressure Profile

The known pressures in each effect of the evaporator are used to carry out an effect-by-effect heat and material balance. A known temperature profile can be used, but usually pressures are more reliable and readily available. In addition to known pressures, accurate information is required for an overall material balance. For example, if the feed rate and concentration are known, together with the product concentration, the total rate of evaporation can be easily calculated. As an alternative, if the feed rate and product rate are known, the vapor rate can also be easily obtained.

The following conditions are assumed for this example (see Fig. 63a):

$$\begin{aligned} F &= 17,500 \text{ lb/hr} \\ X_F &= 18\% \\ T_F &= 182^\circ\text{F} \end{aligned}$$

$$\begin{aligned} P_S &= 18.9 \text{ psia} \\ T_S &= 225^\circ\text{F} \\ X_{Pr} &= 42\% \end{aligned}$$

Pressure Profile (pressure in each effect):

$$\begin{aligned} P_{V_1} &= 10.2 \text{ psia (194}^\circ\text{F sat.)} \\ P_{V_2} &= 5.8 \text{ psia (169}^\circ\text{F sat.)} \\ P_{V_3} &= 0.95 \text{ psia (100}^\circ\text{F sat.) or 28 in. Hg (vacuum pressure)} \end{aligned}$$

Note: P_{V_3} = pressure at the condenser unless there is a significant pressure drop through the vapor piping.

Overall material balance (triple-effect unit):

$$\begin{aligned} \text{Solids in the feed} &= 17,500 \times 0.18 = 3150 \text{ lb/hr} \\ \text{Solids in the product} &= 3150/0.42 = 7500 \text{ lb/hr} \\ \text{Total rate of evaporation} &= 17,500 - 7500 = 10,000 \text{ lb/hr} \\ \text{Approximate steam rate} &= 10,000/(0.9 \times 3) = 3700 \text{ lb/hr} \end{aligned}$$

A heat and material balance around each effect, as illustrated in Fig. 71, yields the following formulas:

For the vapor rate from the i th effect,

$$V_i = \frac{Q_i - (h_i - h_{i-1})L_{i-1}}{(H_i - h_i)}$$

For the liquid rate from the i th effect,

$$L_i = L_{i-1} - V_i$$

(a)

(b)

Figure 70. Logic diagrams for evaporator heat and material balances (a) from pressure profile and (b) from U profile.

where h = enthalpy of the liquid
 H = enthalpy of the vapor
 Q = heat-transfer rate ($\lambda_{i-1} V_{i-1}$)
 λ = latent heat of vaporization

If the heat of solution is negligible and the heat capacity of L_i is assumed equal to L_{i-1} , the formula for V_i can be simplified as

$$\text{Heat-transfer rate } (Q_i) = \lambda_{i-1} V_{i-1}$$

$$\text{Less heating or flash load} = cp(T_{L_{i-1}} - T_{L_i})L_{i-1}$$

$$\text{Net from above} = [H_i - cp(T_{L_i} - 32)] V_i$$

Figure 71. Examples of vapor and liquor designations for heat and material balance around each effect.

For small BPR and heat capacities close to water (1.0), the last expression can be simplified to $\lambda_i V_i$.

We now proceed with the simplified method.

First Effect

Assume BPR = 1.0°F:

$$T_{L_1} = T_{V_1} + \text{BPR}_1 = 194 + 1 = 195^\circ\text{F}$$

Assume steam = 3700 lb/hr:

$$\text{Heat transferred} = 962 \times 3700 = 3.56 \times 10^6 \text{ Btu/hr}$$

$$\text{Less heat for feed} = 0.88(195 - 182)17,500 = 0.20 \times 10^6 \text{ Btu/hr}$$

$$\text{Heat in first vapors} = 3.36 \times 10^6 \text{ Btu/hr}$$

$$V_1 = (3.36 \times 10^6)/981 = 3425 \text{ lb/hr}^*$$

$$L_1 = 17,500 - 3425 = 14,075 \text{ lb/hr}$$

$$X_1 = (17,500 \times 0.18)/14,075 = 0.22$$

The BPR for this concentration is 1.1°F. This is very close to the assumed value. Since the difference is not significant, there is no need to revise the previous calculations, and we can proceed with the next effect.

Second Effect

Assume BPR₂ = 1.5°F:

$$T_{L_2} = T_{V_2} + \text{BPR}_2 = 169 + 1.5 = 170.5^\circ\text{F}$$

$$\text{Heat transferred in second effect} = 3.36 \times 10^6 \text{ Btu/hr}$$

$$\text{Add flash from } L_1 = 0.86(195 - 170.5)14,075 = 0.30 \times 10^6 \text{ Btu/hr}$$

$$\text{Heat in second vapors } (Q_2) = 3.66 \times 10^6 \text{ Btu/hr}$$

$$V_2 = (3.66 \times 10^6)/996 = 3675 \text{ lb/hr}$$

$$L_2 = 14,075 - 3675 = 10,400 \text{ lb/hr}$$

$$X_2 = (14,075 \times 0.22)/10,400 = 0.30$$

Again, we check the BPR for the calculated concentration and find BPR₂ = 1.5. This is the same as assumed, and we continue.

Third Effect

Assume BPR₃ = 3.0°F:

$$T_{L_3} = T_{V_3} + \text{BPR}_3 = 100 + 3 = 103^\circ\text{F}$$

*Note: If the more rigorous expression is used, this becomes

$$V_1 = \frac{3.36 \times 10^6}{1143 - 0.88(195 - 32)} = 3361 \text{ lb/hr}$$

Also, the heat transferred in the second effect becomes

$$Q_2 = \lambda_1 V_1 = 981 \times 3361 = 3.30 \times 10^6 \text{ Btu/hr}$$

$$\text{Heat transferred in third effect } (Q_2) = 3.66 \times 10^6 \text{ Btu/hr}$$

$$\text{Add for flash} = 0.82(170.5 - 103)10,400 = 0.58 \times 10^6 \text{ Btu/hr}$$

$$\text{Heat in third vapors } (Q_3) = 4.24 \times 10^6 \text{ Btu/hr}$$

$$V_3 = (4.24 \times 10^6)/1036 = 4093 \text{ lb/hr}$$

$$L_3 = 10,400 - 4093 = 6307 \text{ lb/hr}$$

$$X_3 = 0.50$$

Since the required product concentration is 42%, we have obviously estimated the steam too high. Incidentally, the BPR for the resulting concentration is 4°F, which is too high. However, since a new assumption for the steam rate is required, no attempt is made to correct the assumed BPR at this time.

A very useful formula for estimating a new or revised steam rate is

$$S \text{ revised} = S \text{ estimated} + \frac{L \text{ calculated} - L \text{ required}}{\text{number of effects}}$$

By substitution,

$$S \text{ revised} = 3700 + \frac{6307 - 7500}{3} = 3300 \text{ lb/hr}$$

Therefore, for the second iteration the steam assumed is 3300 lb/hr.

First Effect

$$\text{Heat transferred} = 3.17 \times 10^6 \text{ Btu/hr}$$

$$\text{Less heat for feed} = 0.20 \times 10^6 \text{ Btu/hr}$$

$$\text{To second effect} = 2.97 \times 10^6 \text{ Btu/hr}$$

$$V_1 = 3030 \text{ lb/hr}$$

$$L_1 = 14,470 \text{ lb/hr}$$

$$X_1 = 0.22$$

Second Effect

$$\text{Heat transferred} = 2.97 \times 10^6 \text{ Btu/hr}$$

$$\text{Add for flash} = 0.30 \times 10^6 \text{ Btu/hr}$$

$$\text{To third effect} = 3.27 \times 10^6 \text{ Btu/hr}$$

$$V_2 = 3285 \text{ lb/hr}$$

$$L_2 = 11,185 \text{ lb/hr}$$

$$X_2 = 0.28$$

Third Effect

$$\text{Heat transferred} = 3.27 \times 10^6 \text{ Btu/hr}$$

$$\text{Add for flash} = 0.62 \times 10^6 \text{ Btu/hr}$$

$$\text{To condenser} = 3.89 \times 10^6 \text{ Btu/hr}$$

$$V_3 = 3755 \text{ lb/hr}$$

$$L_3 = 7430 \text{ lb/hr}$$

$$X_3 = 0.42$$

Total evaporation rate = $17,500 - 7430 = 10,070$ lb/hr. This is very close to the required rate of evaporation. For a final computation of the steam rate, we can use the following formula for revising steam rate,

$$S = 3300 + \frac{7430 - 7500}{3} = 3275 \text{ lb/hr}$$

This is the theoretical steam required. The evaporation rate of water removed per pound of steam is $10,000/3275 = 3.05$.

The heat-transfer coefficients can now be calculated from the following known areas:

$$A_1 = 345 \text{ sq ft}$$

$$A_2 = 540 \text{ sq ft}$$

$$A_3 = 820 \text{ sq ft}$$

The equation for overall heat-transfer coefficient is

$$U = \frac{Q}{A \Delta T}$$

For the first effect,

$$\Delta T_1 = 225 - 195 = 30^\circ \text{F}$$

$$Q_1 = 3.17 \times 10^6 \text{ Btu/hr}$$

By substitution,

$$U_1 = \frac{3.17 \times 10^6}{30 \times 345} = 306 \text{ Btu/(hr)(sq ft)}(^{\circ}\text{F})$$

also,

$$U_2 = 234 \text{ Btu/(hr)(sq ft)}(^{\circ}\text{F})$$

$$U_3 = 60 \text{ Btu/(hr)(sq ft)}(^{\circ}\text{F})$$

The cooling water flow to the condenser, in gallons per minute, can be calculated from the following formula:

$$\text{gpm} = \frac{V(\lambda + T_A)}{500(T_V - T_W - T_A)} \quad (\text{barometric condenser})$$

where V = vapor flow to the condenser, lb/hr

λ = latent heat of the vapors to the condenser, Btu/lb

T_V = saturated temperature of the vapors at the condenser, $^{\circ}\text{F}$

T_W = temperature of the cooling water, $^{\circ}\text{F}$

T_A = approach temperature. (This is a measure of the efficiency of the condenser and is the difference between the temperature of the water leaving the condenser and the saturated vapors.)

Assume $T_A = 5^{\circ}\text{F}$ and $T_W = 80^{\circ}\text{F}$. By substitution,

$$\text{gpm} = \frac{3755(1036 - 5)}{500(100 - 80 - 5)} = 516$$

Determining the Heat and Material Balance from Heat-Transfer Coefficients

It is found for the evaporator in the preceding example that the steam pressure can be increased from 18.9 to 22.0 psia. Under the same feed conditions, what product concentration can be obtained by raising the steam pressure to 22 psia, or a temperature of 233°F ?

The feed conditions remain the same. However, the pressure in each effect will change, and the pressure profile becomes an unknown. The only known pressures are those of the steam to the chest of the first effect and the vacuum in the condenser. The heat and material balance must now be based on heat-transfer coefficients. The following quantities are known:

$$F = 17,500 \text{ lb/hr}$$

$$T_F = 182^{\circ}\text{F}$$

$$X_F = 18\%$$

$$P_S = 22 \text{ psia}$$

$$T_S = 233^{\circ}\text{F}$$

$$P_V = \text{condenser pressure} = 0.95 \text{ psia}$$

$$U_1 = 306 \text{ Btu/(hr)(sq ft)}(^{\circ}\text{F})$$

$$U_2 = 234 \text{ Btu/(hr)(sq ft)}(^{\circ}\text{F})$$

$$U_3 = 60 \text{ Btu/(hr)(sq ft)}(^{\circ}\text{F})$$

The same heat-transfer coefficients are used as those calculated in the previous example. This is a valid assumption because only minor changes take place in the operating conditions.

The first step in the calculations is to assume a steam rate. This can be done by adjusting the present steam rate in proportion to the new steam temperature. Since Total ΔT has increased from 125°F to 133°F , the steam is estimated as

$$S = 3275 \times (133/125) = 3485 \text{ lb/hr}$$

Use 3500 lb/hr. The temperature of the boiling liquid in the first effect is calculated from

$$\Delta T_1 = \frac{Q_1}{U_1 A_1}$$

where $Q_1 = \lambda S = 957 \times 3500 = 3.35 \times 10^6$ Btu/hr. By substitution,

$$\Delta T_1 = \frac{3.35 \times 10^6 \text{ Btu/hr}}{306 \times 345} = 31.7^{\circ}\text{F}$$

The temperature of the liquid in the first effect is

$$T_{L_1} = T_S - \Delta T_1 = 233 - 31.7 = 201.3^{\circ}\text{F}$$

We can now proceed with a heat and material balance around the first effect.

First Effect

Heat transferred to first effect $= \lambda S = 3.35 \times 10^6$ Btu/hr
Less heat for feed

$$= 0.88(201.3 - 182)17,500 = 0.30 \times 10^6 \text{ Btu/hr}$$

Net to second effect (Q_2) $= 3.05 \times 10^6$ Btu/hr

$$V_1 = (3.05 \times 10^6)/978 = 3118 \text{ lb/hr}$$

$$L_1 = 17,500 - 3118 = 14,382 \text{ lb/hr}$$

$$X_1 = (17,500 \times 0.18)/14,382 = 0.22$$

From known physical data, $BPR_1 = 1.1^\circ\text{F}$. The saturated temperature of the vapor is

$$T_{V_1} = T_{L_1} - BPR_1 = 201.3 - 1.1 = 200.2^\circ\text{F}$$

From the steam tables, $P_{V_1} = 11.5$ psia.

Second Effect

$$\Delta T_2 = \frac{Q_2}{U_2 A_2} = \frac{3.05 \times 10^6}{234 \times 540} = 24.1^\circ\text{F}$$

$$T_{L_2} = T_{V_1} - \Delta T_2 = 200.2 - 24.1 = 176.1^\circ\text{F}$$

Heat transferred (Q_2) $= 3.05 \times 10^6$ Btu/hr

Add for flash

$$= 0.86(201.3 - 176.1)14,382 = 0.31 \times 10^6 \text{ Btu/hr}$$

Net to third effect (Q_3) $= 3.36 \times 10^6$ Btu/hr

$$V_2 = (3.36 \times 10^6)/993 = 3384 \text{ lb/hr}$$

$$L_2 = 14,382 - 3384 = 10,998 \text{ lb/hr}$$

$$X_2 = 0.29$$

$$BPR_2 = 1.5$$

$$T_{V_2} = 176.1 - 1.5 = 174.6$$

$$P_{V_2} = 6.6 \text{ psia}$$

Third Effect

$$\Delta T_3 = \frac{3.36 \times 10^6}{60 \times 820} = 68.3^\circ\text{F}$$

$$T_{L_3} = 174.6 - 68.3 = 106.3^\circ\text{F}$$

Heat transferred (Q_3) $= 3.36 \times 10^6$ Btu/hr

Add for flash

$$= 0.82(176.1 - 106.3)10,998 = 0.63 \times 10^6 \text{ Btu/hr}$$

Net to condenser $= 3.99 \times 10^6$ Btu/hr

$$V_3 = (3.99 \times 10^6)/1036 = 3851 \text{ lb/hr}$$

$$L_3 = 10,998 - 3851 = 7147 \text{ lb/hr}$$

$$X_3 = 0.44$$

$$BPR_3 = 3^\circ\text{F}$$

$$T_{V_3} = 106.3 - 3 = 103^\circ\text{F}$$

With an assumption of no pressure drop in the vapor piping, the resulting saturated vapor temperature at the condenser is 103°F . Since the pressure in the condenser is to be maintained at 0.95 psia, or 100°F (saturated), the assumed steam is incorrect. The required temperature at the condenser is lower than calculated; this means that the

evaporator can support a higher steam rate than assumed. A higher steam rate will be used in the next iteration. If the resulting condenser temperature is still significantly different from the required value, another value for the steam rate is assumed until the desired accuracy is obtained. It is recommended that the results from each iteration be plotted to estimate the steam rate for the next iteration. At least three points are necessary before the plot can be used; interpolating between these points usually produces an acceptable answer.

The reader can verify that a steam rate of 3570 lb/hr will yield the desired result, as follows:

Steam rate = 3570 lb/hr

$$P_{V_1} = 11.5 \text{ psia}$$

$$P_{V_2} = 6.7 \text{ psia}$$

$$P_{V_3} = 0.95 \text{ psia}$$

$$V_1 = 3180 \text{ lb/hr}$$

$$V_2 = 3475 \text{ lb/hr}$$

$$V_3 = 3980 \text{ lb/hr}$$

Product concentration = 45.9%

Total evaporation = 10,635 lb/hr

Evaporation per pound of steam = 2.98

Calculations for a Backward-Feed Triple-Effect Evaporator

The calculational methods given in the previous examples can be applied to backward-feed evaporators with minor modifications. For a forward-feed triple-effect evaporator, we assumed the steam rate and carried out an effect-by-effect heat and material balance; if the calculated vapor pressure in the last effect did not match the target pressure, a new steam rate was assumed until a match was obtained. However, in backward-feed evaporators most of the known information is associated with the last effect, not with the steam end. Therefore it is more convenient to begin the calculations at the condenser end, by assuming the vapor flow to the condenser (instead of steam in) and to proceed from there. It is equally valid to assume a temperature difference for heat transfer in the last effect and to start from that point. This approach based on assuming ΔT 's is shown in the calculations to follow.

Consider a backward-feed triple-effect evaporator concentrating a liquid solution under the following conditions (see Fig. 63b):

$$F = 17,500 \text{ lb/hr}$$

$$T_F = 90^\circ\text{F}$$

$$X_F = 18\%$$

$$P_S = 18.9 \text{ psia}$$

$$T_S = 225^\circ\text{F}$$

$$P_{V_3} = \text{condenser pressure} = 0.95 \text{ psia}$$

$$U_1 = 280 \text{ Btu/(hr)(sq ft)}(^{\circ}\text{F})$$

$$\begin{aligned}
 U_2 &= 200 \text{ Btu/(hr)(sq ft)}(^{\circ}\text{F}) \\
 U_3 &= 100 \text{ Btu/(hr)(sq ft)}(^{\circ}\text{F}) \\
 A_1 &= 345 \text{ sq ft} \\
 A_2 &= 540 \text{ sq ft} \\
 A_3 &= 820 \text{ sq ft} \\
 \text{Total } \Delta T &= 225 - 100 = 125^{\circ}\text{F}
 \end{aligned}$$

This represents approximately 40°F ΔT per effect, and it will be used as the assumed ΔT in the third effect of the evaporator. Now we proceed with the heat and material balance, starting with the last effect.

Third Effect

Assume $\text{BPR}_3 = 1^{\circ}\text{F}$.

$$T_{L_3} = T_{V_3} + \text{BPR}_3 = 100 + 1 = 101^{\circ}\text{F}$$

Assume $\Delta T_3 = 40^{\circ}\text{F}$. Since $Q_3 = U_3 A_3 \Delta T_3$, then

$$\begin{aligned}
 \text{Heat transferred}(Q_3) &= 100 \times 820 \times 40 = 3.28 \times 10^6 \text{ Btu/hr} \\
 \text{Less heat for feed} & \\
 &= 0.88(101 - 90)17,500 = 0.17 \times 10^6 \text{ Btu/hr}
 \end{aligned}$$

$$\text{To condenser} = 3.11 \times 10^6 \text{ Btu/hr}$$

$$V_3 = (3.11 \times 10^6)/1104 = 2817 \text{ lb/hr}$$

$$L_3 = 17,500 - 2817 = 14,683 \text{ lb/hr}$$

$$X_3 = 0.21 \text{ (BPR}_3 \text{ assumption is correct)}$$

V_3 could have been assumed, instead of ΔT_3 , and used as a starting point.

Second Effect

Since $Q_3 = 3.280 \times 10^6 \text{ Btu/hr}$ ($U_3 A_3 \Delta T_3$).

$$T_{V_2} = T_{L_3} + \Delta T_3 = 101 + 40 = 141^{\circ}\text{F}$$

$$V_2 = (3.280 \times 10^6)/1012 = 3241 \text{ lb/hr}$$

$$L_2 = 14,683 - 3241 = 11,442 \text{ lb/hr}$$

$$X_2 = 0.27$$

$$\text{BPR}_2 = 1.5^{\circ}\text{F}$$

$$T_{L_2} = 141 + 1.5 = 142.5^{\circ}\text{F}$$

$$\text{Heat in second vapors } (Q_2) = 3.28 \times 10^6 \text{ Btu/hr}$$

$$\begin{aligned}
 \text{Heating } L_3 & \\
 &= 0.86(142.5 - 101)14,683 = 0.52 \times 10^6 \text{ Btu/hr}
 \end{aligned}$$

$$\text{Heat transferred } (Q_2) = 3.80 \times 10^6 \text{ Btu/hr}$$

$$\Delta T_2 = \frac{Q_2}{U_2 A_2} = \frac{3.80 \times 10^6}{200 \times 540} = 35.2^{\circ}\text{F}$$

First Effect

Since $Q_2 = 3.80 \times 10^6 \text{ Btu/hr}$.

$$T_{V_1} = 142.5 + 35.2 = 177.7^{\circ}\text{F}$$

$$V_1 = (3.80 \times 10^6)/990 = 3838 \text{ lb/hr}$$

$$L_1 = 11,442 - 3838 = 7604 \text{ lb/hr}$$

$$X_1 = 0.41$$

$$\text{BPR}_1 = 3^{\circ}\text{F}$$

$$T_{L_1} = 177.7 + 3 = 180.7^{\circ}\text{F}$$

$$\text{Heat in first vapors} = 3.80 \times 10^6 \text{ Btu/hr}$$

$$\begin{aligned}
 \text{Heating } L_2 & \\
 &= 0.82(180.7 - 142.5)11,442 = 0.36 \times 10^6 \text{ Btu/hr}
 \end{aligned}$$

$$\text{Heat transferred } (Q_1) = 4.16 \times 10^6 \text{ Btu/hr}$$

$$\Delta T_1 = \frac{Q_1}{A_1 U_1} = \frac{4.16 \times 10^6}{280 \times 345} = 43.1^{\circ}\text{F}$$

$$\text{Steam temperature} = 180.7 + 43.1 = 223.8^{\circ}\text{F}$$

This is almost the required temperature of the steam (225°F), and the correct solution to the problem has been obtained. We proceed with the remaining calculations.

$$\text{Steam flow} = (4.16 \times 10^6)/962 = 4324 \text{ lb/hr}$$

$$\text{Evaporation per pound of steam} = 9896/4324 = 2.29$$

Determining Additional Area Requirements for a New Effect

Let us consider adding a new effect to the existing triple-effect evaporator, in the second example ("Determining the Heat and Material Balance from Heat-Transfer Coefficients"). For the same feed conditions, we want to obtain a product concentration of 45%. Steam is at 22 psia and 233°F . Calculate the area required in a fourth effect. Since the new effect will introduce an additional BPR, we assume the worst possible condition of an additional $\text{BPR} = 3.2^{\circ}\text{F}$ (equal to that expected in the last effect). The $\Delta T_{\text{available}} = 233 - 100 = 133^{\circ}\text{F}$; therefore, because of the BPR, the new area introduces a ΔT loss of 2.5%.

Furthermore, from existing operating data we decide that the heat-transfer coefficients in the four effects will be 315, 230, 150, and 60 $\text{Btu/(hr)(sq ft)}(^{\circ}\text{F})$. If we calculate an average heat-transfer coefficient, we obtain 200 $\text{Btu/(hr)(sq ft)}(^{\circ}\text{F})$ for the existing evaporator and 189 $\text{Btu/(hr)(sq ft)}(^{\circ}\text{F})$ under the new conditions, the latter representing a 5.5% loss in the heat-transfer rate.

The total existing heat-transfer area is 1705 sq ft, or an average of 568 sq ft per effect. As a first estimate for the new area, we would use 568 sq ft plus an additional amount to compensate for all of the losses mentioned, which add up to an 8% loss in capacity.

$$\text{New area} = 1705 \times 0.08 + 568 \text{ sq ft} = 704 \text{ sq ft}$$

We will use 700 sq ft in our calculations.

Since the steam required for three effects was 1 lb for every 3 lb of evaporation for three effects, we will use 3.9 lb of evaporation per pound of steam for four effects as a first approximation.

$$\text{Required evaporation} = 10,500 \text{ lb/hr}$$

$$\text{Steam} = 10,500/3.9 = 2690 \text{ lb/hr (approximation)}$$

$$\text{Therefore } Q_1 = 2690 \times 957 \times 2.57 \times 10^6 \text{ Btu/hr}$$

First Effect

$$\Delta T_1 = \frac{2.57 \times 10^6}{315 \times 345} = 23.7^\circ\text{F}$$

$$T_{L_1} = T_S - \Delta T_1 = 233 - 23.7 = 209.3^\circ\text{F}$$

$$T_{V_1} = T_{L_1} - \text{BPR}_1 = 209.3 - 1 = 208.3^\circ\text{F}$$

$$\text{Heat transferred } (Q_1) = \lambda S = 2.57 \times 10^6 \text{ Btu/hr}$$

Less heat for feed

$$= 0.88(209.3 - 182)17,500 = 0.42 \times 10^6 \text{ Btu/hr}$$

$$\text{To first vapors} = 2.15 \times 10^6 \text{ Btu/hr}$$

$$V_1 = (2.15 \times 10^6)/973 = 2210 \text{ lb/hr}$$

$$L_1 = 17,500 - 2210 = 15,290 \text{ lb/hr}$$

$$X_1 = 0.21 \text{ (BPR}_1 \text{ assumption is correct)}$$

Second Effect

$$\Delta T_2 = \frac{2.15 \times 10^6}{230 \times 540} = 17.3^\circ\text{F}$$

$$T_{L_2} = T_{V_1} - \Delta T_2 = 208.3 - 17.3 = 191^\circ\text{F}$$

$$T_{V_2} = T_{L_2} - \text{BPR}_2 = 191 - 1 = 190^\circ\text{F}$$

$$\text{Heat in first vapor} = 2.15 \times 10^6 \text{ Btu/hr}$$

Add for flash

$$= 0.86(209.3 - 191)15,290 = 0.24 \times 10^6 \text{ Btu/hr}$$

$$\text{To second vapors} = 2.39 \times 10^6 \text{ Btu/hr}$$

$$V_2 = (2.39 \times 10^6)/985 = 2426 \text{ lb/hr}$$

$$L_2 = 15,290 - 2426 = 12,864 \text{ lb/hr}$$

$$X_2 = 0.24$$

Third Effect

$$\Delta T_3 = \frac{2.39 \times 10^6}{150 \times 820} = 19.4^\circ\text{F}$$

$$T_{L_3} = T_{V_2} - \Delta T_3 = 190 - 19.4 = 170.6^\circ\text{F}$$

$$T_{V_3} = T_{L_3} - \text{BPR}_3 = 170.6 - 2 = 168.6^\circ\text{F}$$

$$\text{Heat in second vapor} = 2.39 \times 10^6 \text{ Btu/hr}$$

Add for flash

$$= 0.84(191 - 170.6)12,864 = 0.22 \times 10^6 \text{ Btu/hr}$$

$$\text{To third vapors} = 2.61 \times 10^6 \text{ Btu/hr}$$

$$V_3 = (2.61 \times 10^6)/997 = 2618 \text{ lb/hr}$$

$$L_3 = 12,864 - 2618 = 10,246 \text{ lb/hr}$$

$$X_3 = 0.31$$

Fourth Effect

$$\Delta T_4 = \frac{2.61 \times 10^6}{60 \times 700} = 62.1^\circ\text{F}$$

$$T_{L_4} = T_{V_3} - \Delta T_4 = 168.6 - 62.1 = 106.5^\circ\text{F}$$

$$T_{V_4} = T_{L_4} - \text{BPR}_4 = 106.5 - 3 = 103.5^\circ\text{F}$$

$$\text{Heat in third vapor} = 2.61 \times 10^6 \text{ Btu/hr}$$

Add for flash

$$= 0.82(170.6 - 106.5)10,246 = 0.54 \times 10^6 \text{ Btu/hr}$$

$$\text{To fourth vapors (condenser)} = 3.15 \times 10^6 \text{ Btu/hr}$$

$$V_4 = (3.15 \times 10^6)/1035 = 3043 \text{ lb/hr}$$

$$L_4 = 10,246 - 3043 = 7203 \text{ lb/hr}$$

$$X_4 = 0.44$$

$$\text{Total evaporation} = 10,297 \text{ lb/hr}$$

$$\text{Evaporation per pound of steam} = 3.83$$

However, since the required evaporation is 10,500 lb/hr, we can better estimate the steam flow as

$$\text{Steam} = 10,500/3.83 = 2740 \text{ lb/hr}$$

This value for steam flow could be used in the next iteration, and a new vapor temperature to the condenser could be calculated along with a new product concentration. The procedure is repeated until the desired accuracy is obtained in the calculated vapor temperature. At this point, if the calculated concentration is too low, a larger area is assumed and the procedure is repeated. In our example we consider that the results are close enough for our purposes. Notice that with the higher steam flow, as estimated previously, the calculated concentration would be somewhat higher and closer to the required 45%.

The preceding procedure is a general one that is applicable to a new area at any position in the evaporator train. We could in fact modify the procedure when the new area under consideration corresponds to the last effect, as is the case here. Follow the same procedure as in the previous example up to the third effect. Then, instead of assuming the new area, calculate T_{L_3} as follows:

$$T_{L_3} = T_{V_4} + \text{BPR}_4 = 100 + 3 = 103^\circ\text{F}$$

$$\text{then } \Delta T_4 = T_{L_3} - T_{V_3} = 168.6 - 103 = 65.6^\circ\text{F}$$

$$\text{A new area} = \frac{Q_3}{U_4 \Delta T_4} = \frac{2.61 \times 10^6}{60 \times 65.6} = 663 \text{ sq ft}$$

$$\text{Heat transferred} = 2.61 \times 10^6 \text{ Btu/hr}$$

Add for flash

$$= 0.82(170.6 - 103)10,246 = 0.54 \times 10^6 \text{ Btu/hr}$$

$$\text{To fourth vapors} = 3.15 \times 10^6 \text{ Btu/hr}$$

$$V_4 = (3.15 \times 10^6)/1035 = 3043 \text{ lb/hr}$$

$$L_4 = 7203 \text{ lb/hr}$$

$$X_4 = 0.44$$

If greater accuracy is desired, a new steam rate would be assumed, and the procedure would be repeated until the desired precision is obtained. Notice that in this procedure no guess is required for the new area. The only drawback is that the method is impracticable unless the new area under consideration is at the last effect.

Chapter 5

GUIDELINES FOR FORMULATING AN UPGRADING PROGRAM

INTRODUCTION

The various alternatives, calculation procedures, and economics for the upgrading of existing evaporators have been discussed in the previous chapters. Not only is the engineer responsible for this study faced with determining the technical feasibility of each upgrading technique, but he also must evaluate the best combination of techniques that will optimize the economics and minimize potential operating problems. In addition, he must coordinate the upgrading steps with existing plant operation and production schedules to ensure maximum return on the costs for the study. However, the effect of higher fuel costs on the costs of evaporation is such that, in many cases, ways can be found to economically reduce the operating costs of evaporation by reducing energy usage.

It is important to realize at the beginning that it is more difficult and more time consuming to upgrade existing evaporators than to install new evaporators in new plants, in part because of the need to evaluate the retrofit costs of the changes into the existing plant. Also, the engineer must reconstruct the past design rationale and set the basis for the evaluation by determining the present operating conditions. Analysis must be done case by case for every single plant. No two plants will give identical results. Retrofitting costs, which are a key item in the economics, are very difficult to estimate accurately because retrofitting may require modifying buildings or other production equipment as well as the evaporation systems. Retrofit jobs usually cost more and take longer than originally estimated, since unexpected problems inevitably arise during the retrofit period.

Quite often the retrofit construction and plant modifications may result in production loss, with the prospect of further losses if the construction schedule slips. Many manufacturers have experienced these unplanned and unestimated costs and will not undertake major retrofits unless the economics is sufficiently favorable to absorb a large amount of unexpected costs. Planning and coordinat-

ing the modifications with the production schedule will greatly reduce the risks of production losses.

Another major factor in an extensive upgrading analysis is the availability of process engineers who have the time to do the detailed engineering and economic evaluations required to establish the techniques, to complete the preliminary economics, and to select the best upgrading techniques. The staffing of most corporations is barely sufficient to operate existing facilities and carry out minor maintenance improvements. Unless there are incentives for improving capacity or product quality, it sometimes becomes easier to delay studies associated with reducing operating costs than to make the effort required. However, reducing operating costs by reducing energy consumption for evaporators can be a most effective way of increasing profits.

An upgrading program can be undertaken even with the negative factors just mentioned. Steps for planning and implementing such a program are given in Figure 72. Formulating and following a plan will help avoid erroneous conclusions and the necessity to repeat steps. Since the various upgrading techniques affect each other, a complete analysis of possible combinations of methods is needed to determine the best new operating conditions and economics.

EVALUATING THE EXISTING EVAPORATOR

The first and most important part of an upgrading program for energy conservation is a thorough evaluation of the existing equipment, as shown in steps 1 to 5 in Fig. 72. There are two important reasons for this. First, an evaluation of the existing system will determine the problem areas, if any, and at the same time provide a base point from which to measure potential savings, as described in Chapter 2. Very often this step will discover savings that can be achieved by minor changes in operating procedures and can be implemented from maintenance and operating funds. Second, a thorough evaluation will provide a wealth

Figure 72. Steps for planning, implementing, upgrading evaporators.

of operating data, which will be invaluable in later phases of the upgrading program.

The evaluation should include heat and material balances so that steam consumption, evaporation rates, heat-transfer coefficients, etc., can be obtained. The evaluation can also examine major consumers of thermal energy outside the evaporator area, especially heat exchangers that service cold process streams.* The most efficient heat-recovery system is usually one that can be used as a heat sink to receive waste heat, and the best location for this may be in another area which can readily receive a high-temperature stream from the evaporator.

Throughout this publication, many variations to the "standard" evaporator have been discussed. Many of these applied only to "special circumstances"; yet all plants are different, and unusual circumstances are to be found in each of them. This individuality provides fertile ground for the technical personnel, familiar with the overall process, to use their knowledge and imagination to exploit these circumstances. Managers should encourage their plant personnel to submit "brainstorm ideas." Those familiar with the plant are in the best position to know if there are any potential uses for waste heat.

OUTSIDE ASSISTANCE

Plants with a limited technical staff can obtain outside help from equipment manufacturing firms, from major processors within the industry, or from independent consulting firms. Outside assistance in identifying the options is a good method and normally can be funded at a lower cost than in-plant studies. Equipment manufacturers offer their services at no charge. However, this type of service is very limited since the vendor's staff are also deeply involved in design and construction of plants. For extensive studies or full-scale tests, it may be necessary to contract for their services. This could be the case where very little data exist on the process and changes to the evaporator are radical. Although major processors offer technical know-how, very often their information is proprietary in nature, or they simply may be unwilling to give up their hard-earned information. Independent consulting firms will fill this gap between plant personnel and vendors by providing inde-

pendent evaluations, and they can be very valuable to companies with a small technical staff or to companies unwilling to give up their engineering time for programs not related to the day-to-day operating problems of the plant. Also, the time of plant personnel can be used more effectively to implement the upgrading methods once they are identified.

COMPUTER SURVEY

Engineers familiar with the calculations for multieffect evaporators are well aware of their time-consuming nature. Even in their simplified form, the calculations involve trial-and-error solutions that require significant technical time. For example, determining the size of a new heat-transfer surface requires trial-and-error procedures to determine a heat balance for an assumed value of the heat-transfer area. Many trials may be required for different areas, each requiring several steps. If determining variable heat-transfer coefficients is one of the calculation requirements, the task becomes impractical for hand calculations. This is most unfortunate, since there might be numerous ideas on how to upgrade an existing evaporator, but, because of the complexity of many evaporation problems, a simple intuitive analysis is not acceptable. Fortunately, however, a computer program can easily handle these evaluations, and each technique can be verified by accurate heat and material balances. Thus such a program greatly simplifies the task of calculating the energy savings and new process conditions and reduces the costs of the evaluation.

DETERMINING THE APPROPRIATE UPGRADING METHODS

Steps 6 to 8 in Fig. 72 involve determining the upgrading methods and obtaining budget estimates of the equipment from the equipment suppliers. The basis for this evaluation is set by defining process conditions for the existing unit according to the procedures outlined in Chapter 2. It is generally better to evaluate each upgrading method separately and then to determine how the individual techniques could be combined to obtain the maximum savings and return on investment. These initial steps also provide the data for obtaining firm quotes from the vendors once the selection is made.

The first eight steps in Fig. 72 provide the basis for the economic analysis. The steam savings and new process condition for each method or combination of methods that meet the required capacity have been determined, and the purchase costs of the hardware have been estimated.

*There are some industries, such as the sugar industry, in which vapors from evaporators are systematically used to heat process streams. This method of heat recovery is similar to thermal recompression, in which vapors are recycled to the evaporator, but the heat economy is achieved elsewhere in the plant, rather than in the evaporator. This economy is just as valid as one obtained directly in the evaporator.

ECONOMIC ANALYSIS AND IMPLEMENTATION OF THE MODIFICATIONS

The final steps in the program start with a detailed economic analysis of each feasible method. The basis for each has been accurately defined, and the purchase costs for the instruments and upgrading equipment have been obtained. The installation costs must now be estimated, and a projection on the future rise in energy costs should be incorporated into the economics in addition to current costs. Vendors or the consultants doing the study can provide estimates of typical installed costs and general guidance.

After the economic analysis is completed, the decision is made concerning the approach to be used. Then the detailed engineering and design of the installation begins.

The data input to the vendors for design will include the process conditions, and their proposals should show, in addition to price, a detailed heat and material balance. This information should closely approximate the in-plant calculations. If there are discrepancies, the balances should be checked, and the vendors should be contacted to make

certain that accurate data were used for the proposals. It is also advisable to meet with the vendor's technical staff to resolve the problem and to discuss any peculiar operating trends that cannot be discussed in a letter.

The vendors should also make a commitment as to a delivery time for the hardware. These dates should be routinely checked after the orders are placed, and performance guarantees should be evaluated.

After the delivery times are set up, a construction schedule must be developed to coordinate the completion of plant modifications, such as rerouting piping and construction of foundations, with the expected delivery of the equipment and to establish the best timing for the conversion. Proper scheduling will reduce the risks of additional costs associated with interruption of production.

After the new equipment is finally installed and started up, the performance should be evaluated to determine the new process conditions and actual savings. Periodic performance evaluations as described in Chapter 2 will ensure that the system is operating at the optimum conditions and maximum reductions in energy consumption are being achieved.

Chapter 6

NEW TECHNOLOGY

Evaporation is one of the oldest industrial technologies known, and it has undergone an extensive evolution. As this publication has shown, current technology can greatly reduce energy consumption in evaporators. The high cost of energy, and anticipation of further price increases, can be expected to lead to additional innovative technical developments which may lead to future energy conservation.

ADVANCED DESIGNS AND CONTROLS

One of the major new developments that can be expected is the use of more sophisticated design techniques to provide evaporator configurations for optimum energy use and to counteract the expected rise in energy costs. Already computers are used to quantitatively establish heat exchanger designs that are much closer to the optimum than those provided by the old rules of thumb, although the practice is not universal.

Recent improvements in computer technology now allow the designer to investigate a multitude of options. It is expected that the major improvements in evaporator technology will result from the more frequent analyses possible because the computer can perform most of the tedious calculations traditionally associated with evaporation systems.

The use of advanced control techniques on evaporators has been widely studied but only rarely applied. One reason for this is the inherent simplicity of evaporator control if the system is properly designed and operated, together with the fact that much larger savings are possible with the approaches outlined in this publication. As with most process equipment, rapid changes in conditions are deadly to evaporators, and completely level operating conditions add to the energy efficiency. Thus feed-forward control, which automatically changes conditions to account for a change (such as feed temperature) might offer advantages. To the authors' knowledge, however, gains beyond those of conventional feed base systems have not yet been demon-

strated on straight, multieffect evaporators. Such a control system might have advantages for recompression systems, where steam venting or changes of feed condition could possibly be reduced.

Adaptive (or "learning") controllers, which change the tuning, or conventional controllers, after they have monitored the system's response, have proved to be valuable in many sensitive control situations. However, it remains to be shown that these controllers return measurable energy savings in evaporation.

The on-line computer optimization of evaporator conditions to minimize energy use is also an area that, on the surface, could offer promise—but only after the first generation of optimization outlined in this manual is complete.

USE OF LOWER GRADE HEAT

The cost of energy has led to an investigation of the use of streams at temperatures not normally considered usable. The "cooling tower" approach, Fig. 73 has been applied to certain evaporation applications in which low-grade heat is available and the material being concentrated can be contacted with large amounts of air. These systems are used mainly for the concentration of waste streams.

The cooling tower evaporators use large amounts of air in contact with the warm product and operate in a manner almost identical to the normal cooling tower. In fact, some plants have used cooling tower systems for evaporation. In this case the goal is to concentrate, rather than cool, the water by evaporation. The design conditions for such applications are based upon air humidity charts rather than steam tables.

Basically, this approach is suitable only for inorganic solutions in which the solute has no vapor pressure, since odors or part of the solute would be released to the atmosphere if there is significant vapor pressure. Usually these systems have been applied to inorganic salt-containing waste streams, such as boiler blowdown.

Figure 73. Cooling tower evaporator.

USE OF SOLAR ENERGY

Solar evaporation is probably the oldest evaporation technique known to man. Solar evaporation ponds are used in several industries. Escalating energy costs and increased emphasis on the design of energy-recovery hardware has increased the opportunity to use solar energy as a supplemental heat source.

At present, the major industrial use of solar heat is expected to be in the production of hot water or very low pressure steam. Energy at this level can be produced with nonconcentrating solar collectors, such as the one shown in Fig. 74. It is expected that evaporation will be one of the unit operations using much of this low-level energy—second only to drying.

Unfortunately, the economics and applicational design information for the industrial use of solar energy have yet to be assembled. Very little guidance is now available for the process engineer, although there are several good publications* dealing with the subject.

Although nonconcentrating solar collectors providing low-level energy are expected to provide the initial thrust of solar energy into the industrial area, the authors think concentrating collectors (Fig. 75) will be the key that allows solar energy to become widely used in industry. The low-level temperatures reached by the present solar concentrators will bring the energy availability to the point where it becomes a desirable prime energy source and can directly replace fuel rather than compete with condensates and cooling water as a low-availability source of energy.

Figure 74. Flat-plate nonconcentrating solar collector. From Jan F. Kreider and Frank Kreith, *Solar Heating and Cooling: Engineering, Practical Design and Economics*, p. 85, Scripta Book Co., Washington, D.C., 1975.

Figure 75. Concentrating solar collector. From Jan F. Kreider and Frank Kreith, *Solar Heating and Cooling: Engineering, Practical Design and Economics*, p. 89, Scripta Book Co., Washington, D.C., 1975.

*Jan F. Kreider and Frank Kreith, *Solar Heating and Cooling: Engineering, Practical Design and Economics*, Scripta Book Co., Washington, D.C., 1975.

HEAT-TRANSFER ENHANCEMENT

The Office of Saline Water, Department of Interior, has funded a multitude of projects aimed at increasing the energy efficiency of evaporation so that it can be economically applied to the recovery of potable water from brackish water. The use of surfactants to enhance the heat-transfer coefficient was one area investigated. The Environmental Protection Agency extended this concept to some industrial waste waters.* This work has shown that by inducing a foaming flow through surfactants heat-transfer coefficients can be increased up to 150%. Applying this technique to existing evaporators obviously requires an evaluation tailored to the specific situation as well as in-plant testing.

OTHER CONCENTRATION METHODS

There are other methods for concentrating solutions with a theoretical energy use lower than that of evaporation, and much funding has been devoted to developing these technologies. The best known of these are the membrane techniques—reverse osmosis and ultra filtration. Although discussion of these technologies is beyond the scope of this report, review articles are listed in the Bibliography. The membrane technologies have special application when a material is extremely heat sensitive or when only a low concentration level is involved. The major problems are related to the failure of membranes to maintain an economical operating cost over the life of the project and their inability to handle streams containing any solids.

CHANGES IN DESIGN RATIONALE

The following trends in evaporator design can be expected:

- Evaporation applications that, in the past, used single-effect designs because of low capacity or the need for expensive construction materials will increasingly use vapor recompression as a means of improving efficiency. As a minimum, thermal recompression will be used.
- An increased number of effects is now economical for multieffect evaporators. Certain critical capital costs, previously prohibitive, have now become acceptable. Each evaporation system design will be analyzed more closely to define the most economical number of effects.
- Extensive heat exchange between outgoing streams (condensate and product) and the incoming feed will be used.
- New evaporators will be equipped with the instrumentation and measurement devices necessary to allow the operators to monitor closely the performance of operating evaporators and to minimize energy use.
- Increased automated controls will be used to maintain optimum operation.
- Increased maintenance is now justified to keep all parts of the evaporation system in excellent condition.
- There will be less use of evaporation schemes that inhibit the recovery of the latent heat energy of the vaporized water. For example, submerged combustion of evaporators will be used only when absolutely necessary.
- Each evaporation system will be analyzed carefully for each source of energy loss, and conservation measures will be taken as far as is economical.

The manager in charge of energy conservation should plan to reevaluate all his existing evaporation systems in line with this philosophy.

*A report on one completed pilot study is given in H. H. Sephton, *Renovation of Industrial Inorganic Wastewater by Evaporation with Interface Enhancement*, EPA-600/2-76-017, University of California, Sea Water Conversion Laboratory, March 1976.

GLOSSARY

A	Heat-transfer surface area	psig	Pounds per square inch gauge (14.7 psia)
(A)	Optional measurement point	Q	Heat-transfer rate (or quantity of heat supplied to or lost from the process)
B	Bleed rate	Q _L	Heat-transfer rate of the liquor
BPR	Boiling-point rise	Q _V	Heat-transfer rate of the vapor
C	Condensate rate	S	Steam rate
cp	Specific heat	T	Temperature
F	Feed rate	(T)	Temperature measurement point
h, H _L	Enthalpy of liquid	ΔT	Temperature difference
h _C	Enthalpy of condensate	T _A	Approach temperature
h _F	Enthalpy of feed	T _F	Feed temperature
H, H _V	Enthalpy of vapor	T _L	Liquor temperature
H _{Pr}	Enthalpy of product	T _S	Steam temperature
H _S	Enthalpy of steam	T _V	Vapor temperature
in. Hg abs	Inches of mercury absolute	T _W	Cooling water temperature
L	Liquor or liquid rate	U	Heat-transfer coefficient
L _{in}	Liquor rate into process	V, V'	Vapor rate
L _{out}	Liquor rate out of process	W	Compression work per pound of steam
LMTD	Log mean temperature difference	W _V	Compression work, Btu/hr
M	Motive steam rate	X	Concentration
P	Pressure	X _F	Feed concentration
(P)	Pressure measurement point	X _{Pr}	Product concentration
P _S	Steam pressure	λ	Latent heat of vaporization
P _V	Vapor pressure		
Pr	Product rate		

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