Reprinted from CHEMICAL ENGINEERING PROGRESS, October 1962

M.J. BEAGLE
Swenson Technology, Inc.
Monee, Illinois

Evaporation 3:

Recompression evaporation

Addition of a thermocompressor will give improved steam economy equivalent to one effect at considerably lower cost.

THE DESIGN OF AN EVAPORATOR requires an economic evaluation of capital cost versus operating cost to arrive at the optimum number of effects. This is done by comparing the cost of each additional body with the resultant steam and water savings over a axed payout period. In addition to the conventional multiple-effect versus single-effect comparison, the designer should also consider the possibility of recompression evaporation as a means of giving the lowest over-all cost, taking both first cost arid operating cost into consideration. The purpose of this article is to discuss the advantages, disadvantages, and limitations of mechanical recompression and steam jet thermocompression.

Recompression evaporation could be defined as an evaporation process in which a part, or all, of the evaporated vapor is compressed by means of a suitable compressor to a higher pressure level and then condensed; the compressed vapor making up a large percentage of the heat required for evaporation.

This article is limited to consideration of the recompression of water vapor only, although other evaporated solvents could be compressed by similar means.

Mechanical recompression

A mechanical recompression evaporator, similar to Figure 1, is generally limited to a single effect, compressing vapor by means of a positive displacement or centrifugal compressor, which can be driven by either electric motor, steam turbine, or gas or Diesel engine. All of the vapor is compressed and returned to the heat exchanger, with no vapor going to a condenser. This eliminates the cooling water requirement normally associated with conventional or steam jet thermocompression evaporators and would be an important advantage where cooling water is costly. Mechanical recompression is ideally suited for locations where power is cheap and fuel is expensive, such as Switzerland, since power makes up essentially all of the operating~ cost for a motor driven compressor.

Advantages

The greatest advantage of mechanical recompression is the high economy. For example, to isentropically compress saturated water vapor from atmospheric pressure to 15 lb/sq. in. gauge requires the addition of only 55 Btu/lb. For a typical compressor efficiency of *660* the actual Btu addition would be approximately 83 BUn/lb. Neglecting radiation losses and sensible heat requirements to preheat the incoming feed, this would result in an economy greater than twelve for an installation cost comparable to a double effect. Even after taking into account radiation losses and preheating feed 1000 F., economies of better than five would be achieved.

In addition to high economy and water savings, there are several other factors, which would tend to favor mechanical recompression. If only high-pressure steam were available, and if there is a demand for exhaust steam, a turbine driven compressor would be indicated. Relatively cheap diesel fuel would favor a diesel driven compressor. Space limitations might justify a recompression evaporator.

With such high economy possible at a considerably lower price than a conventional multiple-effect evaporator having a comparable economy, why aren't all evaporators of the mechanic-cal recompression type? Power costs must be compared with steam costs. The over-all plant power-steam balance must be considered. If this analysis favors mechanical re-compression, consideration must still be given to the limitations of available compressors, limitations of the material being concentrated, and limitations of the evaporator design.

Design considerations

Compressors are not available in a side range of materials of construction. Standard cast iron, steel, and bronze construction are not suitable for many potential applications where corrosion would result in a short ser ice 1 e. Alloy compressors are not readily available, and are quite costly due to their special nature. The evaporator can be designed to minimize carry over of liquor and/or suspended solids but it would be impossible to eliminate carry over completely. Therefore, mechanical recompression applications must generally be limited to those cases where standard materials of construction can be used for the compressor. Alternatively, vapors with corrosive entrained particles can be neutralized in a scrubber between the vapor body and the compressor. Considerable work has been carried out applying corrosion resistant coatings to compressors of standard materials of construction.

Single stage positive displacement compressors appear to be better suited to recompression evaporator applications than multistage centrifugal compressors because of lower cost and their characteristic fixed capacity, dependent only on speed and relatively independent of inlet or discharge pressures. Single-stage positive displacement compressors are generally limited, however, to a maximum compression ratio of about 2:1, although there are now units available capable of 4-5:1 compression ratios. This limits the maximum available ΔT with a single compressor at a 2:1 ratio to less than 35°F, depending on the operating pressure and boiling point elevation of the liquid being concentrated. To obtain a higher compression ratio would require either multiple units in series or use of one of the new high compression ratio compressors. Centrifugal compressors would probably be indicated for high capacities, above 20,000 cu. ft./min.

Operation of compressor

To keep the size of the compressor within its most economical range, it is customary to operate the evaporator at atmospheric or higher pressure. The specific volume of water vapor increases rapidly below atmospheric pressure, this results in a proportionate increase in the compressor capacity. Pressure operation would rule out many heat sensitive materials and would eliminate those crystallizer applications where lower temperatures are required to obtain a desired hydrate. Even with vacuum on the suction, it is desirable to operate the heat exchanger under positive pressure to permit venting non-condensable to the atmosphere.

A liquor having a high boiling point elevation will not be as adaptable to mechanical recompression, due to the reduced ΔT available for heat transfer, which in turn requires more heating surface. For example, when boiling water at atmospheric pressure and compressing the vapor formed to 15 lb./sq. in. gauge, the available ΔT would be 249 - 212 or 37°F. When boiling a saturated NaCl solution having a 16°F boiling point elevation over the same compression range, the available ΔT for the heating element would be 249 - or 21°F.

Power requirements can be hi$_0$h. Compressing 10,000-lb./hr. of water vapor from -atmospheric pressure to 15 lb./sq. in. gauge would require approximately 330 brake horsepower. It is desirable to drive the compressor with a turbine if there is a use for the turbine exhaust steam, since this provides a means of speed variation s well as being less costly to operate. Within limits, increasing the heating surface will reduce the horsepower required. An economic balance between power cost and heating surface cost is required to arrive at the optimum design.

A conventional single-effect evaporator would have a steel shell for the heat exchanger, and the steam condensate would be returned to the boiler. With a mechanical recompresssion single-effect evaporator the condensate will be contaminated with entrained particles, which could make the condensate unacceptable for boiler feedwater without further treatment. The shell of the heat exchanger would have to be fabricated of a material resistant to the vapor and entrained

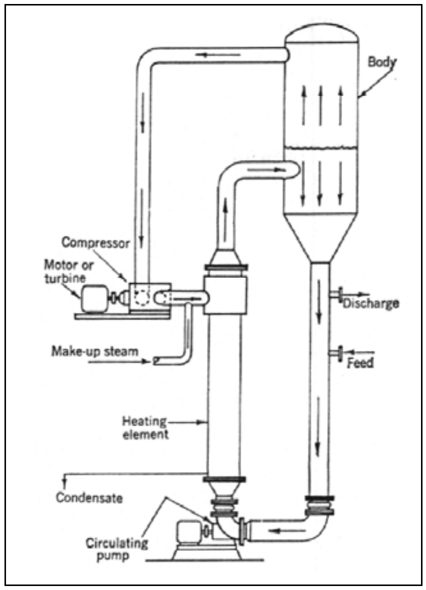

Figure 1. A mechanical recompression evaporator.

particles. Maintenance of the compressor and its driver represent an additional expense not associated with a conventional steam-heated evaporator, and this item could be significant, especially where corrosion is a factor.

In addition to high economy, a mechanical recompression evaporator has the advantage of being very flexible and stable over a wide range of operating rates, assuming that variation in the operating temperature is permissible. A positive displacement compressor at a fixed operating speed delivers a constant volumetric flow. Therefore, changes in the evaporation rate result in a compensating temperature fluctuation due to changes in the specific volume of the vapor. As the rate is decreased the temperature will automatically drop to maintain a con-

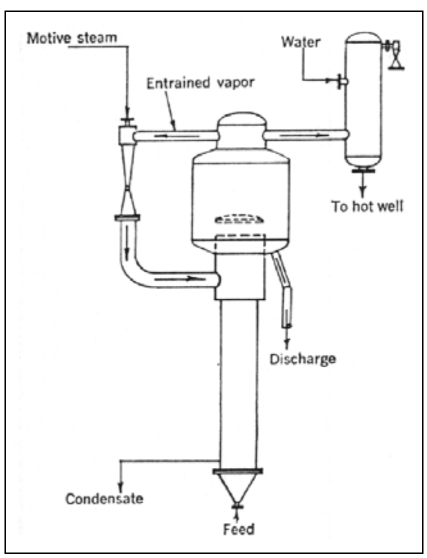

Figure 2. Evaporation with a steam jet thermocompressor.

stant volume of vapor, and as the rate increases the temperature will automatically rise. Therefore, the evaporation rate can be controlled by regulating the make-up steam flow (assuming that the feed temperature is below the operating temperature) to maintain a constant liquor temperature. Fixing the temperature and positive displacement compressor speed fixes the evaporation rate. For example, at a fixed compressor capacity of 10,000 cu. ft./min. the evaporation rate at various saturated vapor temperatures would be:

11,900 lb./hr. @ 180°F
22,300 lb./hr. ® 212°F
36,700 lb./hr. @ 240°F

A mechanical recompression evaporator is particularly well suited for those applications which can be designed for low ~*T*, atmospheric or higher pressure, and for neutral or alkaline liquors having a To *v* boiling paint elevation. Examples are sodium chloride and sodium sulphate evaporator-crystallizers and evaporators for distillation of sea water and concentration of radioactive wastes (1).

Steam jet thermocompression

A steam jet thermocompressor can be used with a single-effect evaporator, Figure 2, or a multiple-effect evaporator. For a multiple-effect installation the most typical arrangement would be compressing vapor across the first effect although it is possible to compress over two or more effects. Dodge *(2)* outlines the basic theory of steam jet compression, and Freneau *(3)* gives an excellent description on the effect of varying steam pressure, suction pressure, and/ or discharge pressure on the performance characteristics of a thermocompressor.

As a rough rule-of-thumb, the addition of a thermocompressor will give an improved steam economy equivalent to the addition of one addition~ 1 effect, and at considerably lower cost. This is based on a 1: 1 entrained vapor: motive steam ratio which is typical hen compressing over a compression range equivalent to approximately 25°F with 100 lb./sq. in. gauge motive steam. In addition to giving an improved economy at low capital cost, the thermocompressor also has the advantages of being available in a wide range of standard and corrosion resistant materials, and being suitable for a wide range of design operating conditions from high vacuum to high-pressure operation. Thermocompressors should be considered when only high-pressure steam is available and the evaporator can be op rated with low-pressure steam. Space limitations

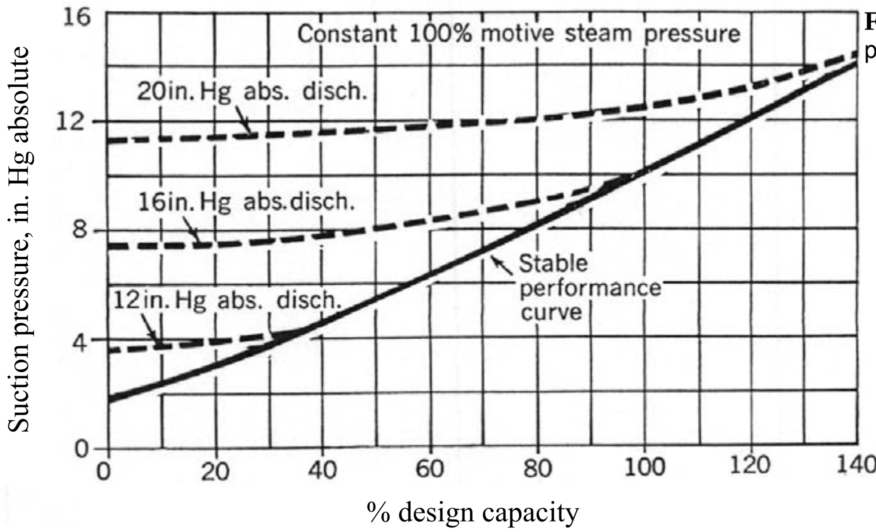

Figure 3. Typical thermocompressor performance curve.

would favor a thermocompressor installation.

As is the case with the mechanical recompression evaporator, the steam condensate will be contaminated with entrained particles and will probably not be acceptable for return to the boiler unless treated. The first effect heat exchanger shell would, therefore, have to be fabricated of a material resistant to the contaminated vapor.

One important limitation of steam jet thermocompressors is that relatively high steam pressures, say over 75 lb./ sq. in. gauge typically, are required. Most multiple-effect evaporators can operate with low-pressure exhaust steam.

Performance

It is relatively easy to design an evaporator with steam jet thermocompressor for a given set of operating conditions. However, once the thermocompressor has been designed and fabricated for this design condition its performance characteristics are fixed, and it is necessary for the evaporator designer to carefully consider the effects of varying the thermocompressor suction (vapor head) pressure, discharge (heat exchanger) pressure, and or motive steam pressure. The problem is simplified with a single-effect evaporator where an absolute pressure controller can be used to maintain a constant suction pressure. Even in this simplified case the discharge pressure will vary as the motive steam rate is varied to change capacity, or as the heat transfer coefficient changes due to change of concentration or fouling of the heating surface. As the discharge (heat exchanger) pressure varies, the capacity of the thermocompressor may or may not decrease depending on the thermocompressor's critical backpressure for the operating motive steam pressure.

Consider a typical thermocompressor performance curve, Figure 3 (4). The solid line represents stable picked-up operation and is the maximum capacity of the thermocompressor at various suction pressures, which cannot be increased by modifying the discharge pressure or motive steam pressure. The capacity can fall off, however, if the backpressure exceeds the critical backpressure for a given motive steam pressure, as shown by the dotted curves. The critical backpressure is the discharge pressure at which the thermocompressor performance begins to break or fall away from the picked-up curve. Increasing the motive steam pressure, hence steam flow, will not alter the stable picked-up curve but will increase the critical backpressure for any given suction pressure. For example, consider a thermocompressor designed for operation at 10 in. Hg absolute suction pressure, discharging at 16 in. Hg absolute, requiring 0.7 lb. motive steam at 100-lb./sq. in. gauge per lb. entrained vapor. At a steady 100 lb. sq. in gauge motive steam pressure and 10 in. Hg absolute suction pressure the capacity would be 100% of design up to the critical backpressure of 16 in. Hg absolute. Above 16 in. the capacity would fall rapidly, dropping to approximately 40-50% of design at 18 in. backpressure, and to 0% at approximately 19 in. Hg absolute backpressure. Raising the motive steam pressure in the hove example would help over come higher backpressure. At 150% of the design motive steam rate, the capacity at a 10 in. suction pressure would be 100% up to approximately 22 in. Hg absolute, falling to approximately 60% at 24 in. backpressure, and 0% capacity at 25 in. Hg absolute backpressure.

For control purposes, it would be desirable to throttle the motive steam flow, but this can introduce problems. Reducing the motive steam flow to 50% in the above example the capacity at 10 in. suction pressure would be 100 up to 12 in. Hg absolute backpressure falling, off rapidly to 0% capacity at about 15 in. Hg absolute backpressure. Actually the system would operate somewhere between 0 and 100% on a broken portion of the performance curve. It becomes virtually impossible for the designer to predict the actual performance due to unknown variations in the heat transfer coefficient and the very steep slope of the broken thermocompressor performance curves. The evaporator designer cannot pinpoint the actual operating conditions nor can the thermocompressor manufacturer predict the broken curves precisely.

The problem becomes still more complex when the thermocompressor is installed on the first effect of multiple-effect unit where the suction pressure is not fixed but varies as the rate and heat transfer coefficients vary. The stable picked-up performance curve is the same for all motive steam flows over a range of 50-150% of design, with entrained capacity approximately proportional to the absolute suction pressure. Then for any given suction pressure the relationship discussed above between motive steam pressure, critical backpressure and capacity applies. Under these conditions accurate prediction of performance of the evaporator at other than design conditions, especially at throttled steam flow, becomes impossible.

Design criteria

Due to the unpredictable performance of thermocompressors in the area of *broken* performance, control is more difficult than for a conventional evaporator where it is necessary to set only steam and feed rates to maintain a constant evaporation rate. With a thermocompressor the quantity of vapor entrained varies and simply establishing a fixed motive steam rate does not necessarily result in a constant evaporation rate. One way to provide rate flexibility with greater operating stability would be to use two or more thermocompressors in parallel.

The evaporator designer therefore, must make every effort to design a thermocompression evaporator so that the thermocompressor can operate on the picked-up portion of its curve over as wide range of operating conditions as possible. Since this picked-up curve represents the maximum capacity of the thermocompressor, at any given suction pressure this will automatically result in the best economy. Thermocompression evaporators have been used for heat sensitive materials whereby all the evaporation can be carried out at low temperature while obtaining economy equivalent to a conventional multiple-effect evaporator. They are also well suited to processes where the operating conditions can be essentially constant with no need for throttling, such as batch wise concentration at a fixed steam rate and suction pressure.

Comparison of evaporators

Several years ago, a comparison was made of a double-effect forced circulation evaporator, a single-effect forced circulation evaporator with turbine dri ‘en mechanical compressor, and a single effect forced circulation evaporator with steam jet thermocompressor, all to be rated at 15,000 lb/hr. evaporation. A summary of this comparison is given in Table 1. For this particular application the feed to the system is Glauber’s salt, $NaSO_4$•10 $H_2$0, which has an extremely high sensible heat load for melting the crystals and heating the resultant slurry to the boiling point. This particular application is in an area which has use for large quantities of exhaust steam, therefore, the evaporator was charged only for the Btu content extracted from the high pressure steam. (This is not typical of most chemical process industries.) On this basis the utilities costs savings definitely justified the slight addition~ 1 first cost for the mechanical recompression evaporator, and this was the unit purchased. The equipment \ ‘as placed in operation in 1958, and operating experience since that time has verified the predicted low operating cost for the evaporator and its relative ease of operation.

A recent installation for concentrating tank water, a meatpacking wet-rendering process by-product, consisted of a double-effect long-tube vertical natural-circulation evaporator with steam jet thermocompressor com-pressing vapors over the first effect. This double-effect with thermo-compressor achieves economy equivalent to a conventional triple effect. Operation is held at an essentially constant feed rate and motive steam rate. There has been little operating difficulty. Another recent evaporator installation with steam jet thermocompression concentrates blood obtaining double effect economy with a single body with thermo-compressor.

Table 1. Cost comparison for three types of evaporator installations.

	Type of Evaporator		
	Double effect	Single effect with mechanical compressor	Single effect with steam jet thermocompressor
Equipment Selling Price (1955)			
Evaporator with motors and instruments	$68,100	$55,400	$58,200
Thermocompressor	......	......	1,800
Compressor, Turbine and Gear	......	15,800*	......
Total	$68,100	$71,200	$60,000
Btu Consumption per hour			
Enthalpy, steam in	12,440,000	13,880,000	9,630,000
Enthalpy, condensate out	2,090,000	763,000	1,264,000
Enthalpy, exhaust steam out	......	8,370,000	......
Btu consumed, net	10,350,000	4,747,000	8,366,000
Utilities Costs per hour			
Steam at 43¢/million Btu	$4.45	$2.04	$3.60
Water at 1¢/thousand gal.	0.15	....	0.15
Power at ½¢/kw. hr.	0.27	0.23	0.23
Total, $/hr.	$4.87	$2.27	$3.98
Total, $/8400 hr. year	$41,000	$19,100	$33,400

**Compressor costs would be much higher today.*

Conclusions

Recompression evaporation merits consideration when evaluating the Optimum number of effects for doing a given evaporation job at minimum cost.

Mechanical recompression is best suited for an application calling for atmospheric or higher operating pressures, mildly corrosive vapor, a low boiling point elevation of the liquor, low ΔT operation for the heat exchanger, and where economy is an important consideration. Adequate power for the motor or high pressure steam for the turbine must be available. This evaporator is flexible over a wide range of operating rates and will give a very high economy per unit capital cost.

Steam jet thermocompression is best suited to single or double effect evaporators where low operating temperatures and improved economy are desired. It costs less to add a thermocompressor than an additional body, both having about the same effect on improving the economy. To obtain reasonable results the ΔT across the compressor should be about 25°F. This type of evaporator is not as flexible as a conventional multiple-effect evaporator due to the variable performance characteristics of the thermocompressor under varying conditions.

M.J. Beagle received his B.S. in Ch. E. from the University of Illinois. He has been with Swenson Technology since 1950. He has spent eight years in the Chemical Equipment design group, and four years in the Houston and Chicago district offices. He is an Associate member of A.I.Ch.E., and of the Chicago Local Section.

LITERATURE CITED

1. Manowitz, B., P. Richards and R. I-Corngan, *Chem. Eng.*, *68*, p. 194 (1955).
2. Dodge, B. F., Chemical Engineering Thermodynamics,” 1 ed., p. 289, McGraw-Hill Book Company, Inc., New York (1944).
3. Freneau, 1’., *Power, 89.* pp, 12-14, 92-95 (1945).
4. Reynolds, *J.* T., private communication to the author May 24. 1957).