Energy Conservation in Edible Oil Processing (US View)

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ABSTRACT

Energy conservation procedures starting with crude edible oil and following through the various process steps are described. Each step illustrates the necessary operating conditions and energy conservation measures now in use or proposed for US edible oil refineries.

INTRODUCTION

Since the early 1970s, the edible oil industry, like all other major industries in the USA, has become acutely aware of the importance of energy conservation to its success and overall profitability. Production steps in this industry consume a large amount of energy. The commonly used oil processing steps of refining, bleaching and deodorizing all consume energy to heat and cool the oil as well as to generate the vacuum levels that are required.

ENERGY FLOW FOR EDIBLE OIL PROCESSING

Each different edible oil requires different process schemes and operating temperature levels to convert crude oil to a finished product. For simplicity, the frequently used steps of caustic refining, bleaching, and deodorizing will be considered here. Of course, any step where the oil passes through two or more temperature levels is a candidate for the steam saving techniques discussed (D.C. Tandy, private communication, 1982).

To determine which of the three steps offers the greatest opportunity for steam savings, it is necessary to review each one to determine the temperature levels required. Table I summarizes these temperature levels. The temperature levels in bleaching and deodorizing are subject to wide variations depending on the type and quality of oil being processed. Also, the storage temperatures vary considerably, depending on the type of oil, location, time of storage, the need for heating, etc. (1). However, those ranges shown are typical for a wide variety of oils. Table I also treats each step independently with regard to storage temperatures.

TABLE I

Typical Temperature Profile for Edible Oil Processing

REFINING	BLEACHING	DEODORIZING
STORAGE (100°- 150°F)	STORAGE (100°- 150°F.)	STORAGE (100°- 150°F.)
PREHEATING (180° F.)	SLURRY PRODUCTION (100°- 150° F.)	PREHEATING (300°F.)
REFINING (180° F.)	HEAT BLEACHING (230°-300°F.)	FINAL HEATING (450°-550° F.)
WATER WASHING (180°F.)	COOLING (150°- 200° F.)	COOLING (150°-200°F.)
DRYING (180°F.)	FILTRATION (100°-200°F.)	STORAGE (150°F.)
STORAGE (180°F.)	STORAGE (150°F.)	

For example, if the oil went from refining to bleaching at the same plant location, storage tanks would be insulated to limit the amount of heat loss so the bleaching system feed temperature would be close to 82 C (180 F). However, if the tanks were not insulated and storage was for a long period, or the oil was transported from one location to another between operations, the temperatures shown might apply.

In caustic refining, the oil comes from storage and is usually preheated to 82 C (180 F) before refining. After mixing with the caustic, the total stream is fed to a centrifuge where the oil and soapstock are separated. The refined oil is then mixed with water and fed to a second centrifuge where the water, containing traces of soapstock, and the oil are separated. The refined, water-washed oil is then dried before being pumped to storage.

Depending on the type of oil and length of storage, the refined oil comes into the bleaching system at 38-66 C (100-150 F) and after addition of the bleaching clay is heated to a bleaching temperature of 110-149 C (230-300 F). The lower temperature range is used for oils that require little or no bleaching such as soybean, while temperatures up to 149 C (300 F) are used for poorer quality oils and where additional reactions other than color transformation must occur in the bleacher, as in the treatment of high free fatty acid (FFA) oils where conventional caustic refining is replaced by an acid pretreat. For the higher bleaching temperatures, the oil must then be cooled prior to filtration and storage.

In the deodorizing system the oil is usually heated to the deodorization temperature of 232-288 C (450-550 F) in two steps, one using steam to reach ca. 149 C and the other using a heat transfer fluid or high pressure steam to reach the final temperature. In some designs the two heating steps are combined and only the heat transfer fluid is used. After the oil is fully deodorized, it is cooled to 66 C (150 F) before being pumped to storage (2-5).

For each of the steps of refining, bleaching, and deodorizing, the oil starts at 38-66 C (100-150 F), is heated to a higher temperature level, and then returns to storage at ca. 46 C (150 F). In the case of caustic refining, this higher temperature level is 82 C (180 F); for bleaching it ranges from 100 to 149 C (230-300 F); and for deodorization, the upper temperature level is 232-288 C (450-550 F). Therefore, it is the heat, in the form of steam or fuel, required to reach this maximum temperature level that can be recovered.

Although many possible uses for this heat could be listed, the obvious one would be to preheat the cold oil coming from storage. However, in the case of caustic refining, using the refined oil at 82 C (180 F) to preheat the crude oil would be counterproductive in a plant where the oil was also bleached, since the cooled refined oil would have to be reheated before bleaching.

Also, it is obvious that, based on the maximum temperature levels shown, bleaching and deodorization offer greater potential for steam saving than caustic refining, and, therefore this paper will discuss specific modifications for heat recovery only with these two processing steps (D.C. Tandy, private communication, 1982).

Energy Conservation – Bleaching

A typical continuous bleaching system is shown in Figure 1. The oil is pumped from storage to the slurry tank where it is mixed with the bleaching clay; this slurry mixture is then pumped into the bleacher which contains coils for steam heating and provides sufficient time to allow the bleaching reactions to take place. An agitator keeps a uniform suspension of clay and oil and also aids in increasing heat transfer efficiency. Also, for most oils it is desirable to remove and keep air out when they are heated to prevent degradation in quality. Therefore, the bleacher is held under vacuum and the oil-slurry mixture is sprayed into the vessel to provide this deaeration step. An alternate approach would be, partially or completely, to heat the oil external to the bleacher and just use the bleacher to provide the retention time and additional heating surface required for bleaching. However, in this case a separate deaerating step would be needed on the crude oil-slurry mixture so all the air is removed prior to the heating. The oil from the bleacher is then pumped through a cooler before filtration. In the filters, the clay is filtered from the oil before the oil is sent to storage. Usually, two filters are used to maintain a continuous operation with one filter on stream while the other is being cleaned. In many operations, precoating of the filter surface is also required.

This bleaching system, modified for heat recovery, is shown in Figure 2. The oil from the bleacher is used to partially heat the oil-clay slurry before entering the bleacher; the coils in the bleacher complete the heating. A smaller oil cooler may be required before filtration. The deaeration step is added external to the bleacher in the form of a separate vessel connected to the bleacher vacuum system.

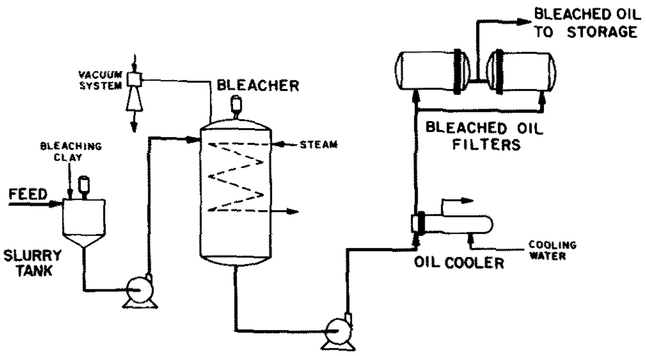

FIG. 1. Bleaching of edible oil.

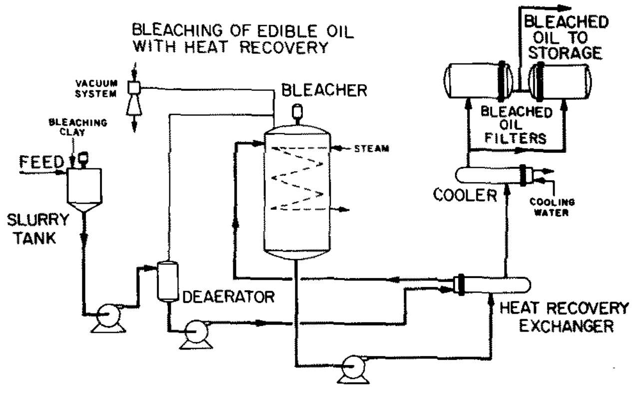

FIG. 2. Bleaching of edible oil with heat recovery.

A 325 MT (30,000 PPHr) bleaching system with heat recovery will conserve 16 MT of steam. The pay-out for the heat recovery equipment is 1-2 years.

Usually the last major step in the processing of edible oils is deodorization, which is required to produce an odorless, tasteless product with low FFA, light color and stability against oxidation (1). Deodorization is essentially a process of steam stripping, whereby volatile compounds, mainly fatty acids, are stripped from the relatively nonvolatile oil. The operation is done at high temperature and vacuum to increase the volatility of these compounds. Application of high vacuum during the operation also serves to protect the hot oil from atmospheric oxidation, and to prevent undue hydrolysis of the oil by reducing the quantity of stripping steam required. This combination of the high temperature and low vacuum makes deodorization a major consumer of plant energy (D.C. Tandy, private communication, 1982).

There are two methods in which this high energy consumption can be reduced; first, by reducing the energy required to heat the oil and, second, by closely controlling the operation of the vacuum equipment to minimize steam consumption (D. Shaw, private communication, 1982).

A complete deodorizing system is shown in Figure 3.

Vacuum system. Steam jet ejectors with barometric condensers are most commonly used to maintain the vacuum in the deodorizer, typically 1-6 mm Hg absolute. The number of ejectors, or stages, depends on the oil throughput and vacuum required as well as the steam pressure and condensing water temperature.

Distillate recovery system. The volatile material which is steam stripped in the deodorizer is commonly called distillate. This recovery system consists mainly of a direct contact distillate recovery tower where cooled distillate is used to condense the fatty acid at conditions that prevent any steam from condensing.

Thermal heating system. Dowtherm vapor heating is the most common method used to heat the oil to the final temperature. However, liquid heating, and high pressure steam systems are also in use.

Antioxidant addition system. Metal ion inactivating agents are added to the feed and/or finished oil to provide for additional stability during and after processing.

Feed/polish filters. Small micron filters are used to remove any solid impurities that may collect in the oil prior to or during deodorization.

Energy Conservation – Oil Heating

Since the oil during deodorizing is first heated to a high temperature and then cooled before going to storage, the first and perhaps obvious method of energy conservation is to heat the incoming oil with the hot, deodorized product before the hot oil is cooled in the cooling tank. The double-shell deodorizer modified for heat recovery is shown in Figure 4. The incoming oil is pumped through the coils in the heat recovery tank where it is heated by a direct oil to oil heat exchange before going to the top of the deodorizer for additional heating by steam and Dowtherm. The entire heating operation is done inside the deodorizer, under vacuum, to eliminate the possibility of air contacting the hot oil. For a feedstock at 49 C (120 F), and a required deodorizing temperature of 260 C (500 F), the feedstock exiting the heat recovery section will be at ca. 146 C (295 F). For a unit deodorizing 325 MT (30,000 lb of oil/hr), a saving of 32.7 tons (3,000 PPHr) of steam will result. At $5/1000 lb steam, this saves $10,268/month of 28 days.

The recovery of heat from the hot, deodorized oil can be taken one step further, as shown on the schematic represen-

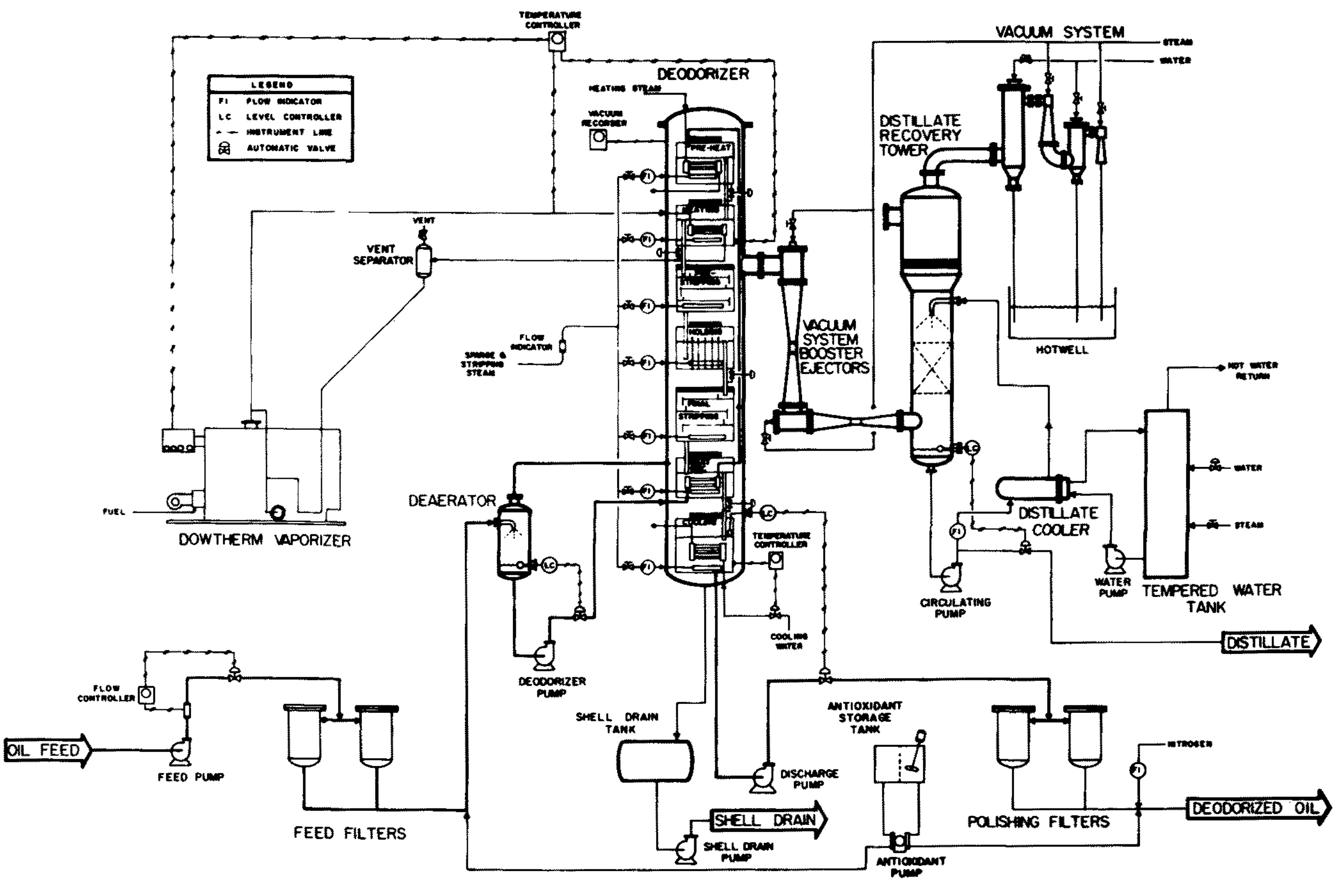

FIG. 3. Complete deodorizing system.

tation (Fig. 5) of a recent installation.

Rather than cooling the oil only with cooling water and rejecting all this heat to the sewer or to a cooling tower, two separate sets of heat exchange coils were installed in the cooling tank. For a deodorizer with a capacity of 325 MT (30,000 lb/hr), ca. 108.8 MT (10,000 lb/hr) of steam is required for steam stripping and in the vacuum system. This steam cannot be recovered for reuse in the boiler, but must be condensed and discarded in the process (D. Shaw, private communication, 1982).

One coil in the cooling tank was used for cooling water as in a regular deodorizer, while the second coil was used to preheat make-up water required to produce this steam. Table II shows the additional savings obtained by preheating the make-up water amounts to more than $4,000/month. Added together, the feedstock and make-up water preheating recover ca. 65% of the heat contained in the hot deodorized oil, with an estimated savings of almost $15,000/month.

Energy Conservation — Vacuum System

The steam jet ejector system is the most common method used to provide the high vacuum required in oil deodorization. The steam jet ejector depicted in Figure 6, is relatively simple in construction, consisting of three basic components: a nozzle, mixing chamber, and diffuser. A pressurized motivating fluid, steam, is directed through a converging-diverging nozzle to produce a high-velocity stream. In the mixing chamber, suction fluid is entrained in the steam and carried into a venturi-shaped diffuser which converts the velocity energy of the steam into pressure energy. The ejector is a simplified type of pump increasing the pressure of the suction fluid. As with any pump, the greater the compression the more energy, in this case steam, is required. This drawing shows a single nozzle for the steam; in larger ejectors, multiple nozzles can be used. Since the steam pressure in commercial installations is usually in the range of 5.4-13.6 atmosphere (80-200 psig), the nozzles are in effect critical orifices and the flow through the nozzle is directly proportional to the inlet pressure. Therefore, performance of ejectors is usually specified as the inlet steam pressure rather than the flow, since pressure is easier to measure. Steam jet ejector systems are usually custom-designed to meet each process application. These units offer a reliable, relatively quiet means for generating vacuum with a minimum of maintenance. They do, however, consume a substantial amount of high pressure steam in their operation.

A typical 4-stage vacuum system used in edible oil deodorizers to provide a pressure of 3 mm Hg absolute is shown in Figure 7. The distillate recovery tower is omitted for clarity. The first two jets, commonly referred to as boosters, are connected in series and discharge into a direct contact barometric condenser in which the suction condensibles and booster steam are condensed. A two-stage ejector with intercondenser follows to compress the remaining noncondensibles up to atmospheric pressure. Steam consumption in the second stage is ca. 50-75% of the total of the four jets, since this stage must handle not only the suction load from the deodorizer, but also the steam from the first stage. In this system the discharge from the first-stage booster cannot be condensed because the pressure at the discharge

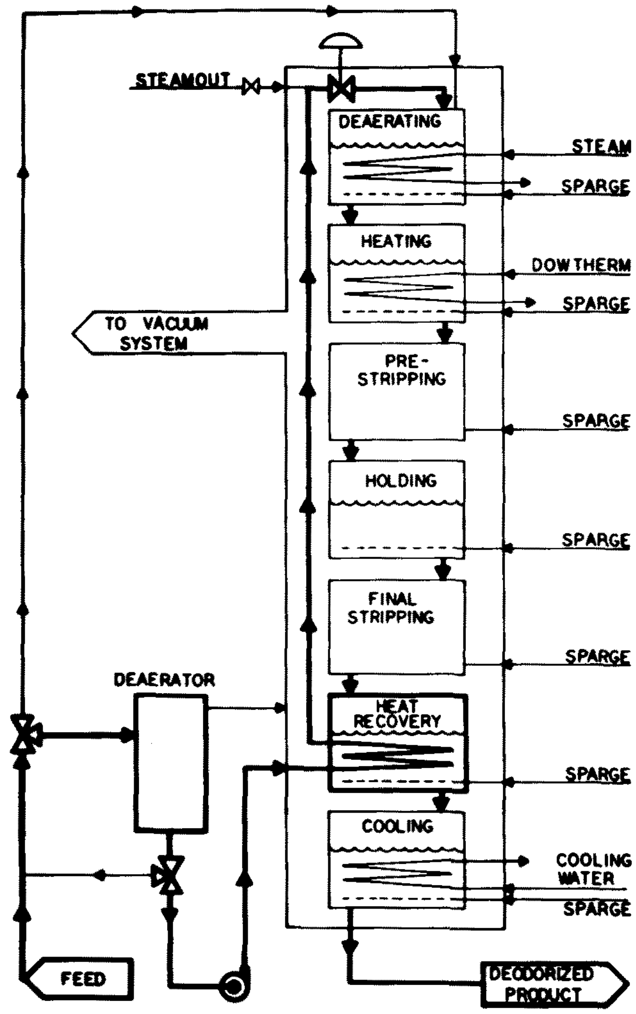

FIG. 4. Double-shell deodorizer modified for heat recovery.

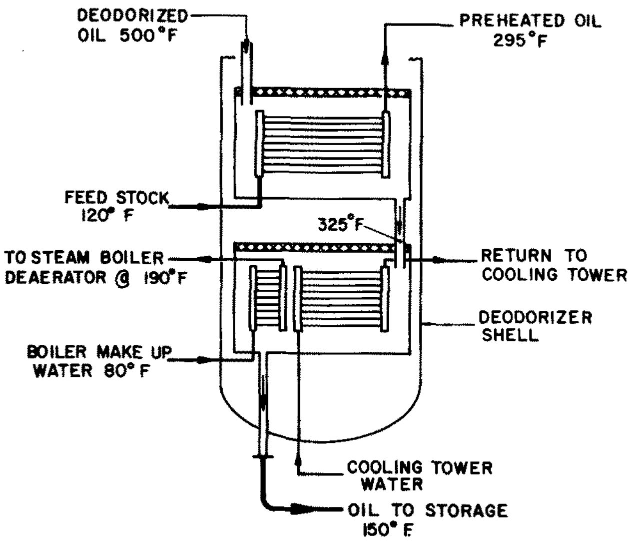

FIG. 5. Heat recovery.

TABLE II

Additional Savings Obtained by Preheating the Make-Up Water

Flow Rate Boiler Feedwater	*20 GPM*
Temperature	*80° F*
Preheated Temperature	*190° F*
Heat Reclaimed	*1,100,000 BTU/hr.*
Equivalent Steam at 150 psig for Preheating	*1,285 #/hr.*
Monthly Savings (Steam at $5.00/1000 Lbs.) 28 Operating Days per Month	*$ 4,318*

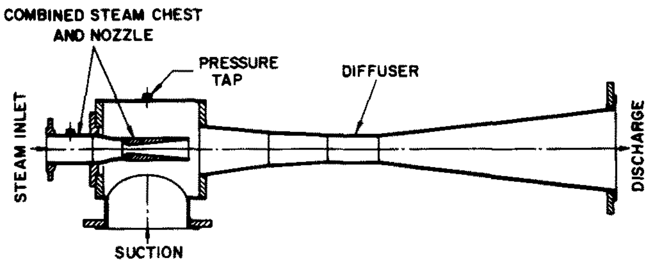

FIG. 6. Steam jet ejector system.

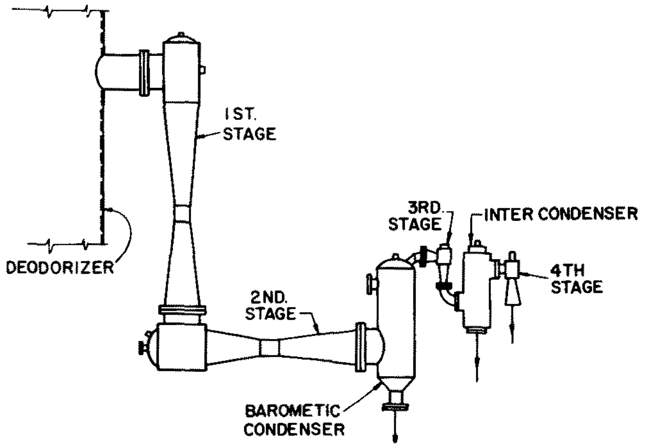

FIG. 7. Four-stage vacuum system.

is much too low to permit condensation with water at normal temperatures encountered in commercial installations.

The compression ratio of the two-stage booster system is determined by the pressure in the barometric condenser, since the suction pressure in the deodorizer must remain constant at the design value. The discharge of the barometric condenser consists of the noncondensibles saturated with water vapor, and thus this condenser pressure is, in effect, the vapor pressure of the water vapor at the condenser discharge temperature. Thus, the compression ratio is directly related to the temperature of the cooling water used. This effect can clearly be shown as follows, along with the data in Table III. If sufficient water at 32.2 C (90 F) is supplied to condense all the condensibles with only a 6.7 (12) degree rise in the water temperature, the water out of the condenser will be at 38.9 C (102 F). If equilibrium is reached, the noncondensibles leaving the condenser would be saturated with water vapor also at 38.9 C (102 F), but in most designs a slightly higher temperature, say 40.6 C (105 F), is used, to allow for the fact that true equilibrium may not be reached. The equilibrium pressure or vapor pressure of water at 40.6 C (105 F) is 57 mm Hg absolute, and therefore the combined compression ratio of the two booster systems is 19.0. If the water temperature drops to 26.7 C (80 F), the equilibrium temperature drops to 35 C (95 F), with a subsequent drop in equilibrium pressure to 42 mm Hg absolute. The ratio then drops to 14.0 or a 26% reduction. The effect on steam consumption in the second-stage booster is shown graphically in Figure 8. The drop in water temperature to 26.7 C (80 F) reduces the steam required by 33%. A similar curve could be prepared for the first-stage ejector.

When the water for the vacuum system is recirculated through a cooling tower, the important parameter in the selection of a vacuum system besides steam pressure is the average wet bulb temperature in the locale where the deodorizer is to be installed. Cooling towers are designed to cool water within a few degrees of this wet bulb temperature. In a typical design, the wet bulb and corresponding water temperature chosen are those that are not exceeded 1% of the time during the hot summer months to ensure that good quality oil can be produced at all times.

In most climates, however, the temperature will generally vary from season to season and from night to day, and therefore, the temperature of the water will be below the design temperature for a substantial part of the time. This means that if steam jets are operated at the design steam flow at all times, a great deal of steam will be wasted that was not required to produce the vacuum. Steam flow to the boosters can be throttled during times of lower than design water temperature.

For the past several years, one vacuum system supplier has provided a multiple nozzle booster where one of the nozzles located in the center of the steam inlet chest was connected to a separate inlet pipe from the remaining nozzles. During periods when the actual water temperature was considerably below that used in design, the steam flow to this center nozzle could be shut off to achieve a 10-15% reduction in steam usage. Trial and error was used for each installation to determine when this nozzle could be cut off. This system or any other system in which the steam flow is reduced manually at lower water temperatures does result in some steam savings, but the optimum savings can never be realized without constant surveillance, which is impractical. Systems which are totally automatic or which can at least be operated from a central location are preferred.

TABLE III

Compression Ratio of the 2-Stage Booster System

COOLING WATER TEMPERATURE, °F	90	80
WATER TEMPERATURE RISE, °F	12	12
WATER OUTLET TEMPERATURE, °F	102	92
EQUILIBRIUM TEMPERATURE, °F	105	95
EQUILIBRIUM PRESSURE, mm. Hg.	57.0	42.0
COMPRESSION RATIO (SUCTION = 3 mm Hg.)	19.0	14.0

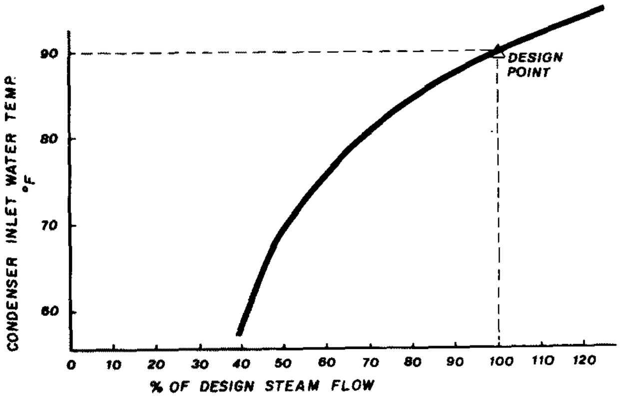

FIG. 8. Approximate steam flow of 2nd-stage booster vs condenser inlet water temperature. 100% flow = 3750 lb/hr steam at 160 psig.

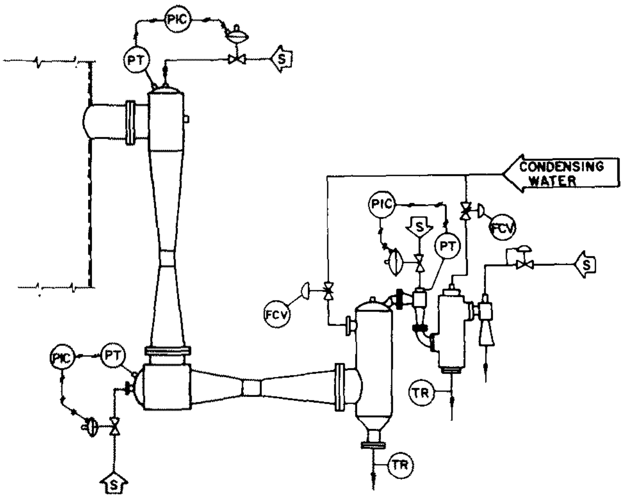

FIG. 9. Vacuum system involving centralized control of individual steam flows.

Figure 9 illustrates a method recently used in a commercial installation to provide greater control, involving the centralized control of the individual steam flows to each jet and the water to the condensers. The first, second, and third-stage jets are equipped with pressure transmitters on their steam chests which send a signal to separate indicating pressure controllers in the control room. Output signals from these controllers adjust the flow of steam to each jet by adjusting the position of a control valve. The fourth-stage steam supply is maintained by a simple self-contained regulator. The barometric condenser tailpipe and cooling tower water temperatures are also monitored. Water flow to both the barometric and intercondenser can be adjusted from the control room. This system allows a single operator at the control panel to monitor closely all variables in the system and adjust the flow of steam and water to minimize consumption. The control is not continuous, however, and thus the maximum possible savings will probably never be realized.

Figure 10 shows a schematic of a second kind of control system that we recently supplied with a 4-stage vacuum system in a Southeastern US installation. Since the second-stage booster consumed 70% of the total steam required, only the flow to this booster was controlled.

The principle component in this system is ratio controller PRC which functions to maintain a line steam pressure in direct proportion to the absolute pressure in the barometric condenser. An absolute pressure transmitter senses the pressure in the condenser and sends a signal to the ratiocontroller. Similarly, a second transmitter senses the steam pressure at the inlet to the booster and sends a signal to the controller. The deodorizer, of course, must remain at a constant 3 mm Hg absolute, the design vacuum.

The set point of the controller is set at the design compression ratio of the booster. As the condenser pressure falls due to cooler water, the steam flow into the booster is reduced by reducing the pressure. During initial operation, the set point is adjusted to provide the minimum flow required to give the proper vacuum in the deodorizer.

The potential savings for this system based on the average climatic conditions for each of four months are shown in Table IV. The design of the cooling tower is assumed to be sufficient to give a water temperature 2.2 C (4 F) above the wet bulb.

In a summer month, like August, the maximum water temperature of 32.2 C (90 F) may be reached frequently during the afternoon hours. The average monthly water temperature however is only 26.7 C (80 F), so savings can be realized by reducing the steam flow on cooler days and during the night-time hours. Of course, substantial savings occur during the winter months, such as February. Expanding this table to cover a full year would show a total annual savings of ca. $64,000. The total installed cost for the controls was ca. $10,000. Also, in this part of the USA, the average wet bulb temperature ranges from 15.6 C (60 F) in February to 24.4 C (76 F) in August, or an 8.8 (16) degree range. In Northern cities, this range may be as high as 17 C (30 F) because of colder winter months. Even greater savings could then be realized in these locations.

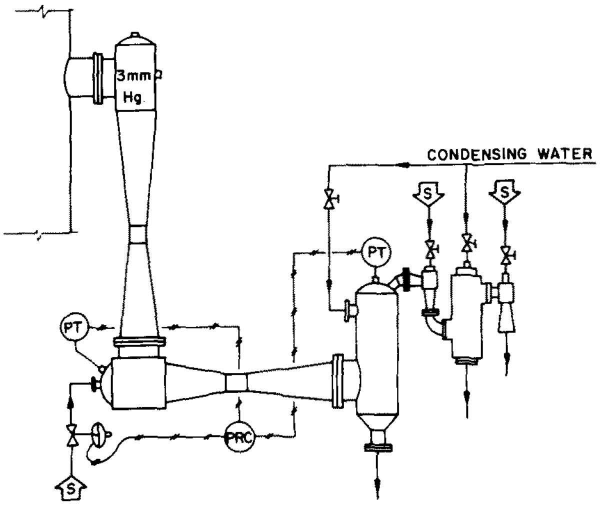

FIG. 10. Control system with 4-stage vacuum. Deodorizer pressure = 3 mm; design water temperature = 90 F; steam consumption (160 psig):

	PPH	%
1st stage	1070	19.1
2nd stage	3750	70.2
3rd stage	305	5.7
4th stage	265	5.0

TABLE IV

Potential Savings for Control System with a 4-Stage Vacuum

MONTH	FEB.	MAY	AUG.	NOV.
AVG. WET BULB, °F	60	73	76	71
CONDENSING WATER TEMPERATURE, °F	64	77	80	75
% OF DESIGN STEAM FLOW	44	61	68	58
STEAM SAVED LBS./HR.	2,100	1,463	1,200	1,575
MONTHLY SAVINGS, $ (STEAM @ $ 5.00/1000 LBS.)	7,056	4,916	4,032	5,292

Deodorizer capacity: 40,000 lb/hr; design water temperature (maximum): 90 F.

EQUIPMENT – DEODORIZER HEAT RECOVERY

When utilizing a heat exchange section located within the deodorizer shell, this basic heat recovery system can economically reduce the heating fuel requirements for the deodorization operation by 45-55%. Even higher heat recovery rates up to 85% can be attained by using specially designed external heat exchangers. But, this method results in the preheating of the feedstock oil to rather high temperatures (218-241 C) (425-465 F) outside of the deodorizer shell and, therefore, is not always an acceptable technique for all processors.

Shell and tube, plate type and spiral heat exchangers are used for external heat recovery.

Capital operating costs, maintenance requirements, weight and space limitations, temperature approach, and pressure and temperature levels play critical roles in the choice between a plate exchanger and tubular exchanger.

A plate exchanger becomes attractive when an expensive material of construction is required. (When mild-steel construction is acceptable, a shell-and-tube exchanger is often more economical). A plate unit need not be insulated, and (for the same duty) can be supported on a less expensive foundation than a shell-and-tube one.

Also, less energy is required for pumping fluids in plate exchangers, which lowers operating cost.

A plate exchanger consists of a frame and corrugated or embossed metal plates. The frame includes a fixed plate, pressure plate, pressing arrangement and connecting ports. Plates are pressed together in the frame. The end-plates do not transfer heat.

Adjoining plates are spaced by gaskets to form a narrow uninterrupted passage through which liquids flow in contact with the corrugated surfaces of the two plates. The corrugation imparts turbulence. Hot and cold streams flow in alternate spaces between the plates. Several arrangements of the fluid streams are possible.

Turbulence is attained in plate exchangers at Reynolds numbers from 10 to 500 (in smooth-pipe laminar flow, Reynolds numbers range up to 2,100) because the corrugation breaks down the stagnant insulating film at the heat-transfer surface. This, combined with the thin plate and lower fouling factor, contributes to the plate exchanger's higher heat-transfer coefficients than those of the shell-and-tube exchanger at the same Reynolds-number flowrate.

The spiral heat exchanger approaches the ideal in heat transfer equipment. Media are in full counterflow. The same flow characteristics are obtained for each medium. The single passage for each side eliminates channeling and diminishes fouling tendencies. The long pass on each side permits close temperature approaches and temperature crosses. Radiation losses are negligible. The spiral exchanger is compact and requires less installation and servicing space than conventional exchangers of equivalent surface. Essentially, the unit is an assembly of two long strips of plate wrapped to form a pair of concentric spiral passages. Alternate edges of the passages are closed so that media flow through continuous leak-proof channels. Covers are fitted to each side of the spiral assembly to complete the unit. Spiral exchangers are fabricated in any metal that can be cold-formed and welded.

Energy Conservation – Hydrogenation

Modern hydrogenation plants use a drop tank located beneath the hydrogenator (Fig. 11). This drop tank is equipped with special design pipe coils for heating the incoming feedstock. Hydrogenation reactions are done at temperatures as high as 220 C (428 F).

When the hydrogenation reaction is completed, the batch of hot oil is drained into the drop tank. Pumping the next batch of feedstock through the pipe coils recovers sufficient heat to start the hydrogenation reaction without the addition of heat from external sources.

The drop tank eliminates the necessity of pumping hot oil.

FUTURE ENERGY SAVINGS – LOW TEMPERATURE

The most obvious high temperature heat recovery situations have been discussed. Additional savings can be obtained in an integrated extraction plant/edible oil refinery. For example, consider a 2000 ton/day extraction plant, 325 MT (30,000 PPHr) oil refinery (N.H. Witte, private communication, 1982). This plant has a soft water make-up of 16,329 kg/hr for the following users: desolventizing toaster, 9,072 kg/hr; deodorizer vacuum system, 375 kg/hr; water wash, 4,000 kg/hr; and boiler blow-down, 5,000 kg/hr. Low temperature and lost heat can be used to heat soft water which can be used for water washing, tank car and tank truck washing, and boiler make-up.

Soft water at 21.1 C (70 F) can be heated to 57.2 C (135 F) by recovering heat from deodorizer high temperature heat stack, and recovery from discharge of air ejectors.

Additional opportunities for energy conservation include operation of more efficient hydrogen gas plants, and use of alternate fuels to create steam.

A major soybean processing plant located at Cameron, South Carolina, faced spiraling oil and propane costs in the 1970s. With the operation running 24 hr/day, 7 day/week, 320 day/year and an average steam flow of ca. 30,000 lb/hr, engineers determined that an energy investigation needed to be conducted. The investigation would pinpoint the opportunities available by conversion of the plant's oil and propane fired watertube boiler to a multifuel system firing coal, oil and propane (R.D. Pincelli, private communication, 1982).

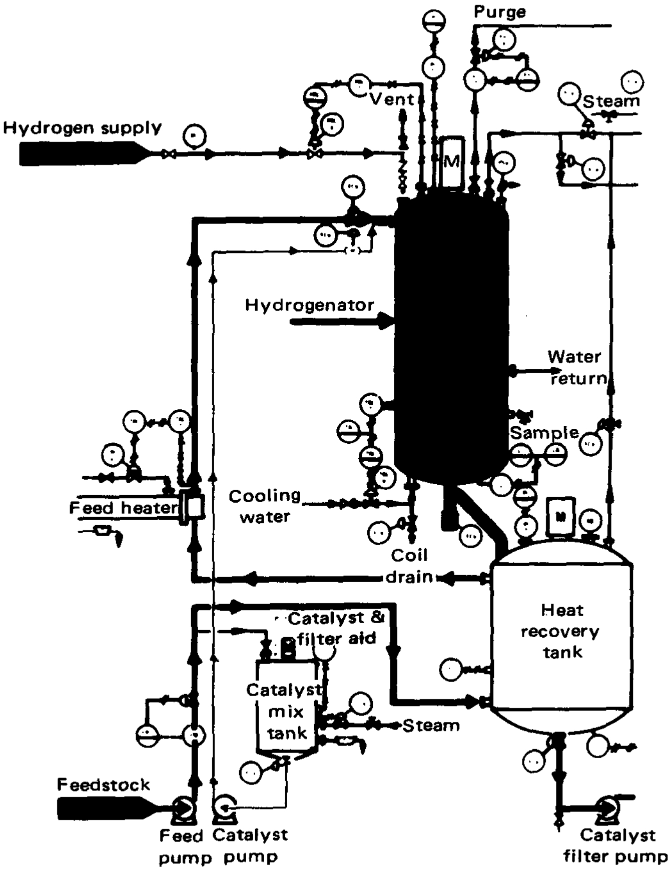

FIG. 11. Hardening plant with heat recovery tank.

The ultimate decision was the outright purchase of a system, with coal being the primary fuel. Coal delivered and unloaded cost $3.10/MM BTU, oil was $6.44/MM BTU and propane was $5.80/MM BTU. All of these figures include the effect of combustion efficiency on the cost per million BTU. Therefore, the cost of coal was 48% of the cost of oil, and 53% of the cost of propane.

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