

UOP ENGINEERING DESIGN SEMINAR

ENERGY SYSTEMS

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ENERGY SYSTEMS

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Elements of a Typical Refinery Steam System

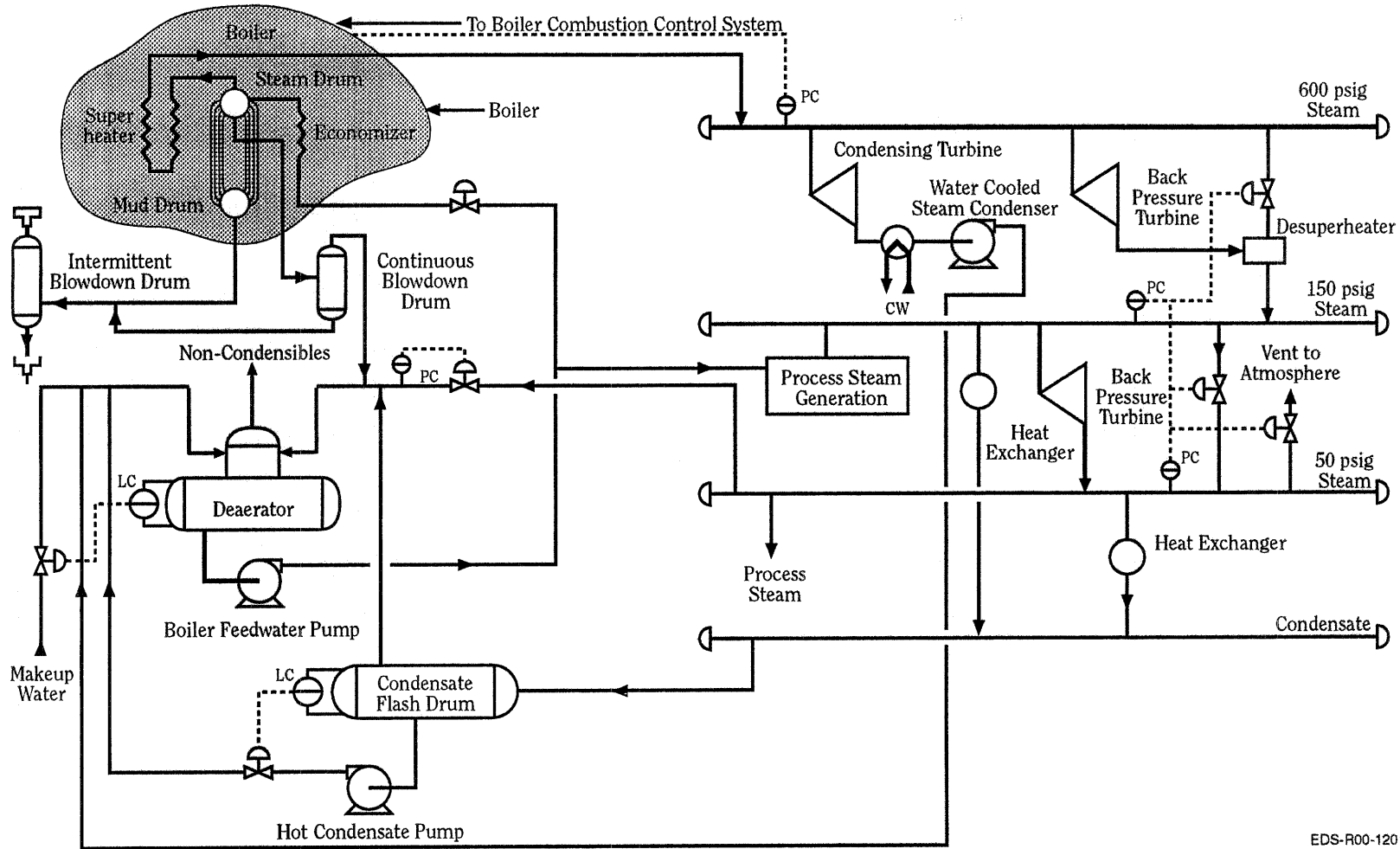


Figure 1

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Turbine Types, by A. A. Luoma, R. Del Vecchio

Let's Talk Turbines

Steam turbines are built in a number of different arrangements to suit the needs of individual installations. Basically, all turbines may be divided into two broad classes: condensing units, which operate at back pressures less than atmospheric; and non-condensing units, with back pressures above atmospheric. This division refers only to exhaust flange pressures, and not to what happens to the steam after it leaves the machine.

Each class may be subdivided according to whether full throttle flow continues through the machine to exhaust, or is withdrawn from the unit after some expansion.

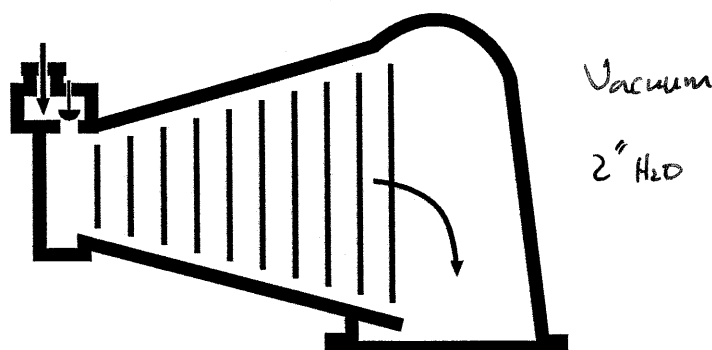
For simplicity, we will cover in these pages the five most commonly used types:

- 1) Straight condensing
- 2) Straight non-condensing
- 3) Extraction
- 4) Induction
- 5) Induction-extraction

Straight Condensing

A straight condensing turbine takes high pressure steam, expands it through turbine nozzles and blading, and exhausts it to a condenser at a pressure less than atmospheric. It is principally used when power must be generated with a minimum steam consumption.

Figure 2



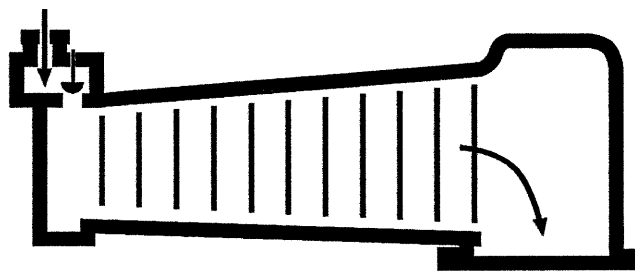
Straight Condensing

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Straight Non-Condensing

A straight non-condensing turbine works the same as the straight condensing type except that steam is exhausted at atmospheric or higher pressures. Its application is common in processes where exhaust steam can be used and when very low cost steam is available.

Figure 3



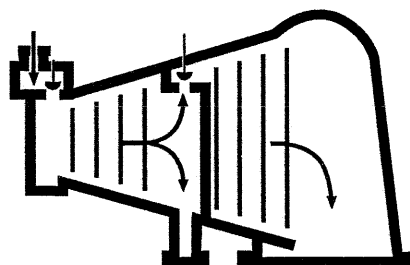
Straight Non-Condensing

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Extraction

An extraction type turbine takes high pressure steam, expands it to an intermediate pressure where part is withdrawn for process and the remainder expanded to exhaust pressure (any pressure from below atmospheric upwards).

Figure 4



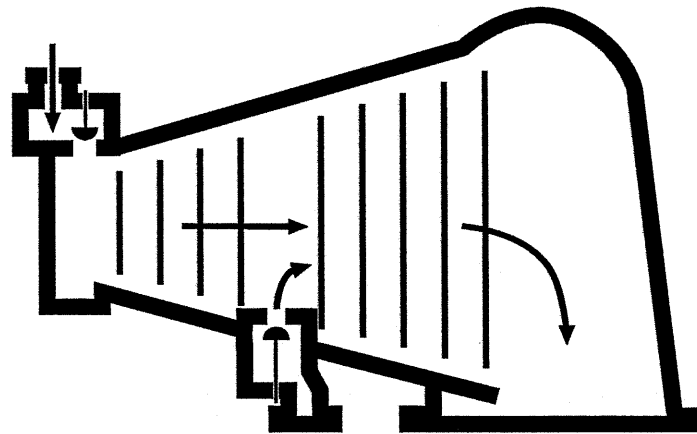
Extraction

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Induction

An induction type turbine is used when sufficient quantities of low pressure process steam are available to furnish a portion of the power requirements. High pressure steam furnishes the remainder. The induction turbine automatically utilizes all low pressure steam available, using only enough high pressure steam as make-up to carry the load.

Figure 5



Induction

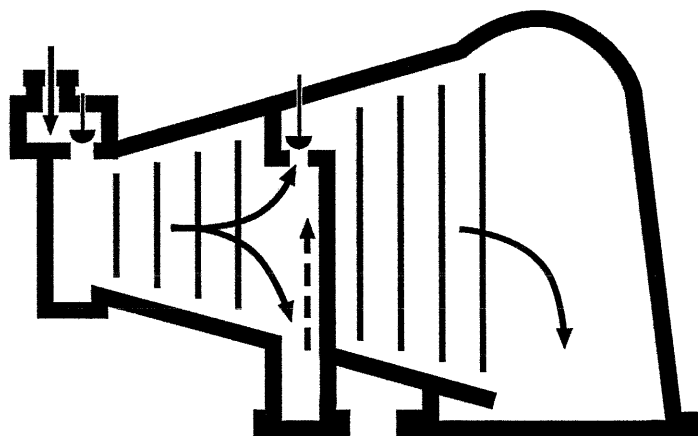
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Induction-Extraction

An induction-extraction turbine is used in connection with a process operating between the turbine inlet and exhaust pressures, either generating too much or too little steam. The turbine automatically inducts or extracts the excess steam to maintain process balance.

Turbines are also available for automatic double extraction where steam is extracted at two intermediate pressures between inlet and exhaust. Pressure at these intermediate point are automatically kept constant despite varying extraction steam demands.

Figure 6



Induction-Extraction

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PUMP HORSEPOWERS

Pump Rated Brake Horse Power (rated bhp)

$$\text{rated bhp} = \text{gpm} \times \Delta P / (1714 \times \text{efficiency})$$

Pump Normal Brake Horsepower (norm bhp)

$$\text{norm bhp} = \text{rated bhp} \times [(\text{norm gpm} + \text{des gpm}) / (2 \times \text{des gpm})]$$

Note: This is a good approximation for estimating purposes

Pump and Single Stage Turbine Example

$$\text{Pump rated gpm} = 433, \text{norm gpm} = 325$$

$$\text{Rated Differential Pressure} = 259.1 \text{ psi}$$

$$\text{Efficiency} = 72\%$$

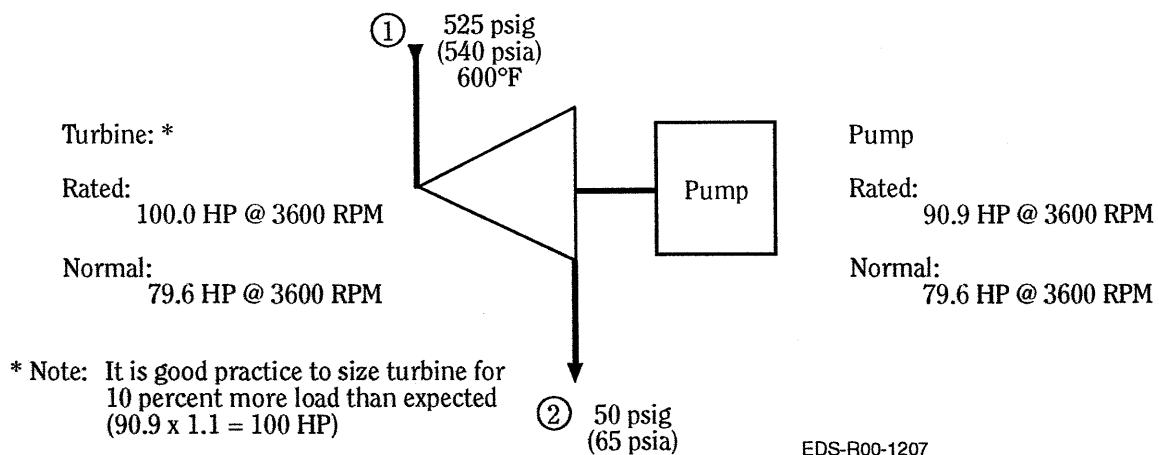
$$\text{rated bhp} = \text{gpm} \times \Delta P / (1714 \times \text{efficiency}) = 433 \times 259.1 / (1714 \times 0.72)$$

$$\text{rated bhp} = 90.9 \text{ hp}$$

$$\text{norm bhp} = \text{rated bhp} (\text{norm gpm} + \text{des gpm}) / (2 \times \text{des gpm})$$

$$\text{norm bhp} = 90.9 \times (325 + 433) / (2 \times 433) = 79.6 \text{ hp}$$

Figure 7



Theoretical Steam Rate:

$$525 \text{ psig} = 540 \text{ psia}$$

$$@ 540 \text{ psia}, 600^\circ \text{F}: h_1 = 1295.7 \text{ s}_1 = 1.5485$$

For Isentropic Expansion(constant s):

$$@ 65 \text{ psia}, s = 1.5485:$$

$$\text{Saturated Liquid: } h=267.63, s = 0.4344$$

$$\text{Saturated Vapor: } h = 1179.1, s = 1.6375$$

$$\text{by interpolation, for an } s = 1.5485 \text{ } h_2 = 1111.67$$

Percent Moisture:

let L be weight fraction liquid

$$h_2 = L \times h_{\text{sat.liquid}} + (1-L) \times h_{\text{sat.vapor}}$$

$$1111.67 = L \times 267.63 + (1-L) \times 1179.1$$

$$L = 0.0740$$

$$\text{weight fraction vapor} = 1 - 0.0740 = 0.926$$

$$\text{Therefore, steam quality} = 92.6 \text{ pct}$$

Theoretical Maximum Work Recoverable

$$h_1 - h_2 = \Delta h$$

$$\Delta h = 1295.7 - 1111.67 = 184.03 \text{ Btu / lb steam}$$

$$1 \text{ kW} = 3413 \text{ Btu / h}$$

$$\text{Therefore } (3413 \text{ Btu/kWh}) / (184.03 \text{ Btu/lb Steam}) = \text{TSR}$$

Theoretical Steam Rate (TSR) = 18.55 lb steam/kW-h

Base Steam Rate (BSR) @3600 r/min

$$AYR = 47 \text{ lb/hph}$$

$$DYR = 28 \text{ lb/hph}$$

Horsepower Loss

$$AYR = 2.2 \text{ hp}$$

$$DYR = 40.0 \text{ hp}$$

Superheat Correction

$$\text{Saturation Temperature @ 525 psig} = 475^\circ \text{F}$$

$$\text{Superheat} = 600 - 475 = 125^\circ \text{F}$$

$$\text{Correction} = 1.04$$

Corrected Steam Rate (CRS)

$$CSR = (BSR / \text{Superheat correction}) [(hp + hp_{Loss}) / hp]$$

$$\text{for AYR} = (47 / 1.04) \times (100 + 2.2) / 100 = 46.2 \text{ lbs/hph}$$

$$\text{for DYR} = (28 / 1.04) \times (100 + 40) / 100 = 37.7 \text{ lbs/hph}$$

Total Steam Consumption

$$AYR = 46.2 \text{ lb/hph} \times 100 \text{ hp} = 4620 \text{ lb/h}$$

$$DYR = 37.7 \text{ lb/hph} \times 100 \text{ hp} = 3770 \text{ lb/h}$$

Part Load Steam Consumption

Assume the AYR turbine will be used.

Find the normal steam consumption.

$$R_1 = (\text{Full Load Steam Rate} \times r/\text{min})/100000$$

$$= (46.2 \text{ lb} / \text{hph} \times 3600)/100000 = 1.66$$

$$R_2 = (\text{norm bhp} / \text{rated bhp of turbine})$$

$$= (79.6/100) = 0.796$$

Fig. 8

From Part Load Chart : C = 0.82

$$\text{Normal Steam Flow} = C \times \text{Rated Turbine Flow}$$

$$= 0.82 \times 4620 \text{ lb/h} = 3788 \text{ lb/h}$$

Using the same method for the DYR :

$$\text{Normal Steam Flow} = 3325 \text{ lb/h}$$

What is the Exhaust Steam Temperature for the AYR Turbine Driving the Pump at Normal GPM?

Normal HP

$$h_1 - h_2 = \Delta h = (79.6 \text{ hp} \times 2545 \text{ Btu} / \text{hph}) / 3788 \text{ lb} / \text{h}$$

$$\Delta h = 53.5 \text{ Btu/lb}$$

$$\Delta h = \frac{79.6 \times 2545}{3325} = 61 \text{ Btu/lb}$$

$$h_1 = 1295.7 (540 \text{ psia}, 600^\circ \text{F})$$

$$h_2 = 1295.7 - 53.5 = 1242.2 \text{ Btu/lb} @ 65 \text{ psia}$$

From Steam Tables:

65 psia: 400°F h=1232.7 Btu/lb

450°F h=1257.9 Btu/lb

By linear interpolation:

For h=1242.2 Btu/lb, T=419°F

SIMPLIFIED STEAM RATE CALCULATION FOR SINGLE STAGE TURBINES

(The steam rate determined from these calculations is a quick approximation, accurate to within ± 5 pct. It is based on Elliott turbines, but is representative of other vendor turbines).

Step 1:

Determine **Theoretical Steam Rate**

Step 2:

Determine **Base Steam Rate** (use curves based upon TSR, Turbine Speed, and Turbine Type).

Step 3:

Determine the **Horsepower Loss** (use curves based upon Turbine Speed, Exhaust Pressure and Turbine Type).

Step 4:

Determine **Superheat** (use table).

Step 5:

Determine the **Superheat Correction Factor** [use curve knowing Superheat ($^{\circ}\text{F}$) and TSR]

Step 6:

Determine **Corrected Steam Rate**
Corrected Steam Rate =

$$\left(\frac{\text{Base Steam Rate}}{\text{Superheat Correction Factor}} \right) \left[\frac{(\text{hp} + \text{hp}_{\text{loss}})}{\text{hp}} \right]$$

EXAMPLE

Steam Conditions

250 psig Inlet
575 $^{\circ}\text{F}$ Inlet
50 psig Exhaust

Design Conditions

Turbine to Develop
500 Horsepower at
4000 r/min

Step 1: TSR = 26.07

Step 2: BSR = 36

Step 3: hp Loss = 55

Step 4: Superheat = 169 $^{\circ}\text{F}$

Step 5: SCF = 1.052

Step 6:

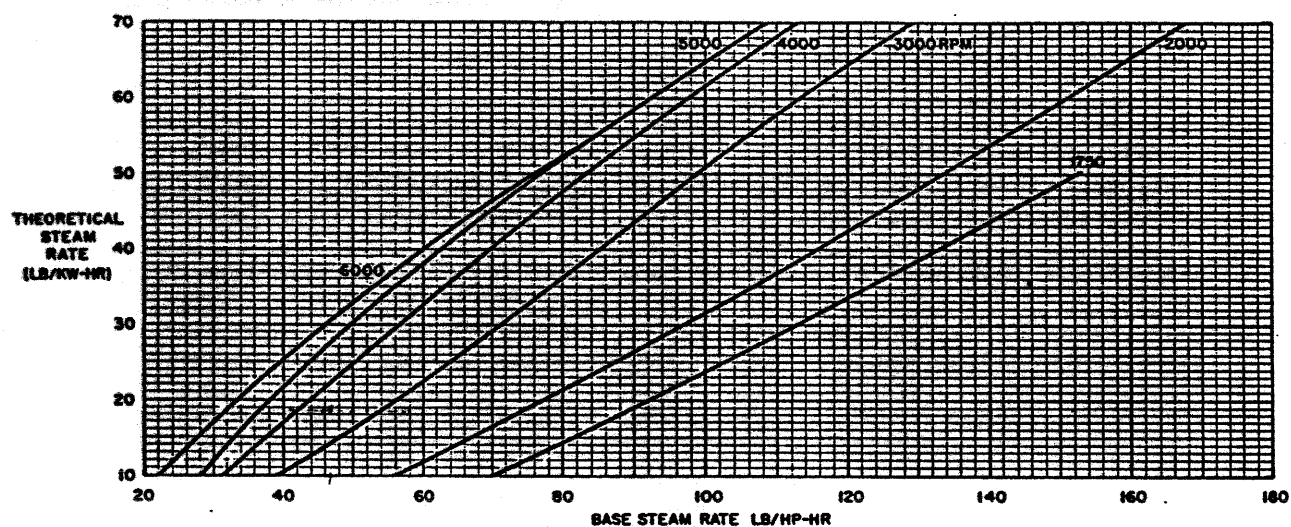
$$\frac{36}{1.052} \times \frac{(500 + 55)}{500} = 38.0 \text{ lb/hph}$$

(Similar calculation for AYR turbine results in Corrected Steam Rate of 49.7 lb/hph)

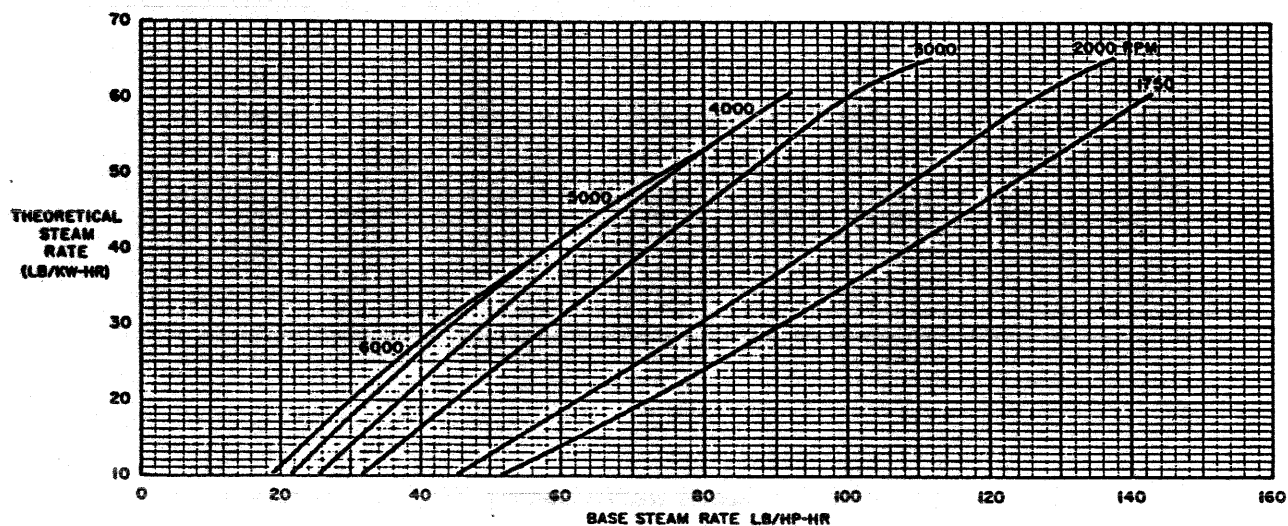
The Simplified Steam Rate Calculations establish that, for most applications, the DYR turbine offers the lowest corrected Steam Rate (see below), the AYR the highest. The DYR is the most expensive turbine. However, consideration of Steam Rate may result in a lower evaluated cost for a larger and more efficient turbine

Base Steam Rates

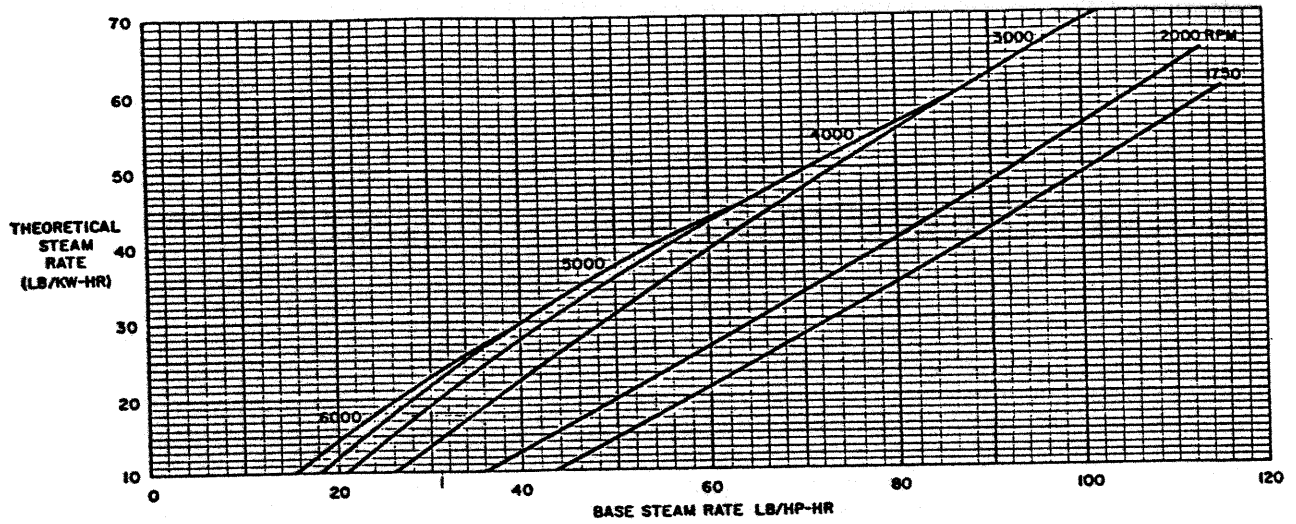
AYR - 14" WHEEL



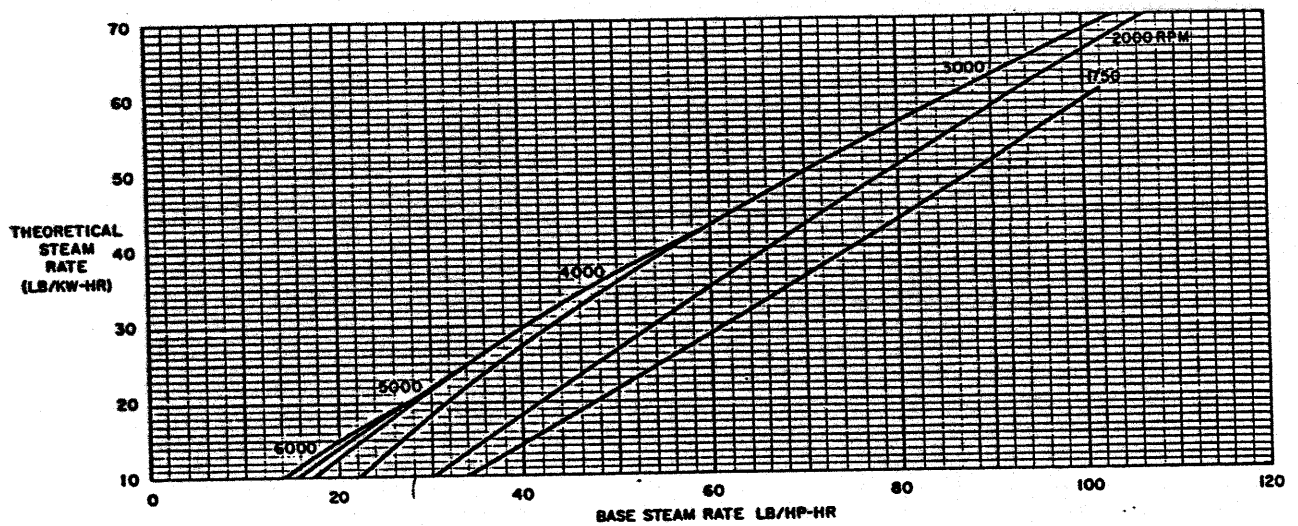
BYR - 18" WHEEL



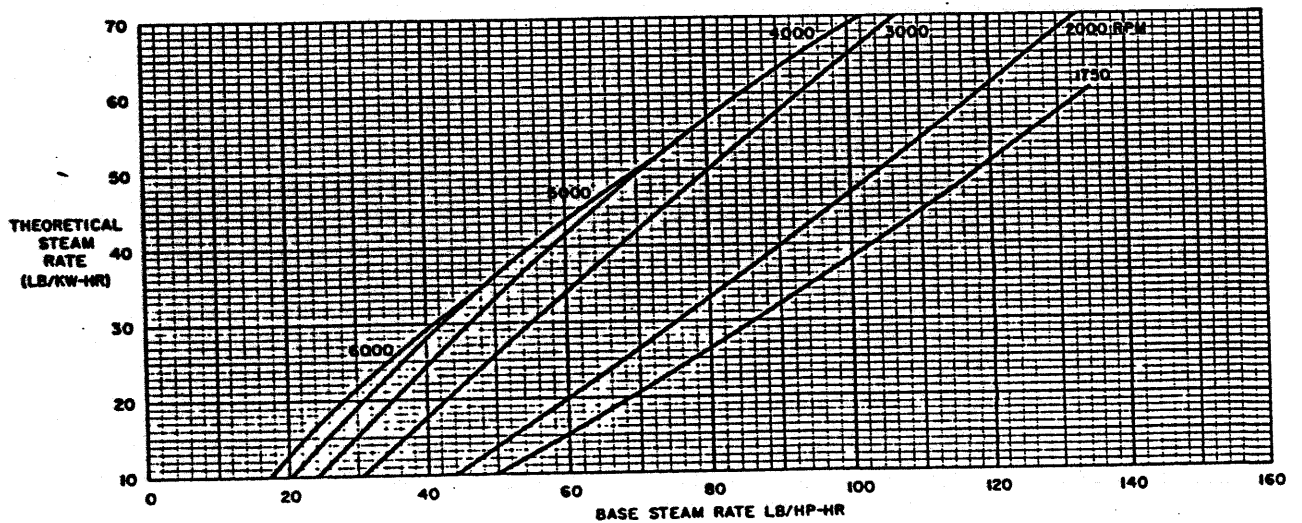
CYR-22" WHEEL



DYR-28" WHEEL

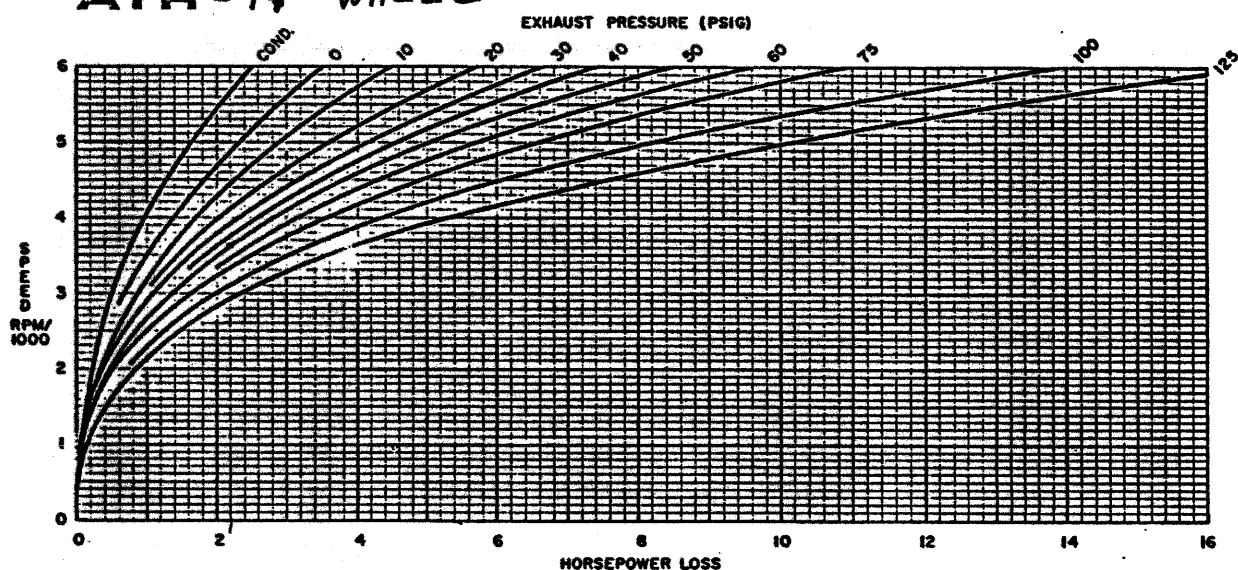


BYRH-18" WHEEL (HIGH BACKPRESSURE)

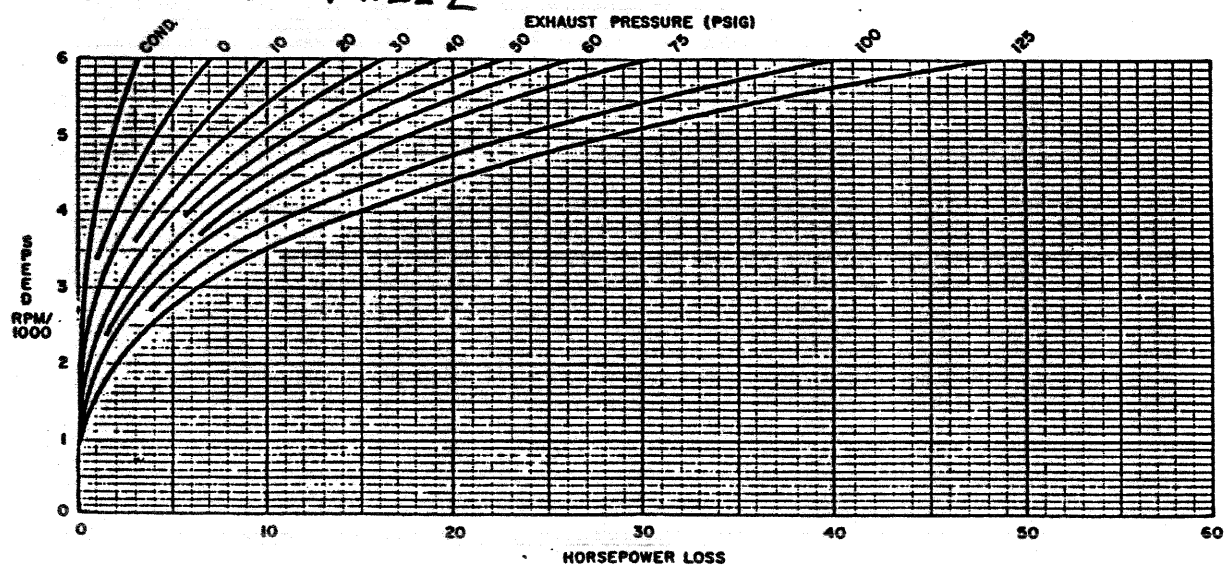


Horsepower Losses

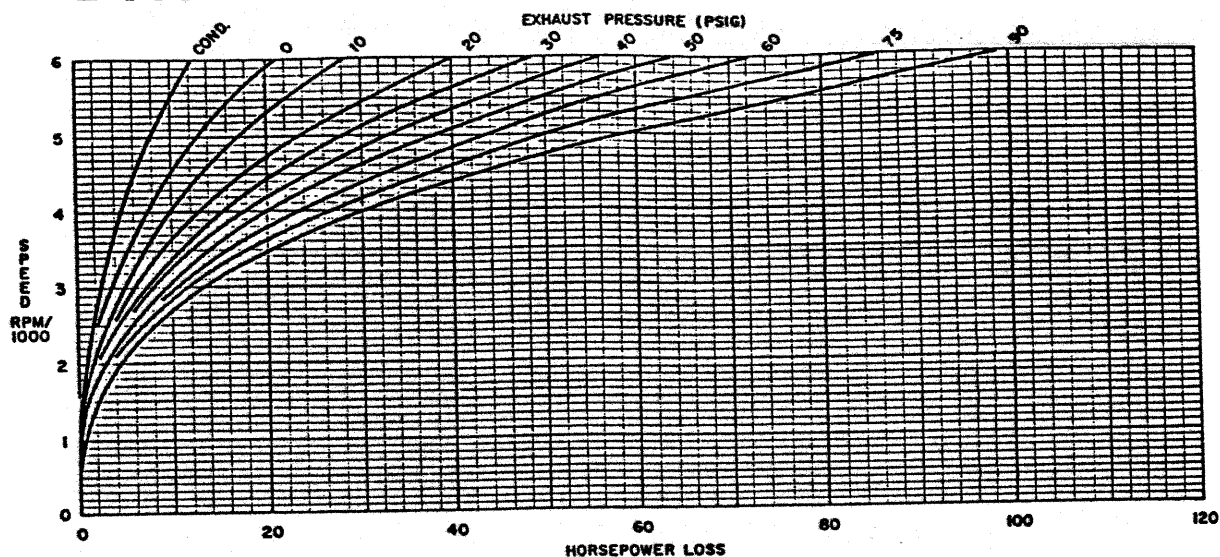
AYR-14" WHEEL



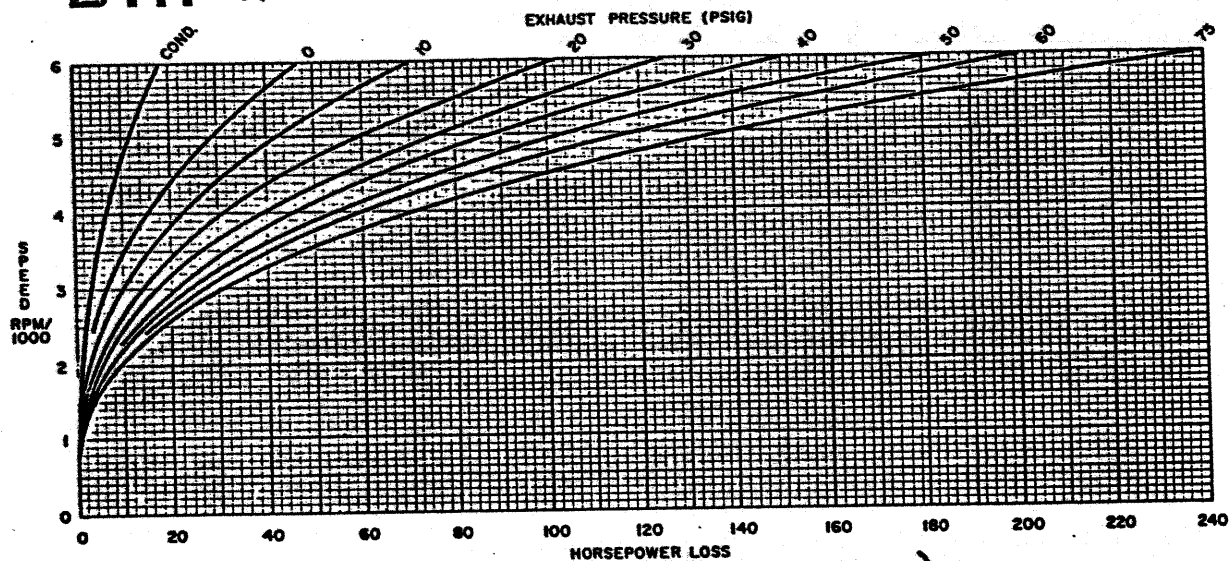
BYR-18" WHEEL



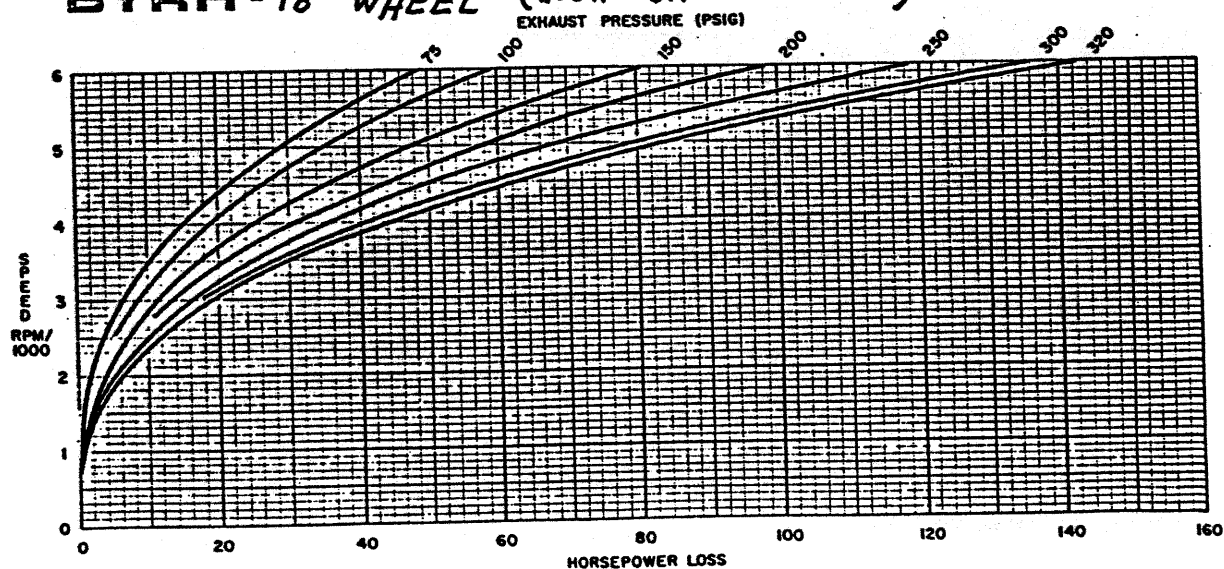
CYR-22" WHEEL



DYR-28" WHEEL



BYRH-18" WHEEL (HIGH BACKPRESSURE)

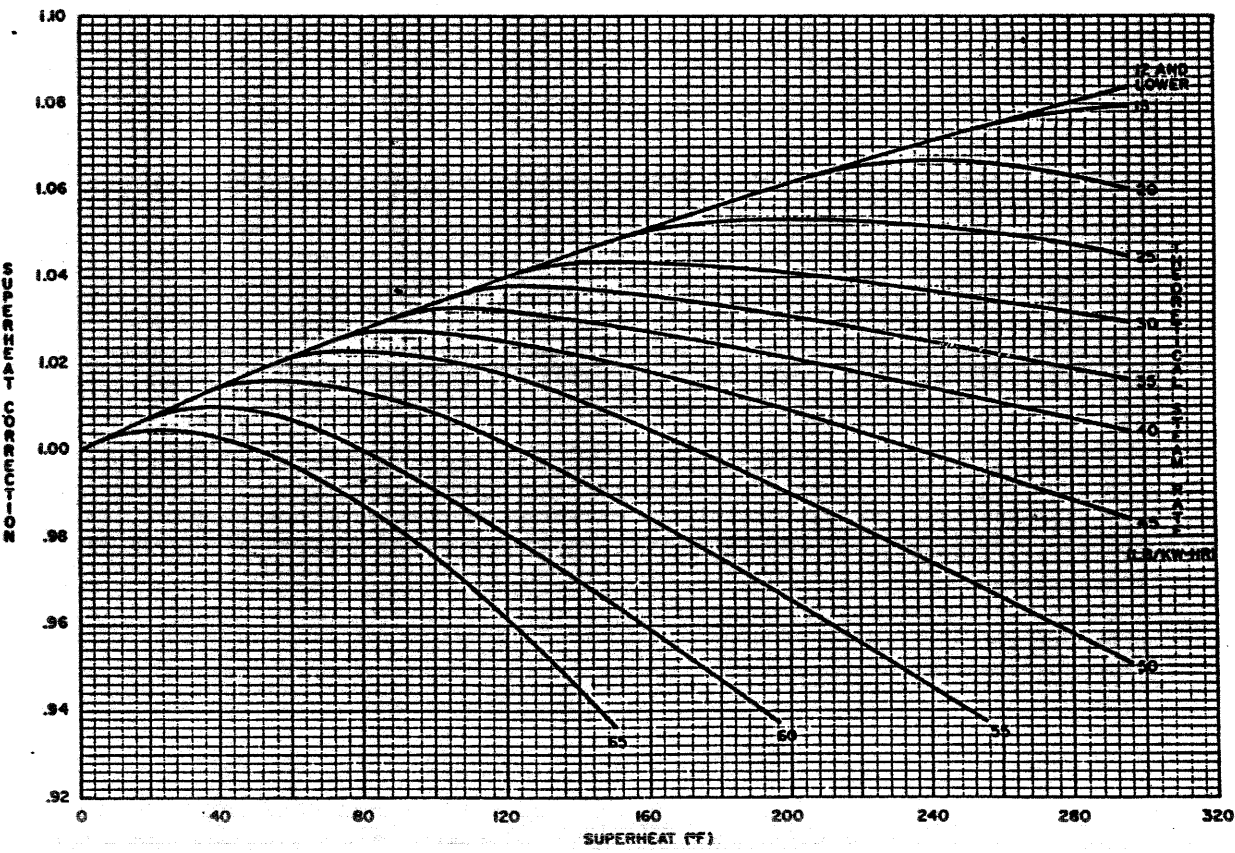


Superheat Correction

Temperature of Dry and Saturated Steam

To obtain superheat in degrees F, subtract temperature given in tabulation below from total initial temperature.

PSIG.	SATURATION TEMP. F	PSIG.	SATURATION TEMP. F	PSIG.	SATURATION TEMP. F	PSIG.	SATURATION TEMP. F
0	213	150	366	300	422	450	460
5	228	155	368	305	423	455	461
10	240	160	371	310	425	460	462
15	250	165	373	315	426	465	463
20	259	170	375	320	428	470	464
25	267	175	378	325	429	475	465
30	274	180	380	330	431	480	466
35	281	185	382	335	432	485	467
40	287	190	384	340	433	490	468
45	293	195	386	345	434	495	469
50	298	200	388	350	436	500	470
55	303	205	390	355	437	510	472
60	308	210	392	360	438	520	474
65	312	215	394	365	440	530	476
70	316	220	396	370	441	540	478
75	320	225	397	375	442	550	480
80	328	230	399	380	444	560	482
85	328	235	401	385	445	570	483
90	331	240	403	390	446	580	485
95	335	245	404	395	447	590	487
100	338	250	406	400	448	600	489
105	341	255	408	405	449	610	491
110	344	260	410	410	451	620	492
115	347	265	411	415	452	630	494
120	350	270	413	420	453	640	496
125	353	275	414	425	454	650	497
130	356	280	416	430	455	660	499
135	358	285	417	435	456	670	501
140	361	290	419	440	457	680	502
145	364	295	420	445	458	690	504



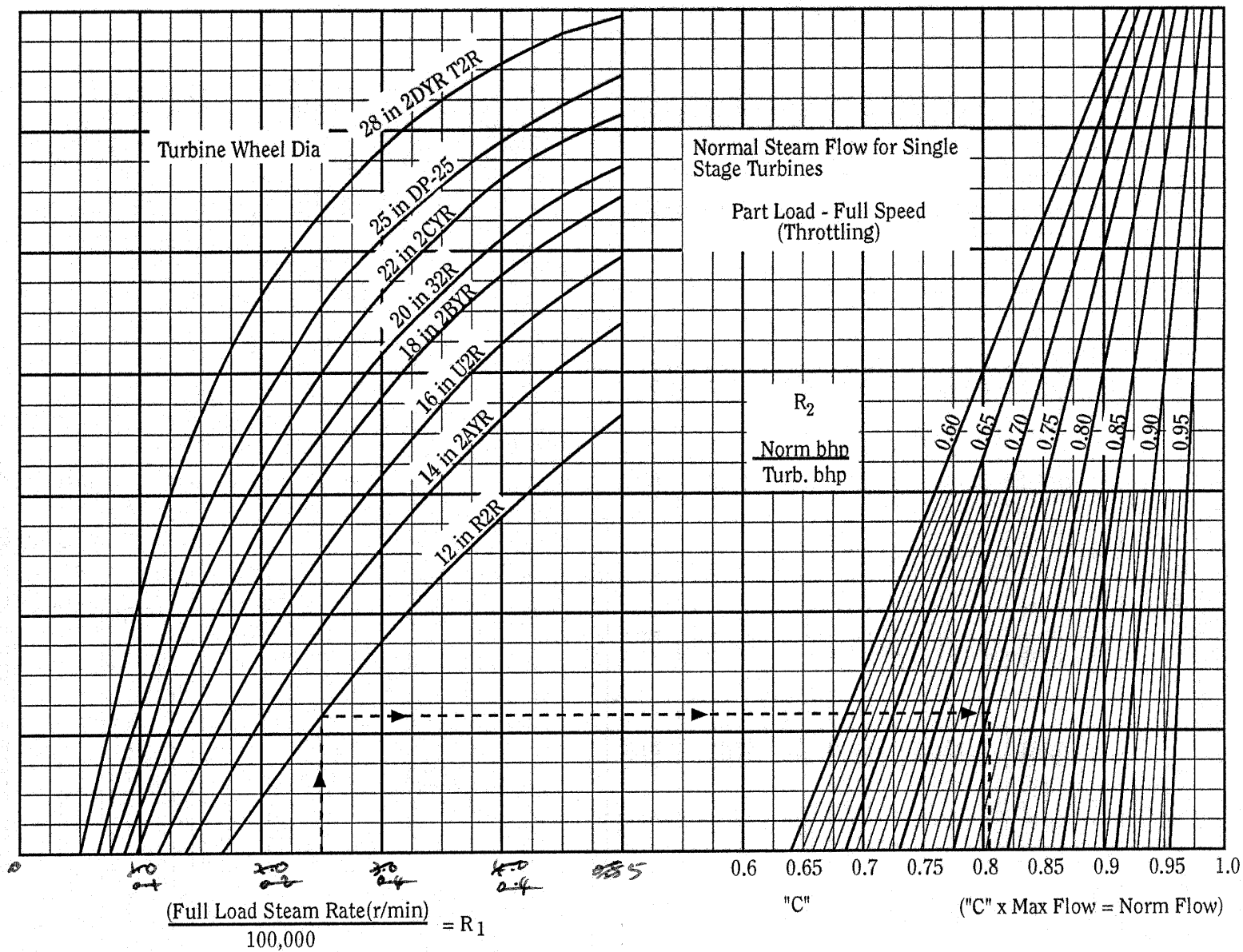
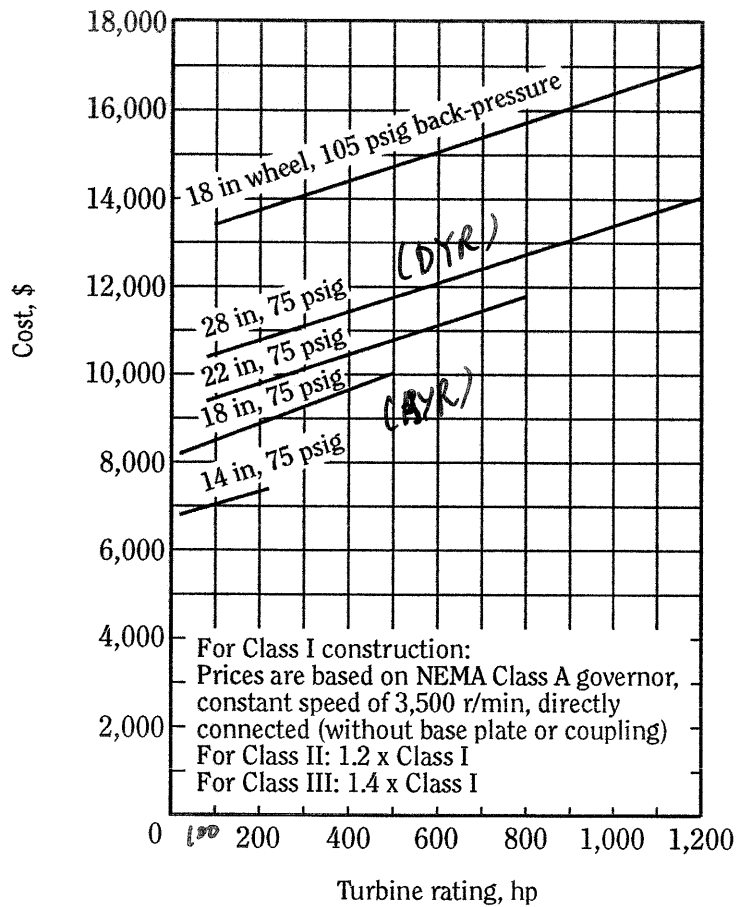


Figure 8

Based on Normal Horsepower, Should the AYR or the DYR Be Used

Figure 9

Steam Turbine Cost Comparison (1978 Prices)



AYR : \$ 7000
DYR : \$ 10400

multiply by 1.2

Approximate costs for single-stage steam turbines.

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CLASS	MAX INLET PSIG	MAX INLET °F	EXHAUST PSIG
I	250	500	75
II	600	600	75
III	250	750	75
IV	850	750	75

Assume:

Cost of Steam = \$3.50/1000 lbs

Operating time = 8000 h/a

8000 h/a

Using calculations and costs from Figure 9:

Turbine	Steam Used lb/h	Steam Cost \$/a	Capital Cost,\$ (Class II)
AYR (14 in)	3788	106,064	1.2 x 7000 = 8400
DYR (28 in)	3325	93,100	1.2 x 10400 = 12480

$$\text{Payout} = \Delta \text{ Capital Cost} / \Delta \text{ Annual Operating Cost}$$

$$(\$12,480 - 8,400) / (\$106,064 - 93,100) / a$$

$$= 0.315 \text{ Years}$$

25.2 Years

MULTISTAGE TURBINE EXAMPLE

Given:	Compressor	Turbine
rated Bhp =	7000	7700
rated r/min =	6000	6000
norm Bhp =	6300	6300
norm r/min =	5400	5400

Turbine Steam Conditions:

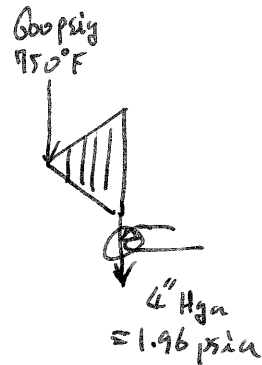
Inlet = 600 psig, 750 °F

Exhaust = 4" inHga = 1.96 psia

Find The Normal Steam Consumption:

TSR (Theoretical Steam Rate)

Use Mollier Diagram



$$h_1 = 1379 \text{ Btu/lb @ } 615 \text{ psia, } 750^\circ \text{F}$$

$$h_2 = 932 \text{ Btu/lb @ } 1.96 \text{ psia, } (s_1 = s_2)$$

$$TSR = (3413 \text{ Btu/kWh}) / (1379 - 932) \text{ Btu/lb}$$

$$TSR = 7.64 \text{ lb Steam/kWh}$$

Efficiency at **Rated** Conditions:

Efficiency $\cong$ 76 pct (from Figure 10) *page 18*

Corrections for normal hp & speed:

$$\text{norm bhp} / \text{Turbine rated bhp} = (6300/7700) = 0.82$$

$$\therefore \text{correction} = 1.035 = f_1 \text{ (from Figure 11)}$$

$$\text{norm } r / \text{min} / \text{rated } r / \text{min} = 5400/6000 = 0.90$$

$$\therefore \text{correction} = 1.05 = f_2 \text{ (from Figure 11)}$$

Normal Steam Consumption:

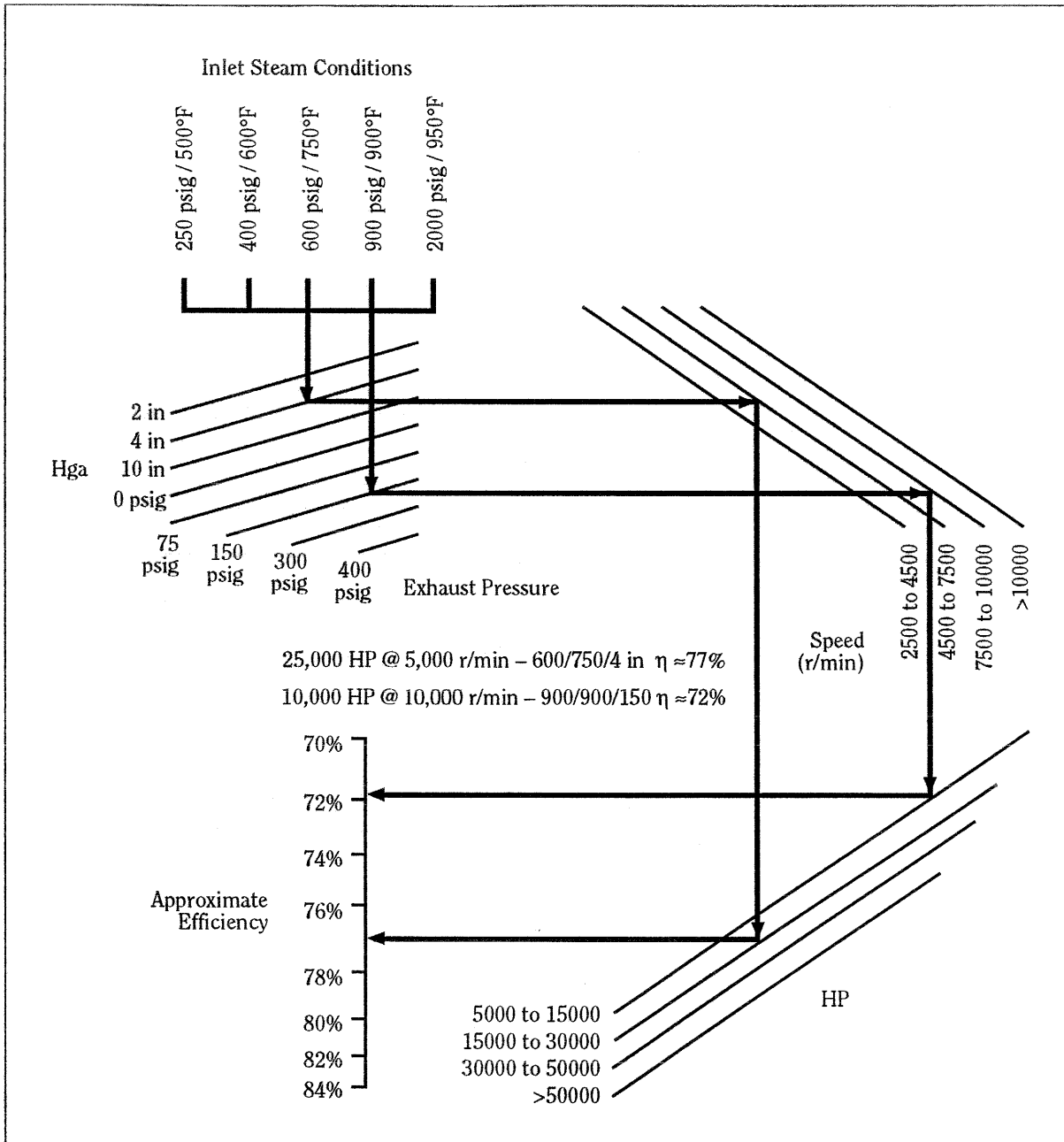
$$[(TSR \times 0.746 / \text{Efficiency})] \times f_1 \times f_2 \times (\text{norm bhp}) =$$

$$[7.64 \text{ lb/kWh} \times 0.746 \text{ kWh/hph}] \times 1.035 \times 1.05 / 0.76 \times 6300 \text{ hp}$$

$$= 51,344 \text{ lb/h}$$

Figure 10

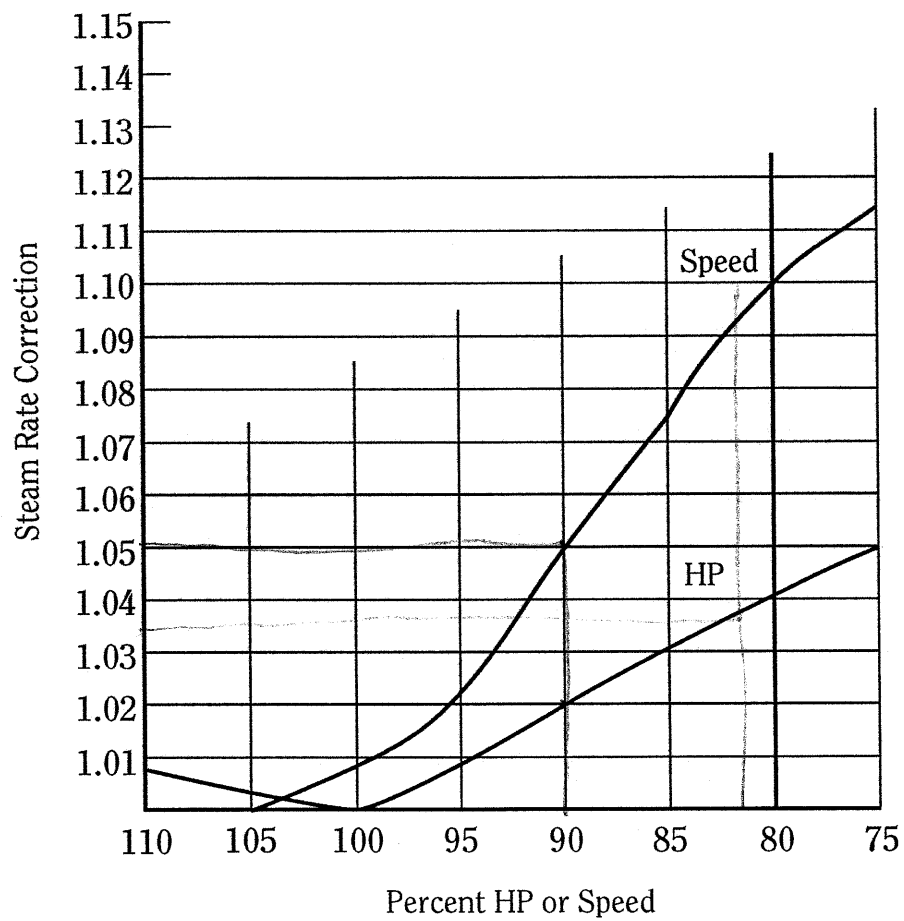
Multistage Turbine Efficiency



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Figure 11

Part-load/speed correction curves

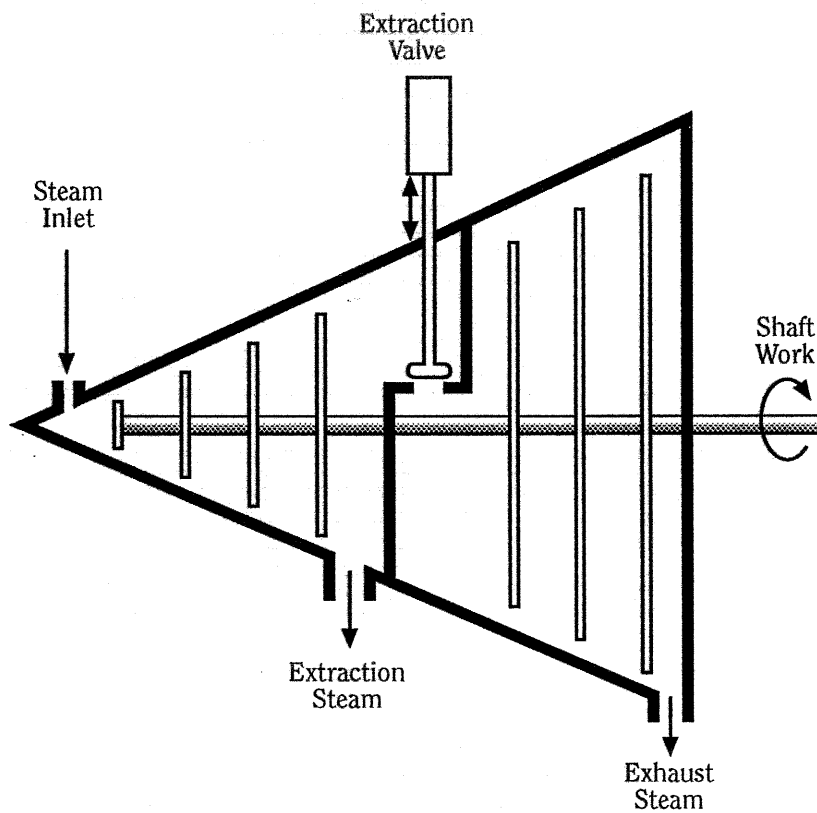


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EXTRACTION TURBINES

Figure 12

Extraction Turbine



EDS-R00-1208

$$\text{STEAM IN} = \text{EXTRACTION OUT} + \text{EXHAUST OUT}$$

$$0 + 140,000 \text{ lb/hr}$$

$$= 50,000 + 0$$

$$210,000 \text{ lb/hr} = 100,000 + X_{\text{exh}}$$

$$X_{\text{exh}} = 110,000 \text{ lb/hr}$$

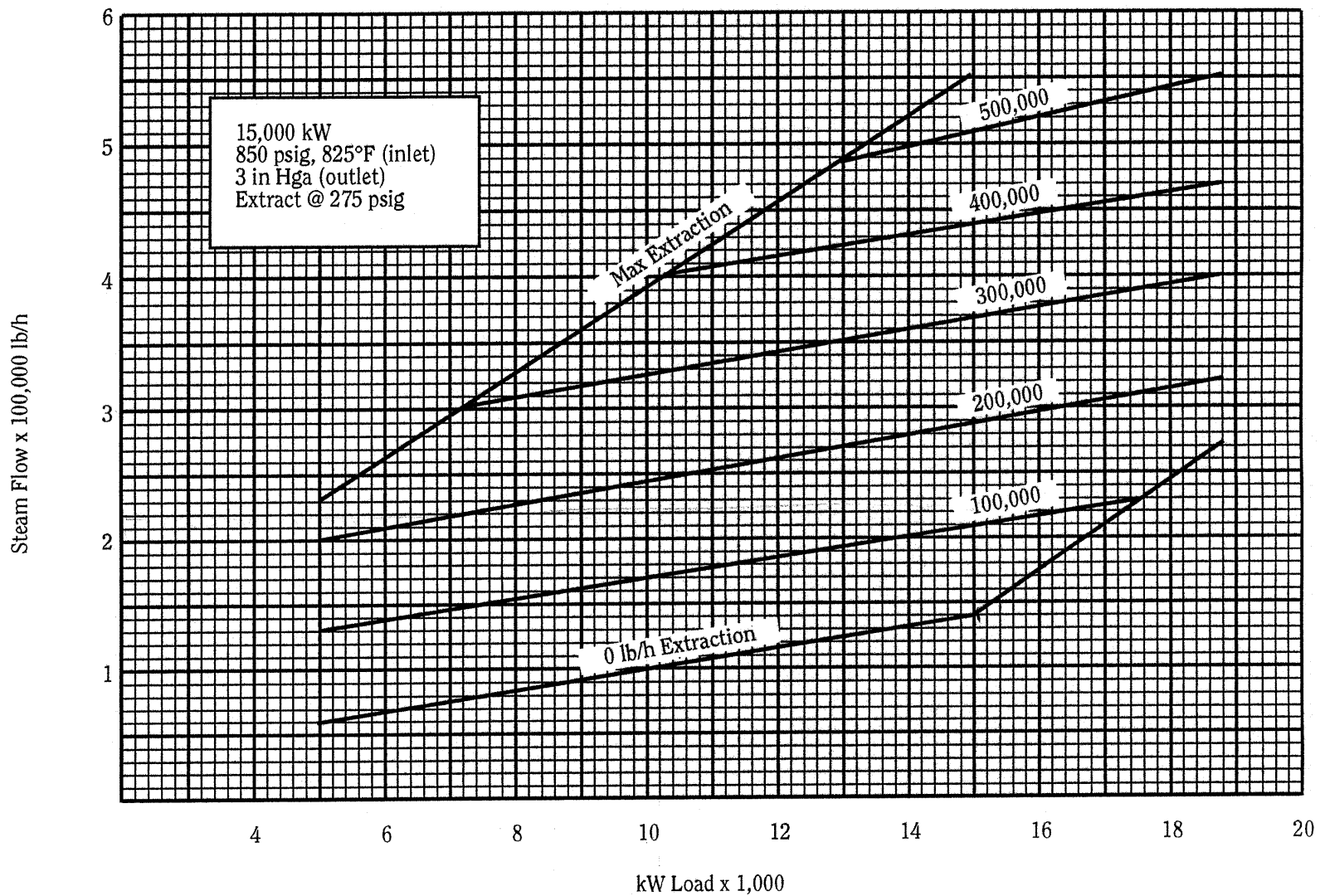
Typical Extraction Turbine Performance Curve

Figure 13

Use of Extraction Turbine for Efficient Steam Header Balance

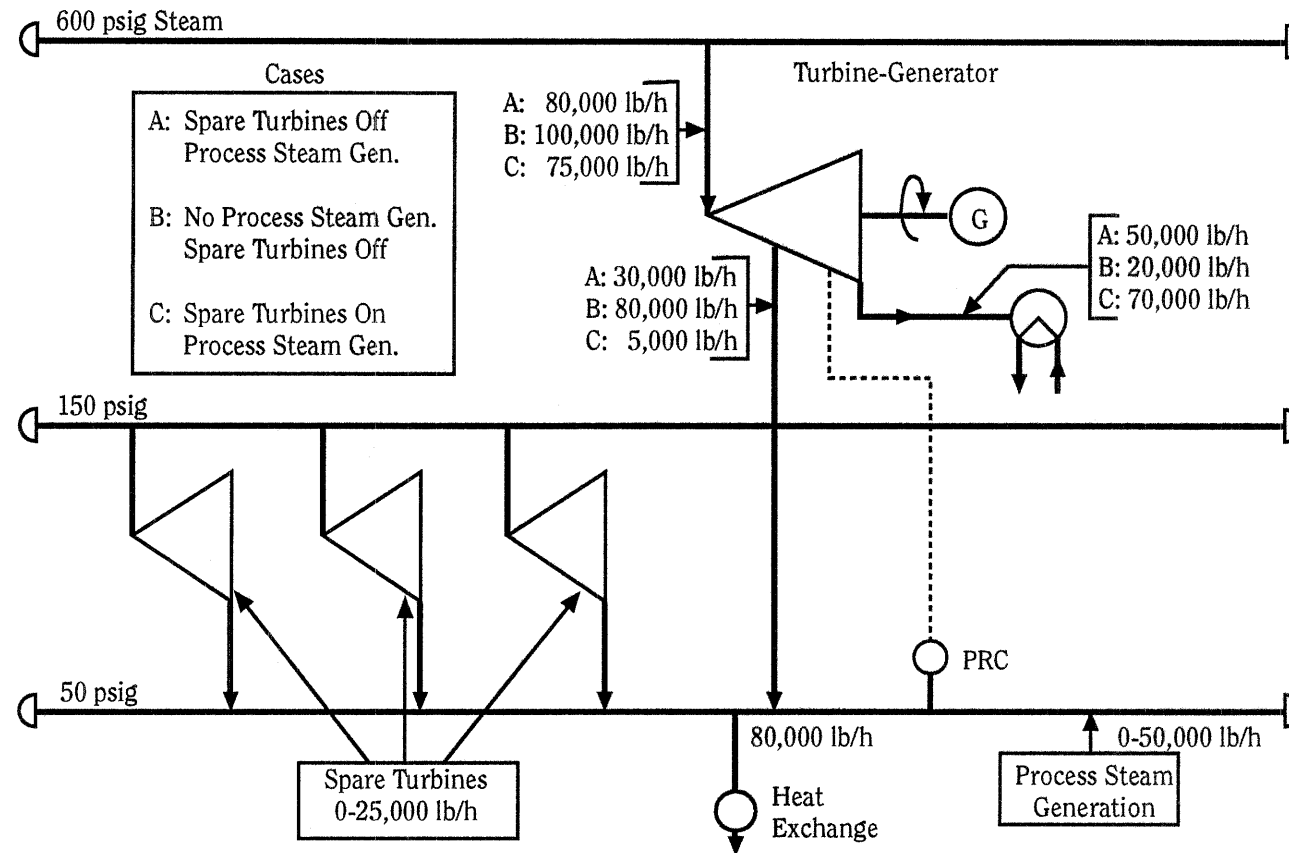
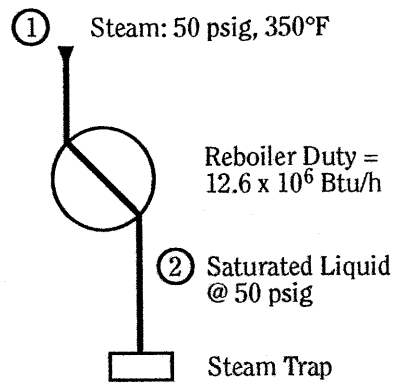


Figure 14

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Heat Exchanger

Figure 15



EDS-R00-1210

Steam Consumption:

$$h_1 = 1207 \text{ Btu/lb@ } 65 \text{ psia \& } 350^\circ \text{F}$$

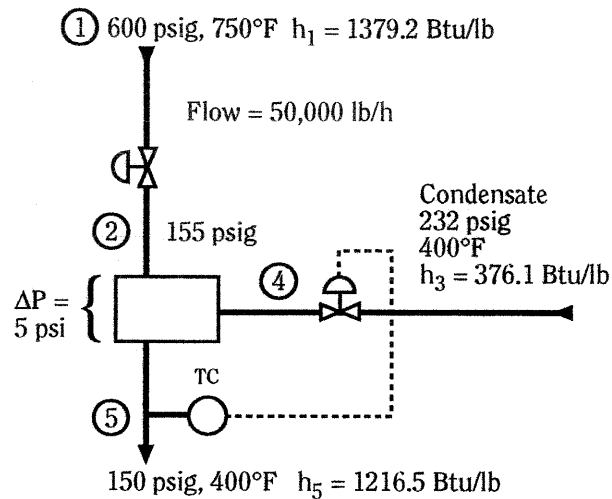
$$h_2 = 267.6 \text{ Btu/lb@ } 65 \text{ psia (sat liquid @ } 298^\circ \text{F)}$$

$$12,600,000 \text{ btu/h} / (1207 - 267.6) \text{ btu/lb} =$$

$$\text{Steam Consumption} = 13,413 \text{ lb/h}$$

Control Valves/Desuperheaters

Figure 16



EDS-R00-1211

Temperature Into Desuperheater

$$h_2 = h_1 (\text{is enthalpic expansion})$$

$$\text{For } h_2 = 1379.2 \quad T = 707^\circ \text{F} @ 170 \text{ psia}$$

Desuperheater Water (W lb/h)

$$(\text{Mass of Steam in}) = (\text{btu/lb Steam in})(\text{Mass Water in})(\text{Btu/lb Water})$$

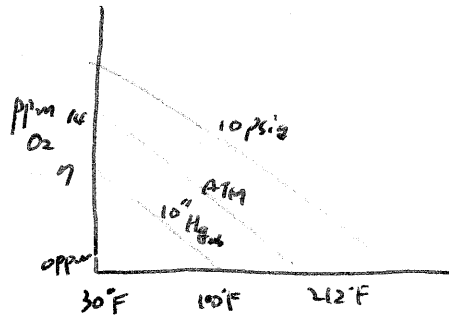
$$= (\text{Mass of Steam Out})(\text{Btu/lb Steam Out})$$

$$50,000 \times 1379.2 + W \times 376.1 = (50,000 + W) \times 1216.5$$

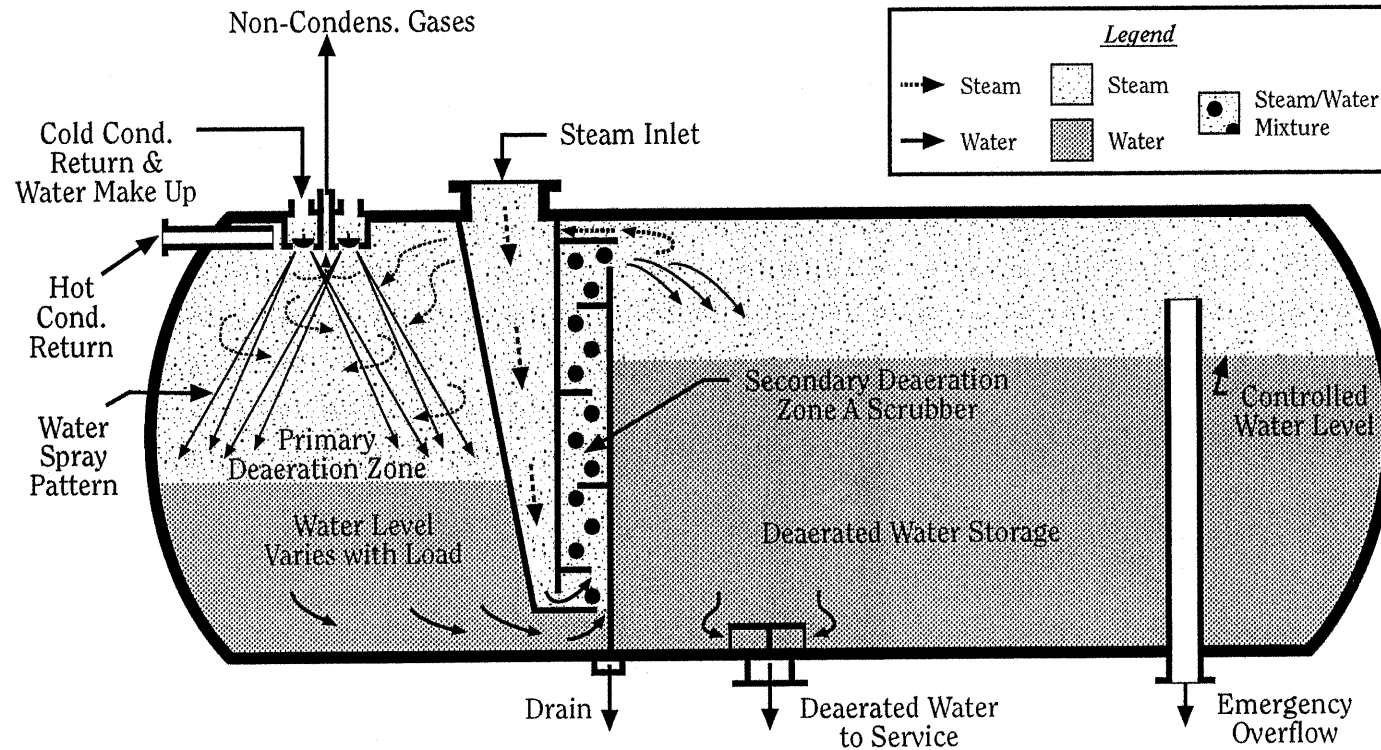
$$W = 9680 \text{ lb/h}$$

DEAERATORS

Figure 17



Spray Type Deaerator



EDS-R00-1212

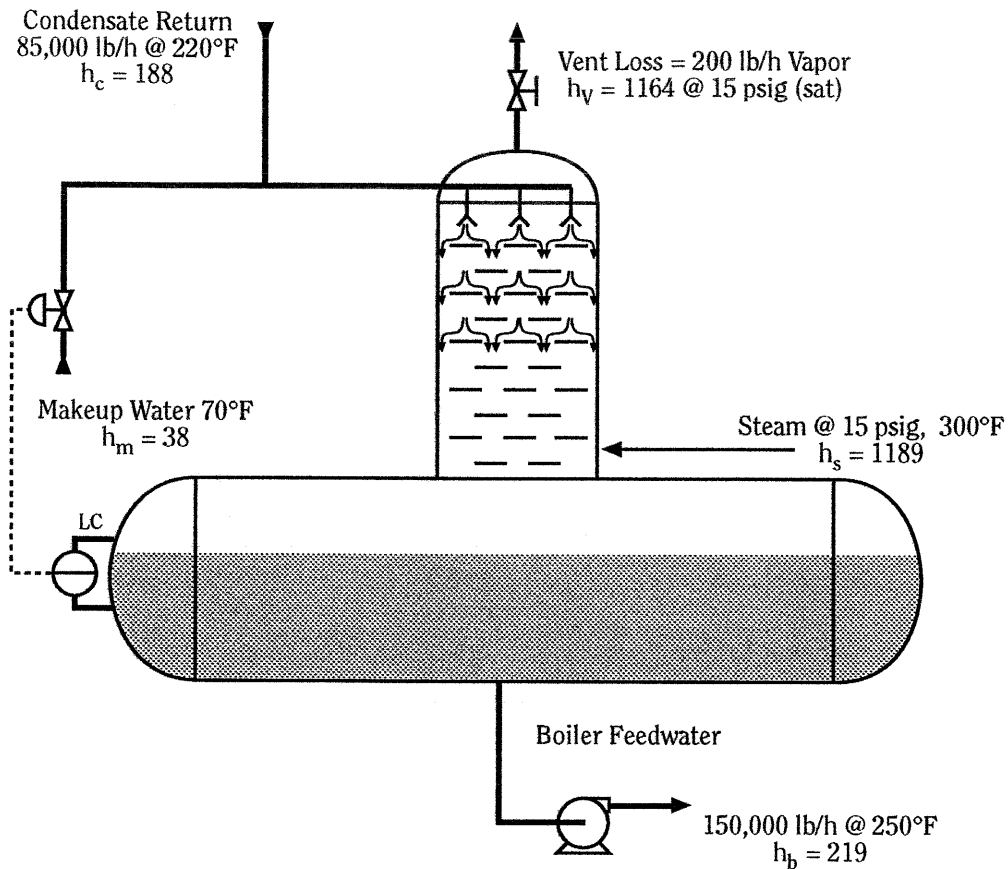
9 ppm → 9 ppb → 0
 { .15 ppm Residual }
 .05 ppm
 (HYDRAZINE) N₂H₄
 N₂H₄ + O₂ → 2H₂O + N₂↑
 1 ppm Hydrazine → 1 ppm DO

SODIUM SULFITE
 Na₂SO₃
 2Na₂SO₃ + O₂ → 2Na₂SO₄
 8 ppm Sodium Sulfite ⇒ 1 ppm DO
 BOILER PRESS.
 300-600
 600-900
 > → Hydrazine

Residual
 20-30 ppm SO₂
 10-15 ppm SO₃

Figure 18

Tray Deaerator



EDS-R00-1213

How much makeup water and steam required?

Let:

m = lb/h Makeup Water
 c = lb/h Condensate
 v = lb/h Vent
 s = lb/h Steam In
 b = lb/h Boiler Feedwater

Mass Balance

$$In = Out$$

$$m + c + s = v + b$$

$$m = v + b - c - s$$

$$m = 200 + 150,000 - 85,000 - s$$

$$m = 65,200 - s$$

Heat Balance

$$In = Out$$

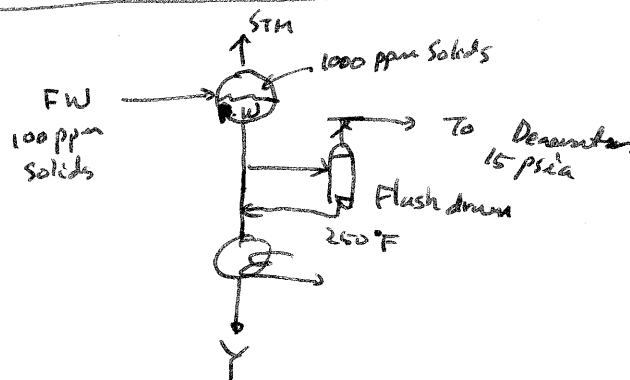
$$m \times h_m + c \times h_c + s \times h_s = v \times h_v + b \times h_b$$

$$(65,200 - s) \times 38 + 85,000 \times 188 + s \times 1189 =$$

$$200 \times 1164 + 150,000 \times 219$$

$$s = 12,707 \text{ lb/h} = \text{Steam In}$$

$$m = (65,200 - s) = 52,493 \text{ lb/h} = \text{Makeup Water}$$



A REALISTIC APPROACH TO ENERGY CONSERVATION

ABSTRACT

This paper surveys the techniques used in designing and operating for energy conservation in the process industries, with emphasis on analyzing the refinery system as a whole rather than as a collection of discrete entities. The potential for waste heat recovery and the methods actually employed are illustrated. Covered are such topics as heat integration by direct and indirect methods, heat pumps and the process engineering of combustion. The thermodynamic principles involved and the definition of co-generation are presented, along with the technological potential and the implications for future design.

In the existing plants the designer has built in certain capabilities that can be used for conservation. Some opportunities are presented and the strategy of "monitor, maintain and modify" are discussed.

INTRODUCTION

Rapidly changing economic and operating constraints have strained industry and also have put a great burden on the designer. As a process designer, UOP has needed to reexamine virtually every step of hydrocarbon processing: from the big picture of feed and product slate to the details of improving the efficiency of pumped systems. One must remember that the goal is not energy conservation only, but to achieve the most economical processing systems that will accomplish production objectives while simultaneously satisfying all constraints of safety, pollution control, reliability of operation and energy allocation.

The Process Plant As A System

When examining the processing steps used in refining for fuel production or petrochemicals, one should keep in mind that a refinery is a system. Figure 19 illustrates this principle. The energy consumed in processing a crude oil or other charge stock into refined products shows up either as purchased fuel and electric power or as lower saleable product recovery. Consequently, any processing change of alternative scheme must be evaluated on the basis of its probable eventual effects on what "crosses the fence". For instance, for the same charge rate, the change must either decrease purchased energy or increase saleable product. Obvious as it may seem, this fact is easily forgotten when evaluating energy saving schemes, especially those dealing with low temperature heat distribution systems.

Approaches to Energy Conservation

Most of the techniques used in designing for energy conservation can be grouped into two basic categories:

- a. "First law" approach or waste heat utilization analysis
- b. "Second law" approach or thermodynamic efficiency analysis.

The most readily acceptable schemes have come from the effective utilization of "waste heat" - heat that in the past simply has been rejected to the environmental sink via air coolers, cooling towers, flare and smoke stacks and radiation.

The "second law" approach is concerned with the relative utility of energy levels, recognizing, for example, that Btu's in shaft work are worth more than Btu's at 600°F, and that Btu's at 600°F are worth more than Btu's at 150°F. This concept is used only qualitatively in most waste heat analyses.

Energy conservation through improvement in thermodynamic efficiency always has been a goal of the energy system designer. Historically, low fuel costs have precluded many possible applications. As energy conservation becomes a necessity, improving thermodynamic efficiency then must be routinely considered in the design of every system.

WASTE HEAT UTILIZATION

The primary goal of all energy conservation schemes is to reduce the quantity of heat rejected to the environment--often referred to as "waste heat management"--by recycling the heat back into the process. A familiar example would be the use of additional heat exchanger surface between process streams, preferably between interconnected streams like, reactor effluent versus feed, and fractionator product versus feed. The quantity of heat exchanged is limited by any of a number of constraints, viz., cost of the additional heat exchange surface, increased pressure drop through the system which raises operating and capital cost of pumps and compressors and reduced flexibility and control of the process.

Process Integration For Heat Recovery

After the practical limit has been reached on heat exchange between coupled or interconnected streams, large quantities of heat may still be rejected to the atmosphere in some systems, while other unassociated processes and offsite systems require heat input. There are two choices: one would be to integrate or couple the systems together directly and the other would be to utilize an independent vehicle such as steam generated from waste heat to move the recoverable heat to where it may be used.

There are, of course, advantages and disadvantages to these choices. For example, use of steam requires two sets of exchanges that in the case of equal heat transfer coefficients means a minimum of four times the surface area, Figure 20. When the temperatures of the source and sink are close together, it may not be practical to use a heat transfer fluid because the two temperature differentials become so small.

The main advantage offered by using a heat transfer fluid is unit operability. The heat source and sink are not directly connected but interfaced by a steam, water or hot oil system that, together with a back up system, should be able to accommodate the shutdowns, process changes and turn downs that could upset directly coupled streams in unassociated units. A heat transfer fluid scheme also facilitates distribution of heat collected from many small sources to larger sinks.

Some compromise must be made between integration and operability. Heat integration has been an old standard practice with Fluid Catalytic Cracking (FCC) and Gas Concentration units. Circulating oil streams are used to transfer heat from the main column, at a wide range of temperatures, to the gas concentration unit where the heat is used in strippers and fractionators to separate the lower boiling reactions products. This concept has recently been extended to other FCC associated units that have not previously been integrated, i.e., HF Alkylation and Butamer process units. The FCC unit can supply much of the heat required for fractionation in these units, especially if the overhead reflux on the FCC main column is reduced to the minimum required for fractionation. Heat is made available at a useable temperature level for these associated units.

The type of heat integration must be handled with care, however. There must be some "handle" so that required processing conditions can be maintained. This control can be either a demand for outside heat or a heat rejection from the units that can swing freely with processing changes.

A different type of integration involves two process units in series, where the product from one unit becomes the feed to the other. A common example would be a naphtha hydrotreating-UOP Platforming® process unit, where the hydrotreated naphtha is fractionated to obtain the required initial boiling point for the Platforming unit feed (see Figures 21 and 22). The splitter column bottoms product is the feed to the Platforming reactor section. Close attention to the degree of exchange between splitter column bottoms and feed is required for maximum economy. If the raw oil feed to the Platforming reactor section is high (no feed-bottoms exchanger), the temperature to the reactor products condenser will be high. This means that more heat must be rejected to the atmosphere in the reactor products condenser. Since it is desirable to reduce the temperature of the reactor effluent to the products condenser it is necessary to increase the heat exchanged between the splitter column bottoms and the feed. This not only requires additional surface in the splitter feed-bottoms exchanger, but also more surface

Heat Pumps For Heat Recovery

in the Platforming reactor feed-effluent exchanger to compensate for the colder feed temperature. This, in turn, could increase gas compressor capital cost as well as add to the cost of compression of the recycle gas. The consequence of increasing the splitter feed-bottoms heat-exchange is to effect a fuel savings by reducing the splitter reboiler duty. Too much feed preheat, however, can have a detrimental effect on the fractionation performance of the splitter column which can be restored only by increasing reflux. Obviously, a balance must be found for the right amount of feed-bottoms exchange. Careful considerations of capital cost, energy consumption and controllability must be made before the most efficient and profitable way is chosen to connect the two units.

There are many instances in the process industry where large quantities of heat at a temperature too low to be useful must be rejected to the atmosphere. Much attention has been given to the potential for heat pumping, or raising the temperature of waste heat to a useable level. UOP has designed fractionation facilities to separate propane-propylene mixtures where a heat pump was used and a comparison with a conventional fractionation system is interesting. A conventional fractionation scheme for splitting propylene-propane is shown in Figure 23. The temperature of the cooling medium available sets the minimum allowable pressure to run the column so that the overhead product can be condensed. This fixes the top and bottom temperatures and the relative volatility (α), from which the reflux rate and number of trays is calculated.

If a colder heat sink were available, the column pressure could be reduced, which results in an increasing in the relative volatility and a reduction in the reflux rate and number of trays. A refrigerated column scheme is shown in Figure 24. The operating pressure has been lowered to 620 kPa(90 psia), the reflux ratio to 6.5 and the number of trays to 170. Figure 25 shows a classical heat pump scheme, where the overhead vapors are compressed to a pressure where the propylene bubble point temperature exceeds the temperature in the reboiler. The operating conditions of the column are the same as for the refrigerated condenser case but a considerable reduction in energy consumption has been achieved.

It is possible to design a "reverse" heat pump system as is shown in Figure 26. Here the propane product is flashed to a lower pressure where its dew point temperature is lower than the bubble point temperature of the overhead vapor and it is used as the condensing medium. the vaporized propane, after compression, supplies heat to the column. For the

case selected, this scheme showed lower power consumption than the classical heat pump case.

To determine the energy savings between the various schemes presented in Table 1, the consumptions must be related to what "crosses the fence". If the compressor is motor driven, 7,750 kw can be added to the purchased electricity, which is equivalent to about \$1.8 MM/per year where power costs 3c per kw. If the reboiler heat requirement is to be supplied by firing fuel directly in a heater, the fuel cost would be about \$6.5 MM per year where fuel costs \$2/GJ. But 54°C (130°F) is rather a low temperature to use a fired reboiler heater. In many places throughout the refinery heat is rejected to the environment at temperatures greater than the required 54°C (130°F). There may be condensing steam turbine duties that could be switched to back pressure stream turbines exhausting at near atmospheric pressure that would provide less expensive heat. It is impossible to say which scheme is superior in a general sense. Each application must be considered within the constraints and environment of a specific installation, and the energy consumptions and cost related to what "crosses the fence".

Fractionator Heat Cascading

Besides compression, there are other methods available to the designer of increasing the temperature of waste heat. Perhaps the simplest way is to produce the waste heat at a higher temperature. Heat cascading distillation utilizes the latent heat of vaporization of the fractionator overhead vapor to reboil other associated columns in a fractionation train. The bubble point of the overhead vapor of one column is raised by increasing the pressure of that column. This, of course, lowers the relative volatility (α) of the high pressure column which means that more reflux (hence more boiler duty) and trays are required for a given split. The benefit is that the other low-pressure columns can be reboiled by condensing the overhead vapor from the high-pressure column (see Figure 27).

This type of system has been demonstrated commercially by UOP at the Aromatic Complex of Veba Chemie, Gelsenkirchen, West Germany. Here the xylene splitter overhead vapors are used to reboil two columns in the UOP Parex process unit. Although particularly well suited to aromatics processing, this technique or its variants could give significant savings in many applications, and, in fact, has shown consistently better economics than heat pumping schemes when fractionating higher boiling materials. Table 2 summarizes the economic advantages of this processing scheme.

The single largest consumption of energy in refineries is the combustion of fuel directly in process heaters to transfer heat to process streams. Since combustion is simply another chemical reaction, it would make sense to attack the combustion of fuel with the same techniques employed for process units.

A first law, or waste heat analysis, of a fired heater usually compares heater performance by defining a combustion efficiency:

$$\text{Efficiency, } P_{ct} = \frac{\text{Heat "usefully" absorbed by process} \times 100}{\Delta H_{c \text{ of fuel used at } 15^{\circ}\text{C} (60^{\circ}\text{F})}$$

Given complete combustion, the single most important parameter affecting heater efficiency is the temperature of the flue gas leaving the heat absorption section. This temperature is limited by the inlet temperature of the coldest fluid being heated or by the dew point of the flue gas. If the fuel is low in sulfur, and if the primary fluid has a high heater inlet temperature, a secondary fluid requiring heat at a lower temperature can be utilized to lower the stack gas temperature. The secondary fluid can be another process fluid, steam, boiler feed water, or the ambient temperature combustion air itself.

The addition of an air preheater, as shown in Figure 28, is the usual direct way to lower fuel firing in process heaters without tying together unassociated streams of the steam system. Furthermore, generating steam lower than boiler plant pressure is not as efficient as an air preheater. An air preheater of any type also would require that forced draft fans be used; a liability that can be turned into an asset by providing positive flow control of the combustion air for maximum efficiency.

THERMODYNAMIC EFFICIENCY

The classical waste heat approach to energy conservation is highly beneficial and has led to the development of more efficient processing plants. However, even after this kind of waste heat utilization has been maximized, large quantities of heat still must be transferred to and rejected from the process. The systems approach is used to distinguish quantitatively between schemes. The second law of thermodynamics can be used to indicate the degree that energy use can be minimized.

Energy is often defined as the "ability to do work". Any process involving heat transfer can be described in terms of work, e.g., the work of reaction or the work of separating components in a distillation column. A refinery can be thought of as a machine. Crude oil is fed in and the machine performs work on the crude, changing it into products. Energy in the form of fuel and electricity is used to power the machine, and the exhaust heat is rejected to the environment, as was shown in Figure 32.

Just as the efficiency of a compressor relates the actual amount of work required to the theoretical minimum, the refinery thermodynamic efficiency relates the actual energy used in a refinery to the theoretical minimum energy necessary. Thermodynamic efficiency can be defined as:

$$\text{Thermodynamic efficiency} = \frac{\text{Work actually performed by a process}}{\text{Work that could have been done, had all steps in the process been reversible}}$$

The quantity of work that a given amount of heat has the potential to perform in a heat engine is dependent upon the absolute temperature at which the heat is available, and upon the absolute temperature at which the heat can be exhausted:

$$\text{Work (maximum)} = \frac{T_{\text{hot}}}{T_{\text{hot}} - T_{\text{cold}}}$$

As an example, a fractionating column might have an overhead condenser rejecting 15 megawatt (51.2 MM Btu/h at 100°C(212°F)). If cooling water is available at 29°C(85°F), the maximum amount of work that could be produced by a heat engine operating between the column overhead and the cooling water is:

$$\text{Work (maximum)} = 15 \text{ MW} \times \frac{(100 + 273.15) - (29 + 273.15)}{(100 + 273.15)} = 2.8 \text{ MW} = 2800 \text{ kW} (3750 \text{ hp})$$

The actual amount of work that could be generated is much lower, because finite temperature differentials are required to transfer the heat. The working fluids (water, butane freon, etc.) are not perfect, and the pumps and expansion turbines are less than 100% mechanically efficient, but this simple expression can do much to indicate which path energy transfer should take in order to reduce consumption.

Co-generation can be thought of as the concept of producing work, as a by-product, whenever, the principal concern is the transfer of heat through large temperature differentials.

Although the combustion efficiency of a conventional furnace is high, the thermodynamic efficiency is low. This is because of the transfer of heat from the hot flame at 1400-1600°C(2500-3000°F) to a process fluid at relatively low temperatures 260-540°C(500-1000°F). The high potential of the flame is lost through the irreversible process of heat transfer. The concept of co-generation employs a heat engine to transfer heat from the flame to the process. The heat engine utilizes the high temperature potential of the flame as the heat source and the process fluid as the heat sink.

The principle of co-generation can be used in a topping cycle at the high temperature end of the process where the heat engine operates between the flame and the process fluid. The heat engine can be a gas turbine, diesel engine or steam turbine as shown in Figure 32. In the bottoming cycle, the heat engine operates between the process fluid and the atmospheric sink as in Figure 33. Either low pressure steam or some other desirable fluid can be employed in a Rankine cycle. When these new elements are incorporated, the processing system takes on a new appearance as is shown in Figure 34.

Savings From Co-Generation

A more concrete example of how co-generation can be used in the processing system can be illustrated by considering a hot oil belt. Hot oil belts are widely used, especially in petrochemical complexes. One central furnace supplies heat to a circulating organic liquid that, in turn, supplies the various process heat requirements via individual exchangers (Figure 35).

Assume, for example, that the process absorption rate is set at 36.6 MW (125 MM Btu/h), and that the circulating liquid picks up 7.3 MW (25 MM Btu/h) from waste heat sources. The heater must supply 29.3 molecular weight (100 MM Btu/h) to heat the oil from 260-340°C (500 to 644°F)

An obvious example of co-generation would integrate an electric generating plant with the processing plant. High pressure, superheated steam would be generated by the power

plant, expanded through topping turbines and then exported to supply the heat needs of the processing plant. The difficulty in employing steam as the transfer media to heat the oil belt is the high temperature required. For steam to heat a fluid to 340°C(644°F), the steam pressure would have to be greater than 16000 kPa(2300 psi).

To utilize the principle of co-generation with the hot oil system, a gas turbine or diesel engine must be employed.

The fuel fired equivalent of a conventional heater, shown in Figure 36, can be improved by the addition of an air-preheater to lower the stack gases to 150°C (302°F), or lower depending upon the SO₂ content of the flue gas (Figure 37).

The concept of co-generation would recognize the potential of generating work with a simple cycle gas turbine and using the exhaust gases in a heat exchanger to heat the process fluid (Figure 38). This case uses considerably more fuel than even the conventional heater without air preheat. This is because larger amounts of air are required to drive the turbine, 70 kg of air per kg of fuel (70 lbs of air per lb of fuel) than combustion air, 19 kg of air per kg of fuel (19 lbs of air per lb of fuel) in the conventional heater. The overall economy of this scheme depends upon how the 340°C(644°F) stack gases are utilized. There is no simple method to make the system self-contained. Preheating the air to the turbine would only decrease the efficiency of the turbine cycle. After compression, the air would be too hot to pick up additional heat from the stack gas.

The stack gases must be used to supply some lower temperature heat requirement. If the stack gases can be cooled from 340°C(644°F) to 150°C(302°F), then 32.9 molecular weight (112.38 MM Btu/h) will be made available for exchange. Exactly how much equivalent fuel this saves depends on how the heat is used. A reasonably optimistic scheme would perhaps save the equivalent of 32.9 molecular weight (112.38 MM Btu/h) in fuel in some other equipment, say boiler feed water preheat, making the equivalent fuel consumption for this scheme 19.3 molecular weight (65.7 MM Btu/h) as is shown in Figure 39.

Figure 40 shows a simpler, much more easily controlled and self-contained scheme, which would again use a gas turbine, but the hot excess oxygen leaving the turbine would be used as combustion air for a conventional heater. The gas turbine could be based loaded to a point that would ensure adequate oxygen availability for the auxiliary fuel to freely swing to maintain process control. This scheme could also more easily

accommodate the condition of continuing operation even when the turbines were down for maintenance.

The economy of the re-fired gas turbine scheme could also be improved by lowering the flue gas temperature to 150°C (302°F) Figure 41. If the equivalent fuel saved by cooling the fuel gases is 3.3 molecular weight (11.3 MM Btu/h), then the net fuel consumption in this scheme is 31.1 molecular weight (106 MM Btu/h). A comparison of these schemes is presented in Table 3. The performance of the gas turbine was estimated using a heat rate of 5 kW fuel per kW work (12727 Btu per hph). Turbines with superior heat rates are available, and would improve the overall system efficiencies shown.

Perhaps the major factor in the economics of energy conservation is the rate of return on capital or the "payout" time acceptable. The cost of co-generation, therefore, will depend upon the duty desired and the type of scheme chosen, recognizing that the size of the system could affect the economics significantly. It is not difficult to envision that the actual routes used in any kind of large scale co-generation scheme could involve multi-company and/or multi-industry integrations, akin to the so-called "energy parks".

An energy park might consist of an electric generating company that would generate very high pressure steam, expand it through turbines to some medium pressure levels and distribute the heat to other industries. The refinery could export lower temperature heat to other industries, or back to the electric generating plant to preheat condensate or combustion air. Such a major integration is without precedent, and the associated problems could be so large that a completely integrated industrial park may never be realized. The potential energy savings can be great, however, and the conservation advantages are widely recognized. The industry should be aware of and plan for this possible eventuality.

FCC Power Recovery--Topping Cycle Co-generation

If the status of co-generation as a viable approach seems rather hazy at this time, there is at least one associated application that already has been quite successful, power recovery units on Fluid Catalytic Cracking units. The FCC regenerator can be thought of as the front half of a gas turbine. Air is compressed and used to burn off the coke deposited on the catalyst. The complete combustion of the coke raises the flue gas temperature well over 650°C (1200°F) and the hot gas is discharged from the regenerator.

long period of time.

A number of techniques and strategies have been presented that enable the designer to arrive at an economical processing scheme. The plant based on the most efficient scheme on paper, however, once built and operating, must be continually monitored, maintained and modified to actually achieve the best economy in real operations over a

OPERATE TO CONSERVE ENERGY

Power recovery systems are particularly well suited to grass roots installations or major revamps, but also hold great potential for existing plants. As always, the decision to employ any power recovery scheme must be based on a thorough analysis for the specific installation. Feedstock, yield, plot area, and the economies of size are all important parameters. Doubtless the primary concern is process operability, i.e., a FCC unit is meant to process oil, not generate power, although the successful operation of many units over a period of years has proved the viability of this power recovery technique. Economic pressure for energy conservation plus technological advances combine to cause designers and processors to consider for their projects the use of one proven power recovery turbine and similar energy saving systems.

Table 4 compares the work produced by the regenerator flue gas in this scheme with work produced in a conventional system employing only heat recovery, at various steam pressure levels. It assumes that the steam generated in the heat recovery system is used to drive condensing turbines at an exhaust pressure of 10 kPa, absolute--46°C(3 inches of Hg--(115°F), and that the steam turbines are 75% efficient.

The flue gas is at such a high temperature leaving the regenerator that the recovery of this heat, usually with steam generation, is almost universally practiced. The steam generated is then used to drive rotating equipment such as the air blower and the wet gas compressor. The amount of work that can be produced depends upon the pressure and temperature of the steam. The thermodynamic efficiency of the process is greatly improved by employing a hot gas expander ahead of the heat recovery equipment. The expander produces sufficient shaft work to drive the air blower thus completing the gas turbine cycle. This application of a heat engine in a topping cycle is a good example of economically applied co-generation. Figure 42 is a schematic drawing of a typical installation.

Only so much can be anticipated at the design stage, and the older a plant gets the more likely it is that it will be operating farther from the original design basis. This is true for individual items of equipment and whole processing schemes.

Heater Firing and Insulation, The First Steps

The designer can trim fired-heater duty to the bone, but complete and efficient combustion must go along with improved design for real savings to occur. The single most direct conservation step can be the monitoring and control of fuel firing in furnaces and boilers together with scheduled cleaning and maintenance of burners and tubes.

Another simple technique to improve fuel efficiency is the addition of insulation to piping, vessels and tankage. It is safe to assume that the plants constructed during the era of very cheap fuel were not adequately insulated on top of being generally wasteful in the use of energy.

Monitor Heat Exchangers

Even well-designed heat exchanger systems meet fouling problems with time, decreasing their performance and increasing operating costs. Monitoring the performance of heat exchange equipment will enable proper maintenance to be scheduled that will ensure high performance.

Vacuum Jets, Cheap Conservation

When a vacuum distillation column is designed and equipment specified, the designer cannot know precisely what the load on the steam jet will actually be. Equipment is selected that will ensure good performance over wide operating ranges, and the capability of operating the jets at full load, 2/3 or 1/3, is built into the unit. It is expected that the operator will monitor performance and adjust the steam jets accordingly for maximum economy. Good maintenance of equipment to prevent air leaks is also a routine, cheap way to conserve.

Fractionation Requires Special Attention

A similar situation exists with fractionating columns. The designer specifies equipment that will perform well over a wide range of operating conditions. So that this equipment

can be operated most efficiently, control schemes are provided that enable the operator to minimize actual utility costs by minimizing product quality giveaway.

In older plants especially, the actual composition of the streams processed may differ significantly from the original design basis so that a different feed tray location may be desirable. Its exact location can be fixed through process design calculations and design drawings made. This can be a simple and economically rewarding step, reducing the quantity of reflux required or increasing column throughput for the same reboiler duty.

Steam Systems

The one area in processing plants, that always has been designed using principles of thermodynamic efficiency as the main guide, is the steam system. As plants are modified by expansions and revamps, however, the steam system is often neglected, with changes and additions being made on an immediate, "as needed" basis, without enough attention paid to the system itself.

With more emphasis on energy and water conservation it is important that offsite systems receive the same attention as the process units. This does not mean that the crude unit must be operated for the benefit of the steam system, but it does mean that processing conditions should be examined with an eye to total refinery economy. Items in this area include monitoring and reviewing stripping steam rates and pressure levels, and re-examining the critical drive philosophy and operating driver choice. It can extend to even switching some column reboilers from medium to low-pressure steam which may be feasible now when it was not in the past. But, again, examining the operation from a total refinery basis is the main point.

SUMMARY

Energy conservation in the process industries is an extremely complex and broadly based issue. The increased cost of fuel has prompted, at first, a multitude of "housekeeping" procedures; more insulation, improved heater firing efficiency, and more attention given to processing variables such as recycle stream circulation and column reflux. As operating costs continue to increase, conventional schemes are modified by larger capital expenditures; air preheaters are added, heat pumps installed, waste heat is recovered and distributed by steam or circulating fluids, fractionating columns are heat integrated by

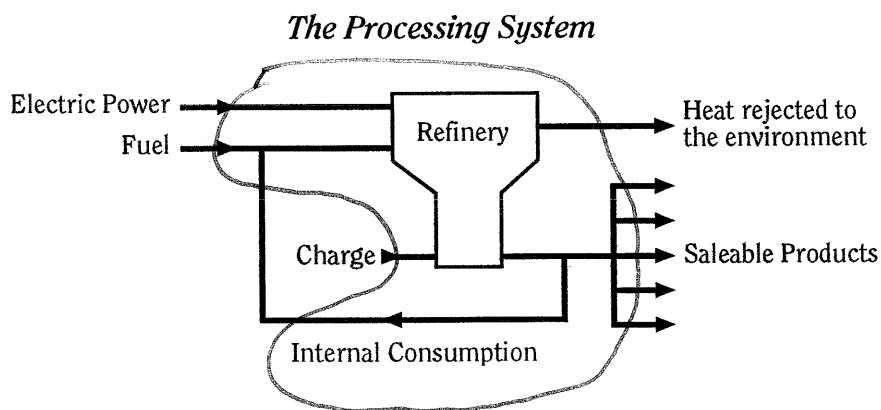
the principle of multi-effect distillation, shaft work is produced from process streams whenever possible by the installation of expansion turbines and pumps.

As economic or regulatory constraints force the industry to greater conservation efforts, new methods must be found and old methods modified, to increase the thermodynamic efficiency of processing plants. If the potential of co-generation is to be fully realized, the industry must prepare for its implementation through proper planning.

No matter what schemes are used, more emphasis must be placed on sound operating practice. Continuous monitoring and control, better management, and maintenance will become even more important than they are today. The basis of all decisions must be economic if real savings are to be realized. Economic does not necessarily mean cheapest equipment or longest catalyst life, neither does it mean lowest energy consumption. It implies that the required products can be made safely, reliably and with minimum resource use of the entire system.

The hydrocarbon processing industry is facing a great challenge. Quite possibly fuel prices may rise faster than other costs, which will make planning very difficult. More regulations will be imposed that are meant to anticipate economic changes and force conservation. New raw materials, such as coal or shale oil, may become common. But, if the industry can continue its excellent record of sound judgment and innovation, the challenge can be met.

Figure 19

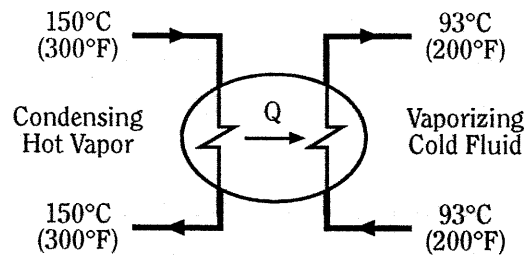


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Figure 20

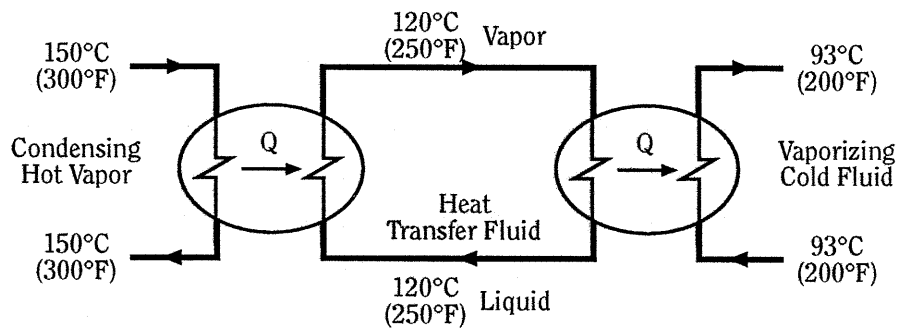
Integration for Heat Recovery

Direct Integration:



$$\text{Area} = \frac{Q}{U\Delta T}$$

Integration via a Heat Transfer Fluid:



$$\text{Area} = \frac{Q}{U \frac{\Delta T}{2}} + \frac{Q}{U \frac{\Delta T}{2}} = 4 \frac{Q}{U\Delta T}$$

EDS-R00-1242

Series Integration Without Heat Exchange

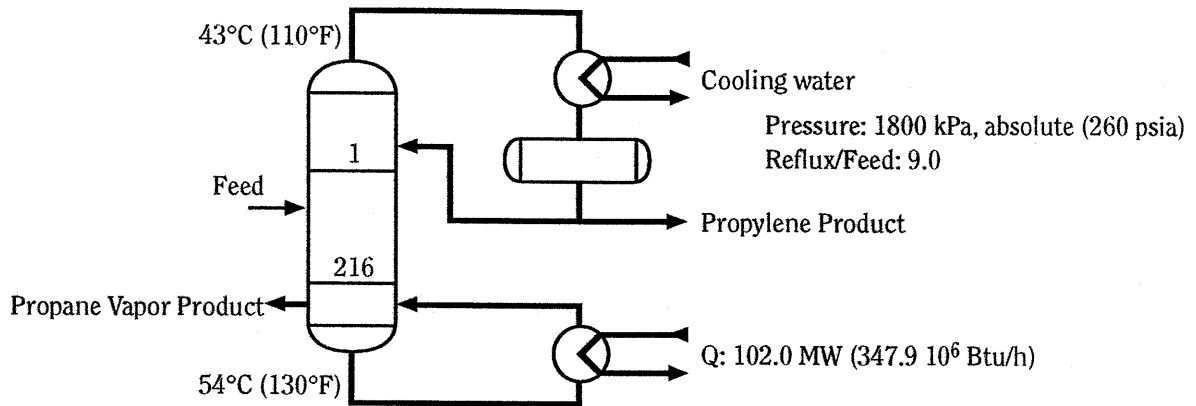


Series Integration With Heat Exchange



Figure 23

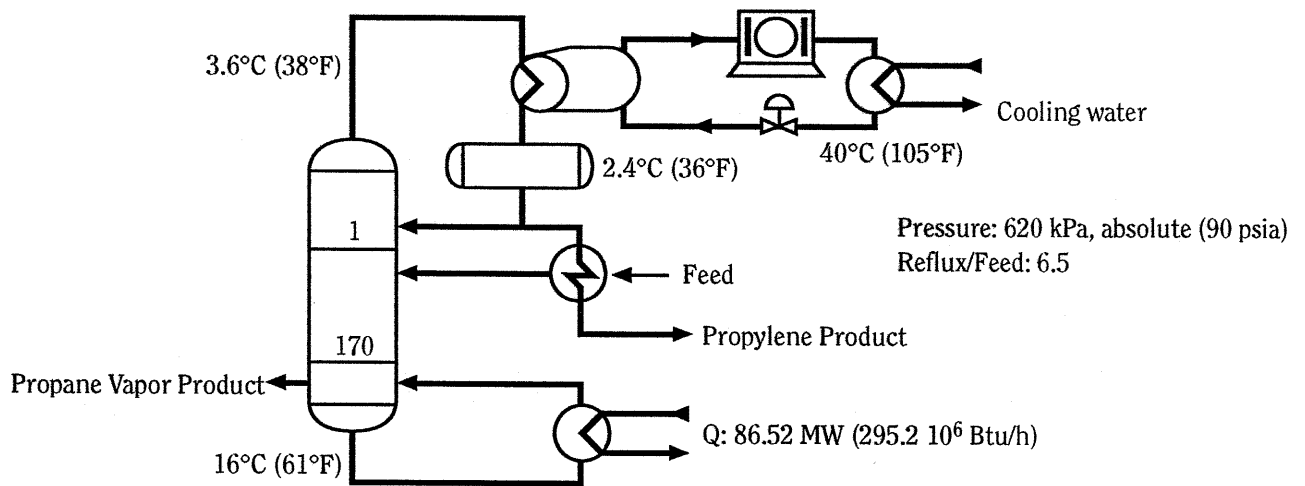
*Propylene - Propane Separation
Conventional Fractionation System*



EDS-R00-1245

Figure 24

*Propylene - Propane Separation
Refrigerated Condenser*



Work of compression: 18,400 kW (24,700 hp)

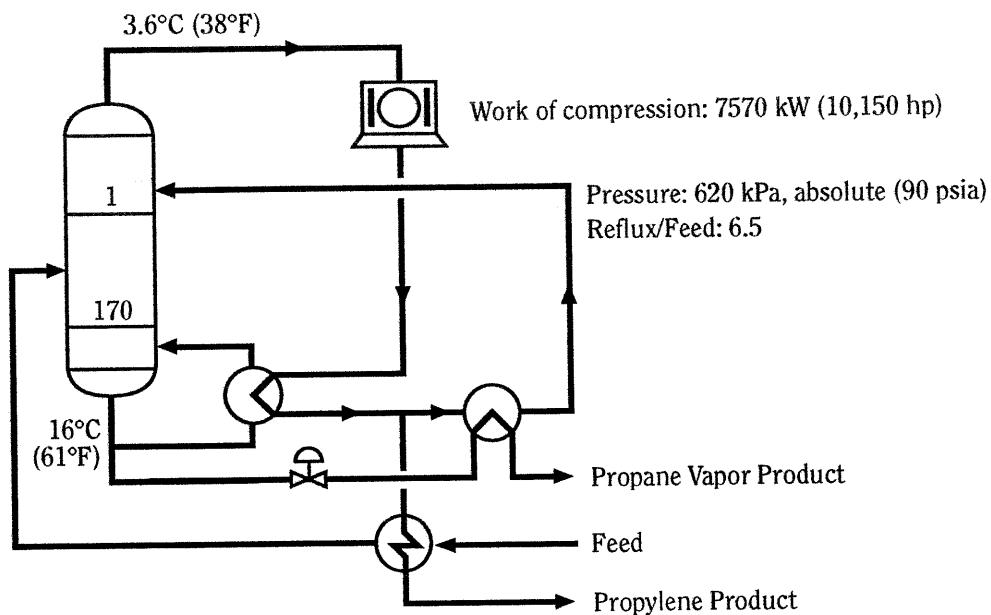
EDS-R00-1246

$$18400 \text{ kW} \times \frac{10000 \text{ Btu/hr}}{\text{kW-hr}} = 184 \text{ million Btu/hr}$$

$$\frac{295}{479}$$

Figure 25

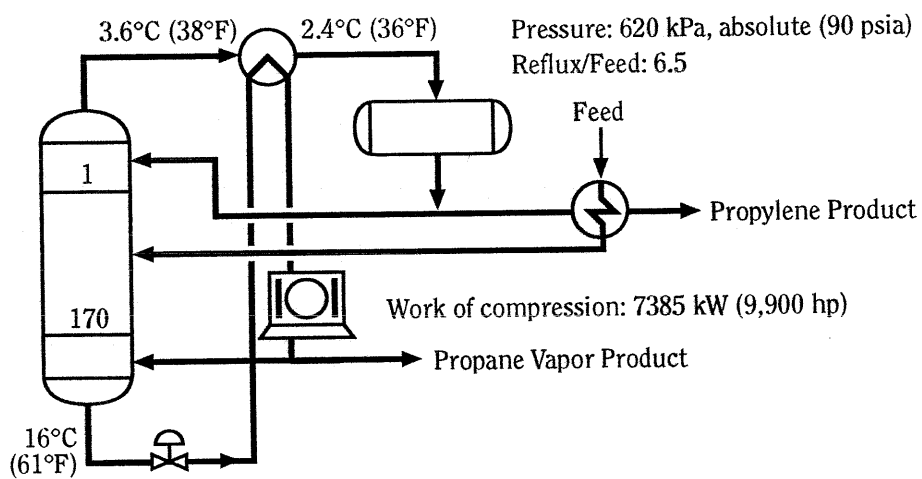
*Propylene - Propane Separation
Classical Heat Pump*



EDS-R00-1247

Figure 26

*Propylene - Propane Separation
"Reverse" Heat Pump*

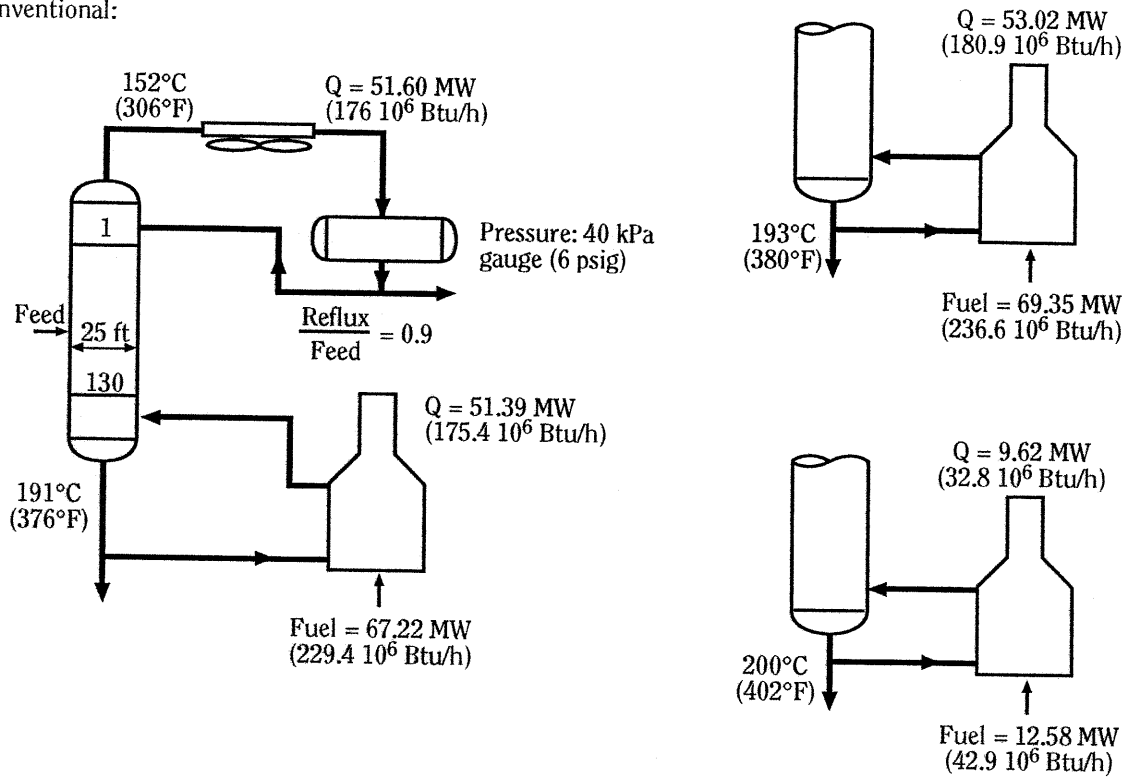


EDS-R00-1248

Figure 27

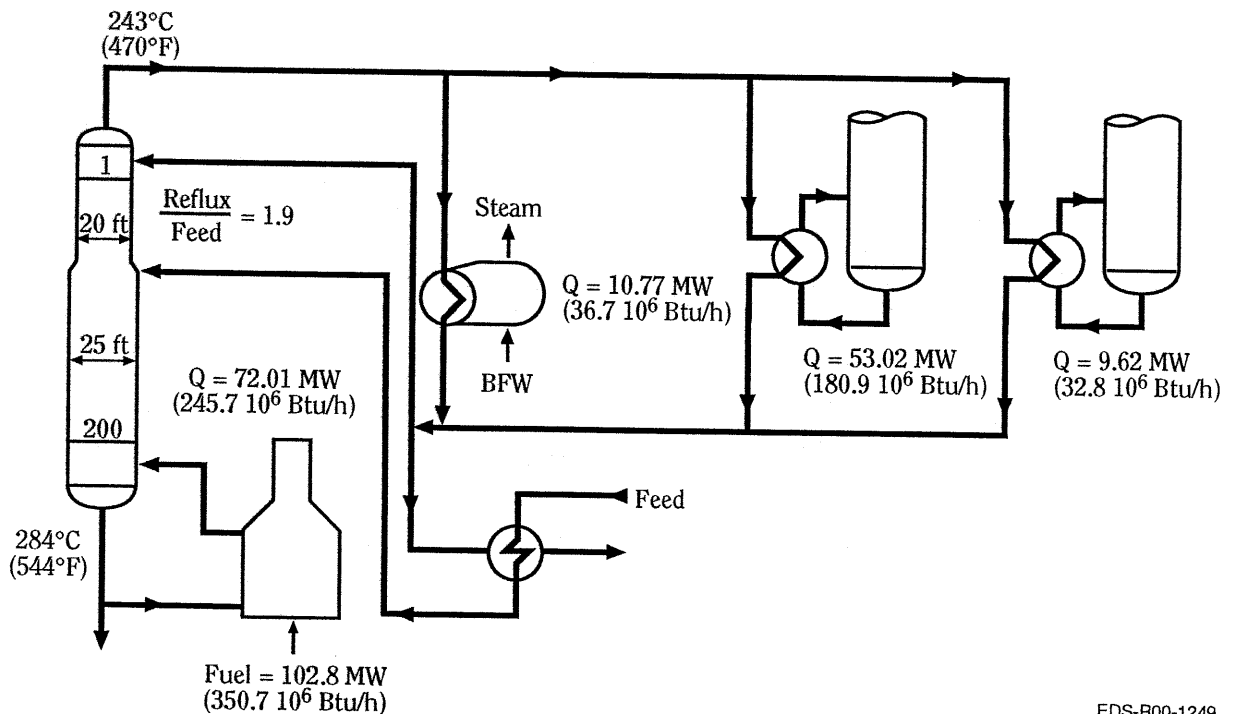
Fractionation Column Integration

Conventional:



Heat Cascading:
 Pressure: 790 kPa gauge (115 psig)

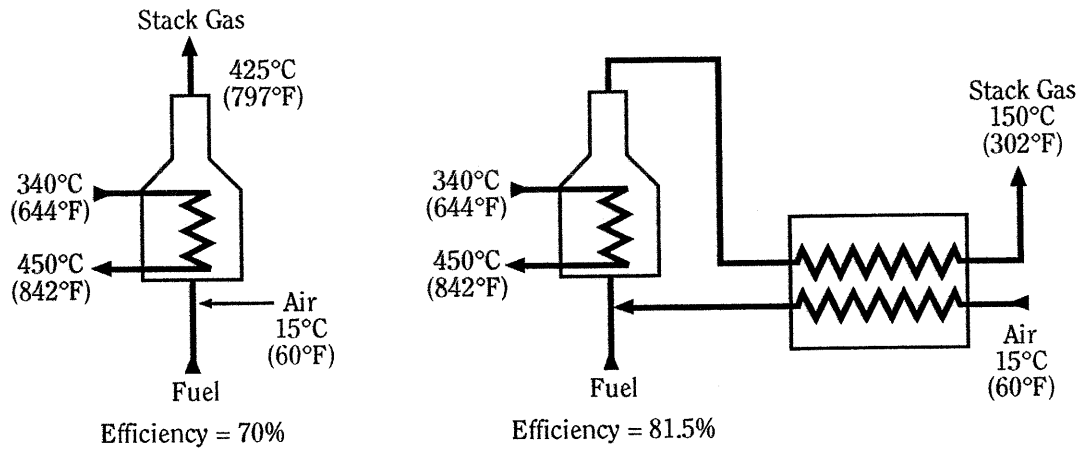
Xylene Fractionator



EDS-R00-1249

Figure 28

Effect of Air Preheating on Heater Efficiency



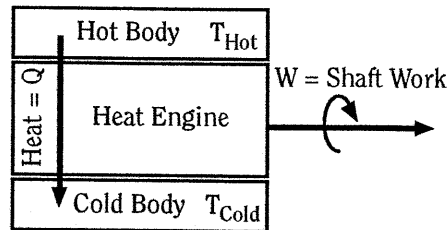
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$$\eta = \frac{\text{Heat abs.}}{\text{Fuel Fired}}$$

Flue gas : 40°F / 1 %

ADVANCED (SECOND LAW) APPROACH TO ENERGY SYSTEMS ANALYSIS

Figure 29



EDS-R00-1214

Second Law Of Thermodynamics:

In converting heat to work, some of the heat must be rejected.

Conversely:

Where heat transfer occurs, there is an opportunity to do work.

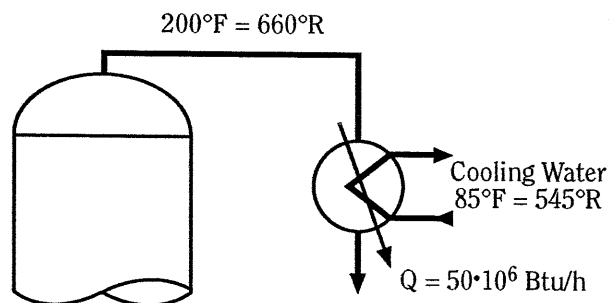
$$W = Q \times \left(\frac{T_{hot} - T_{cold}}{T_{hot}} \right)$$

$W = \text{Theoretical Max. Work Recoverable}$

$Q = \text{Heat Transferred}$

$$\left(\frac{T_{hot} - T_{cold}}{T_{hot}} \right) = \text{Thermodynamic Efficiency}$$

Figure 30



EDS-R00-1215

Maximum Work Recoverable:

$$W = Q \left(\frac{T_{HOT} - T_{COLD}}{T_{HOT}} \right)$$

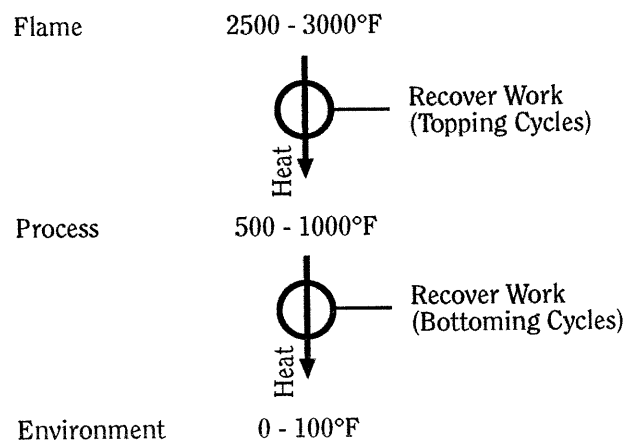
17.4 %

$$W = 50 \times 10^6 \frac{Btu}{h} \left(\frac{660 - 545}{660} \right) \frac{1kWh}{3414 Btu}$$
$$W = 2552 kW = \$1.1 \text{ Million/a!}$$

COGENERATION

Producing Work As a Byproduct In The Transfer Of Heat

Figure 31

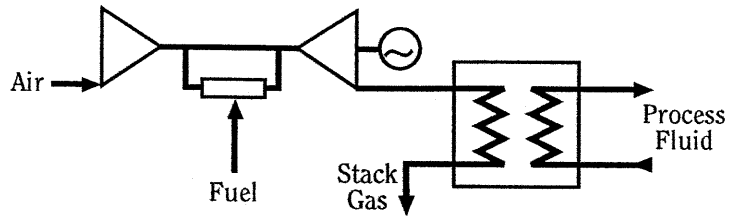


EDS-R00-1216

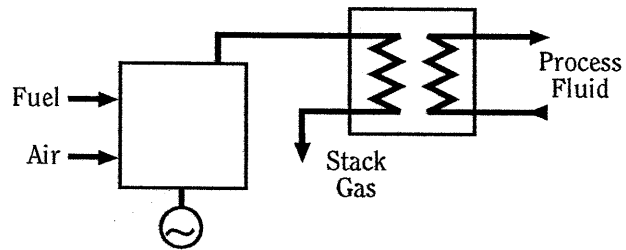
Figure 32

Co-Generation Topping Cycle

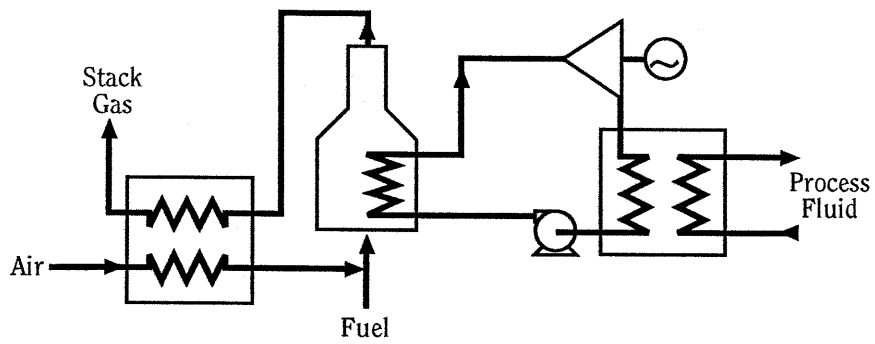
Gas Turbine



Diesel Engine



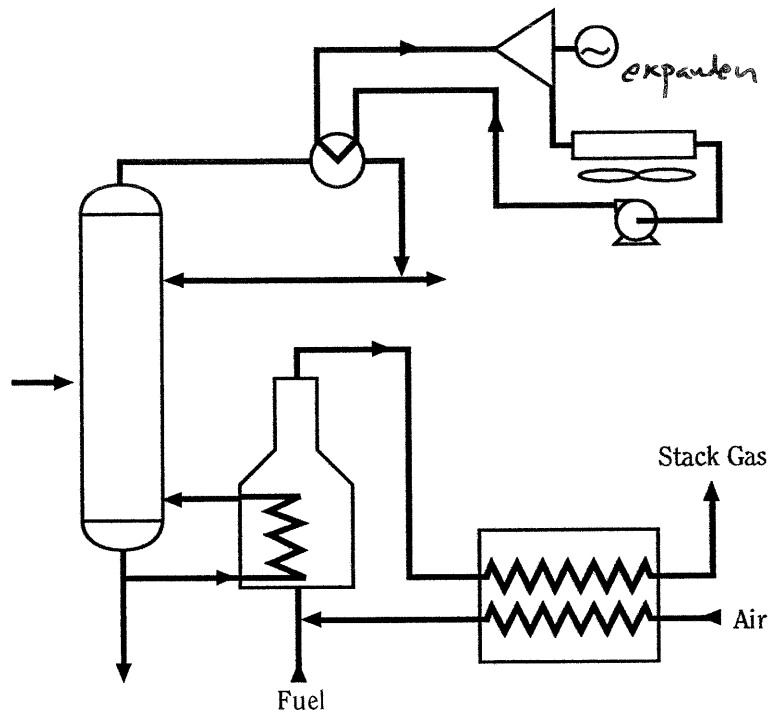
Steam Turbine



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Figure 33

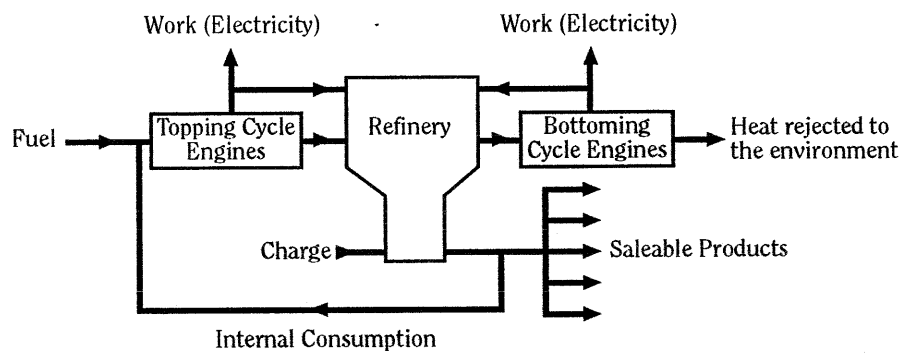
Co-Generation Bottoming Cycle



EDS-R00-1252

Figure 34

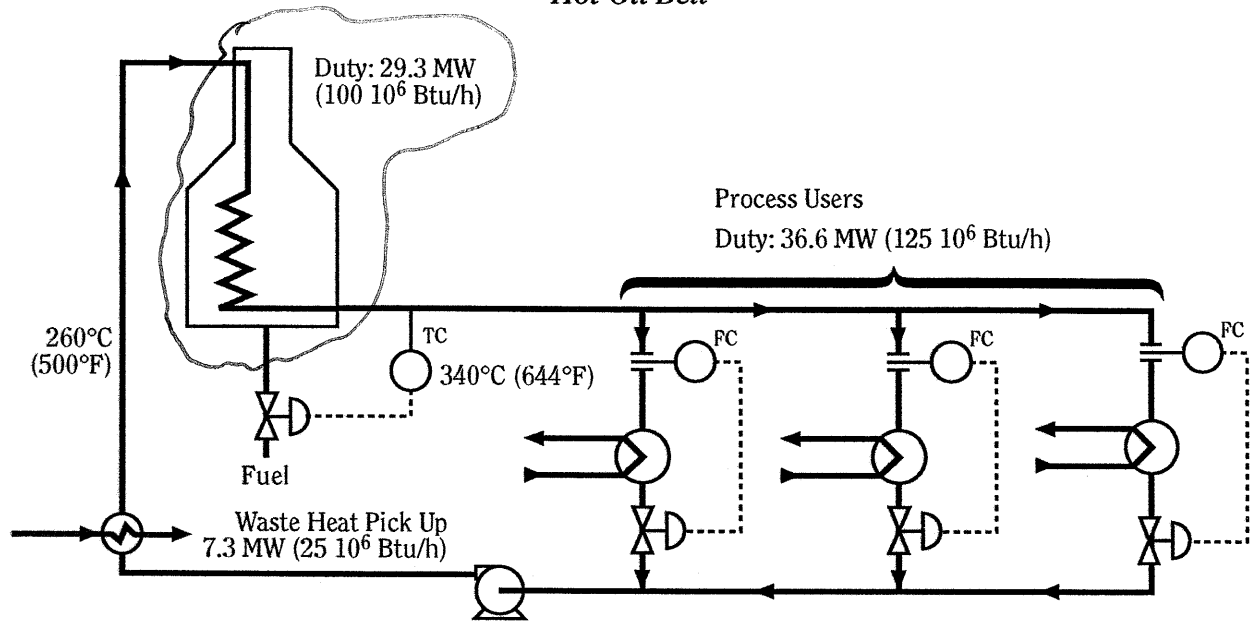
The Processing System with Co-Generation



EDS-R00-1253

Figure 35

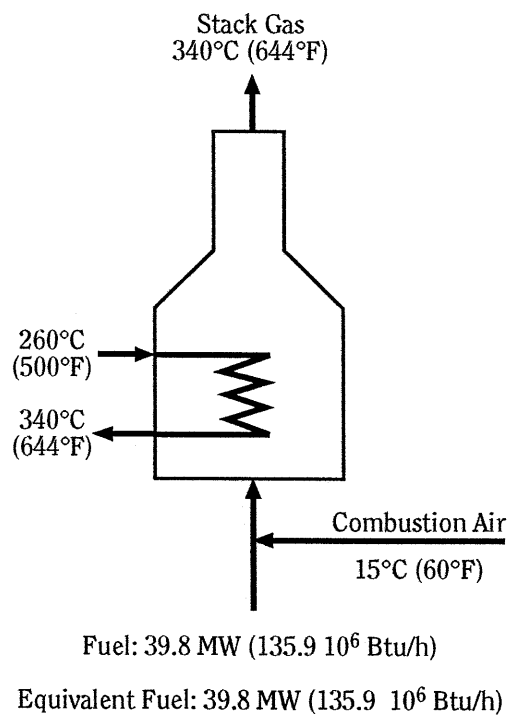
Hot Oil Belt



EDS-R00-1254

Figure 36

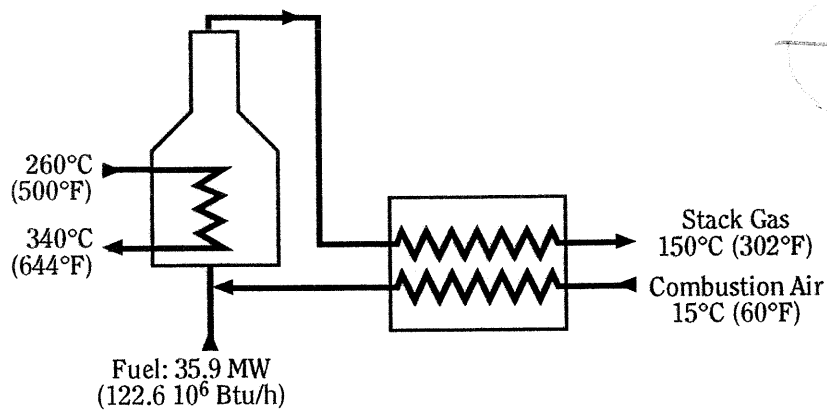
Conventional Heater



EDS-R00-1255

Figure 37

Fired Heater with Air Preheater



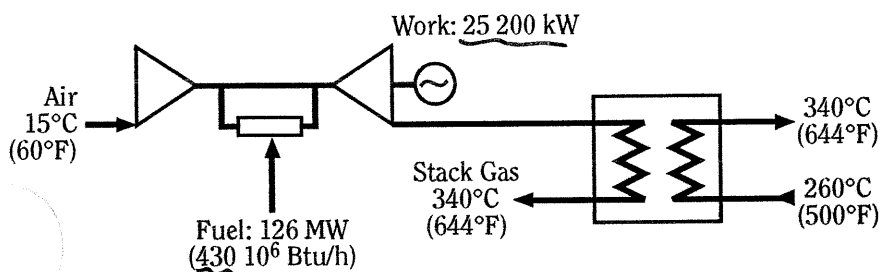
Equivalent Fuel: 35.9 MW (122.6 10^6 Btu/h)

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19 # AIR
FUEL

Figure 38

Simple Cycle Gas Turbine



Equivalent Fuel: 52.2 MW (178 10^6 Btu/h)

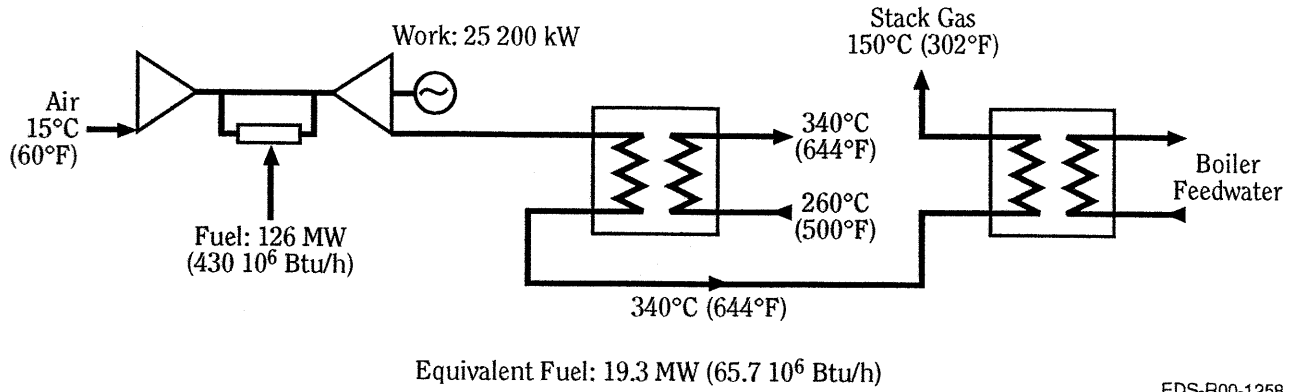
EDS-R00-1257

70 # AIR
FUEL

430
- 252
178

Figure 39

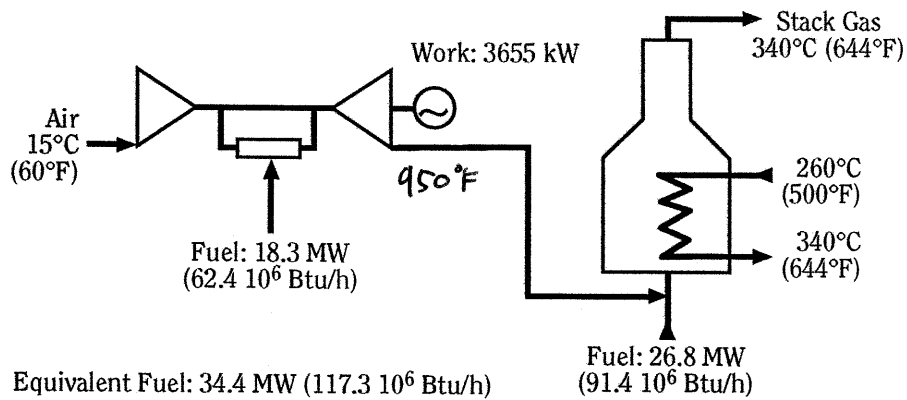
Simple Cycle Gas Turbine with Auxiliary Heat Recovery



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Figure 40

Refired Gas Turbine



EDS-R00-1259

$$\eta = \frac{\text{output}}{\text{input}} =$$

Refired Gas Turbine with Auxiliary Heat Recovery

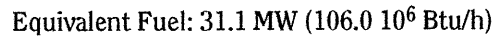


Figure 42

The diagram illustrates the process flow of a fluid coking unit. Key components and parameters include:

- Air Feed:** 151,800 kg/h (334,700 lb/h) enters from the left.
- Coke Rate:** 10900 kg/h (24000 lb/h).
- Gas Oil Feed:** 215 m³/h (32500 bpd).
- Electricity:** Net Export 512 kW.
- Temperatures:** 730°C (1350°F), 255 kPa Absolute (37 psia), 704°C (1300°F), 544°C (1012°F).
- Flow Streams:** Quench, Separator output (158,800 kg/h / 350,000 lb/h), Steam Generation, Stack Gas.

Page 56

Table 1
Comparison of Propylene - Propane
Separation Schemes

	<u>Conventional</u>	<u>Classical Heat Pump</u>	<u>Reverse Heat Pump</u>
ENERGY INPUT:	102 MW (347.9 MM Btu/h) At 54°C (130°F)	7570 kW (10 150 hp)	7385 kW (9900 hp)

Table 2
Economic Advantages of Fractionator Heating Cascading

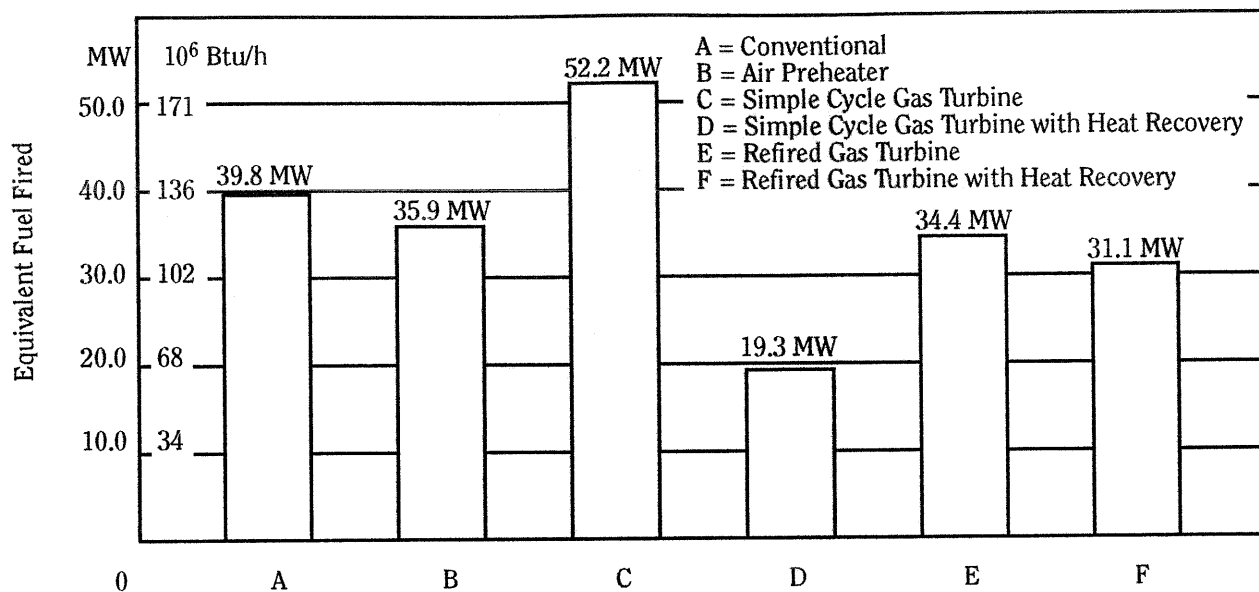
		<u>Conventional Case</u>	<u>Heat Cascade Case</u>
Total Fuel Fired	MW (10 ⁶ Btu /h)	149.15 (508.9)	102.80 (350.7)
Steam Generated	Kg/h (lb/h)		15100 (33400)
Fuel Saved in One 8000 h/a	Peta Joules (10 ¹² Btu)		1.33 (1.26)
Dollar Savings Due To Direct Reduction in Fuel Use	MM Per Year		2.66
Dollar Credit For Steam Generation	10 ⁶ /a		0.76

Total Dollar Savings of Heat Cascade
Integration Over Conventional Scheme: \$3.42 x 10⁶/a

Estimated Erected Cost Differential
Gulf Coast 1976 \$4.2 x 10⁶/a

Table 3

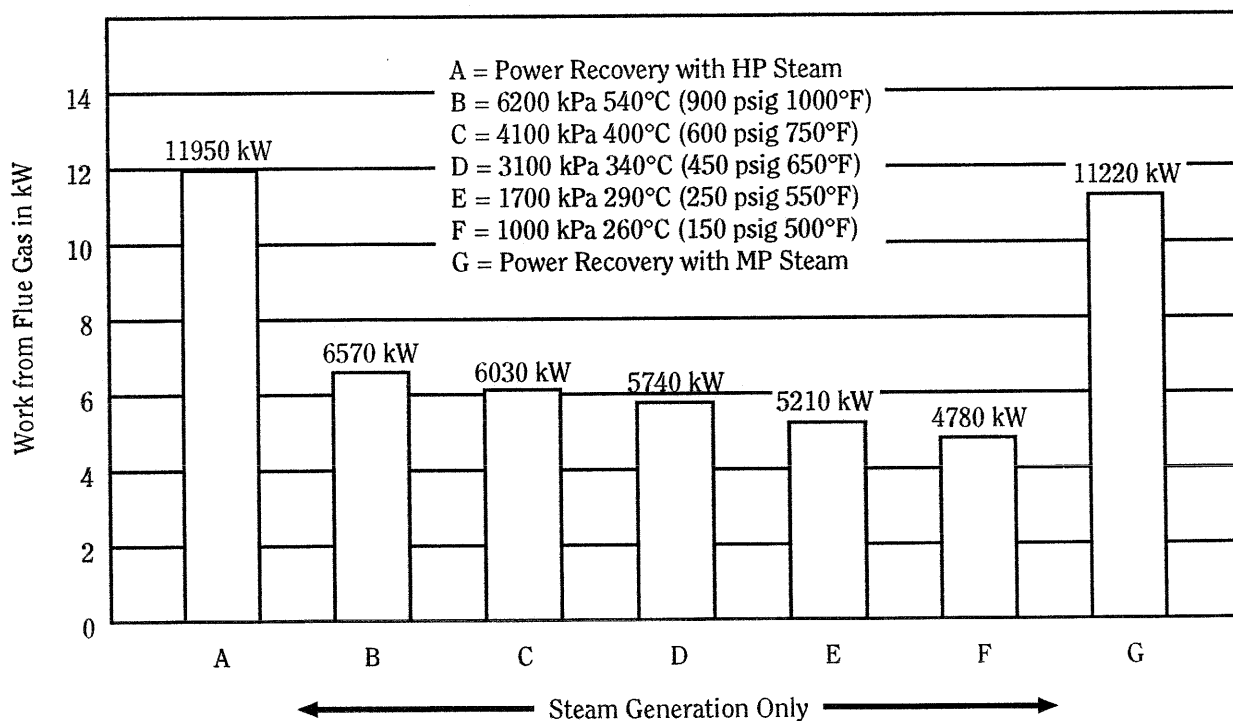
Equivalent Fuel Fired to Supply Hot Oil System Duty



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Table 4

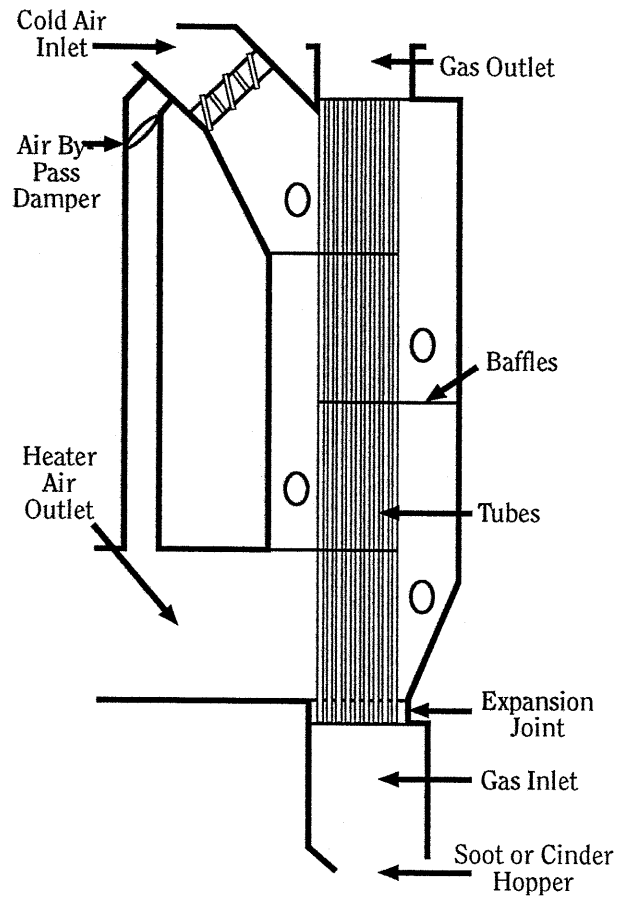
Power Recovered from FCC Flue Gas



EDS-R00-1263

Figure 43

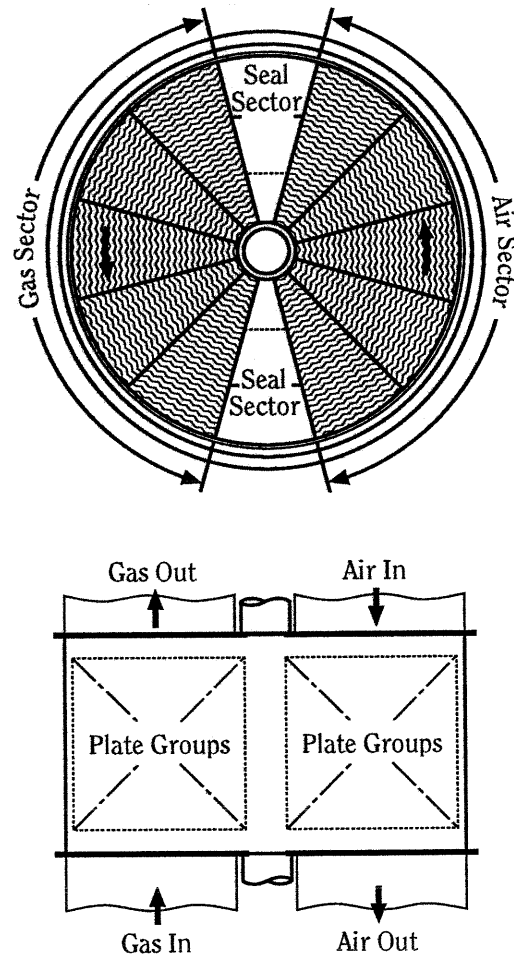
Tubular air heater arranged for counterflow of gas and air, with an air bypass to control metal temperature at air inlet end.



EDS-R00-1217

Figure 44

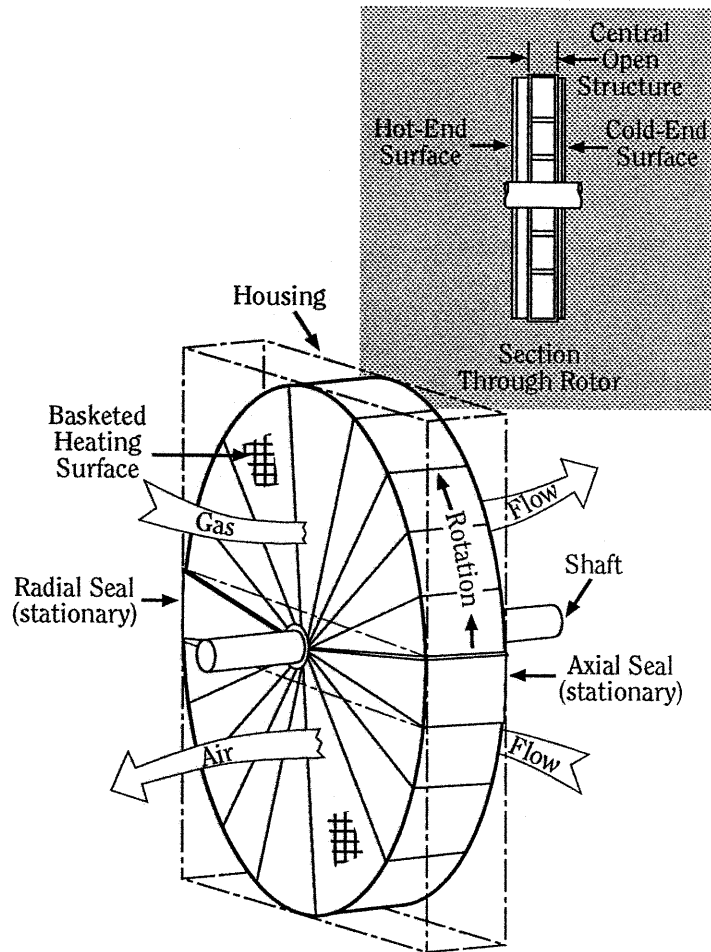
Diagrammatic illustration of rotary regenerative air heater (vertical shaft arrangement) with gas and air counterflow.



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Figure 45

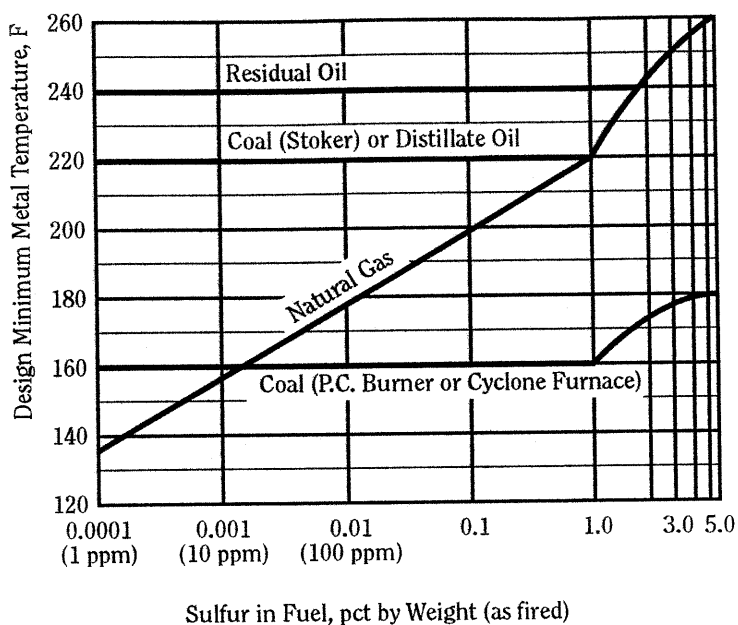
Element of a rotary regenerative air heater.



EDS-R00-1219

Figure 46

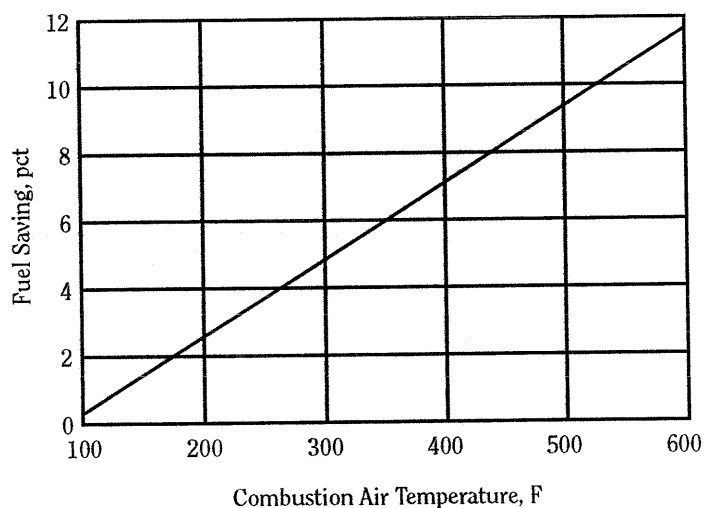
Limiting tube-metal temperatures to avoid external corrosion in economizers or air heaters when burning fuels containing sulfur.



EDS-R00-1264

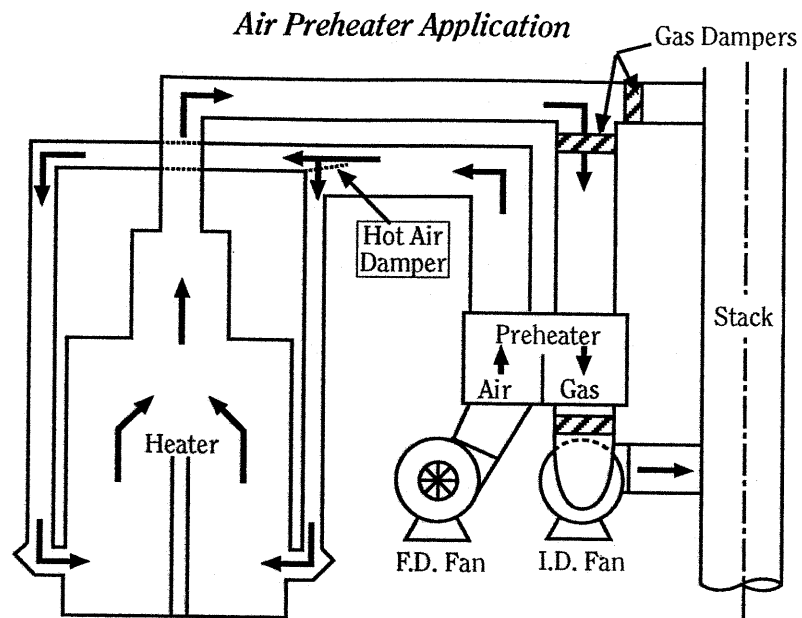
Figure 47

Approximate improvement in efficiency when heated combustion air is used in boiler units.



EDS-R00-1265

Figure 48

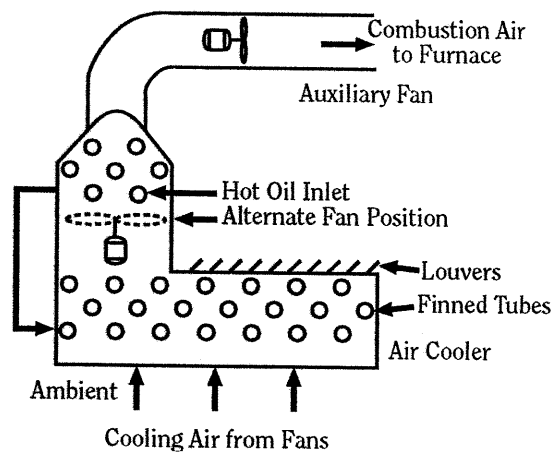


F.D. = forced draft. I.D. = induced draft.

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Figure 49

Combustion air preheat from air coolers.

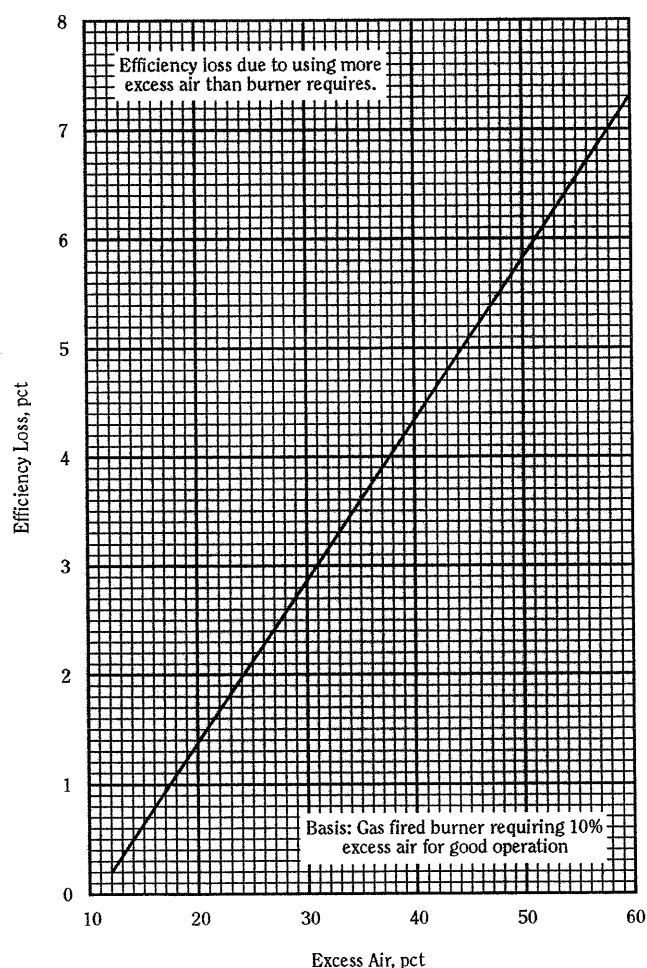


EDS-R00-1221

Comparison of Furnace Efficiency With And Without An Air Preheating System

Operating Factor	Without Air Preheat System	With Air Preheat System
Absorbed duty, million Btu/h	83	83
Ambient air (temperature)	80°F, 26.7°C	80°F, 26.7°C
Air to furnace (temperature)	80°F, 26.7°C	660°F, 349°C
Gas to air heater (temperature)		815°F, 435°C
Gas leaving air heater (temperature)		319°F, ≈160°C
Gas to stack (temperature)	815°F, 435°C	291°F, ≈ 143°C
Excess air, percent	30	15
Furnace efficiency, percent	75.7 (lower heating value)	90.2 (lower heating value)
Million Btu fired	109.6	92.0

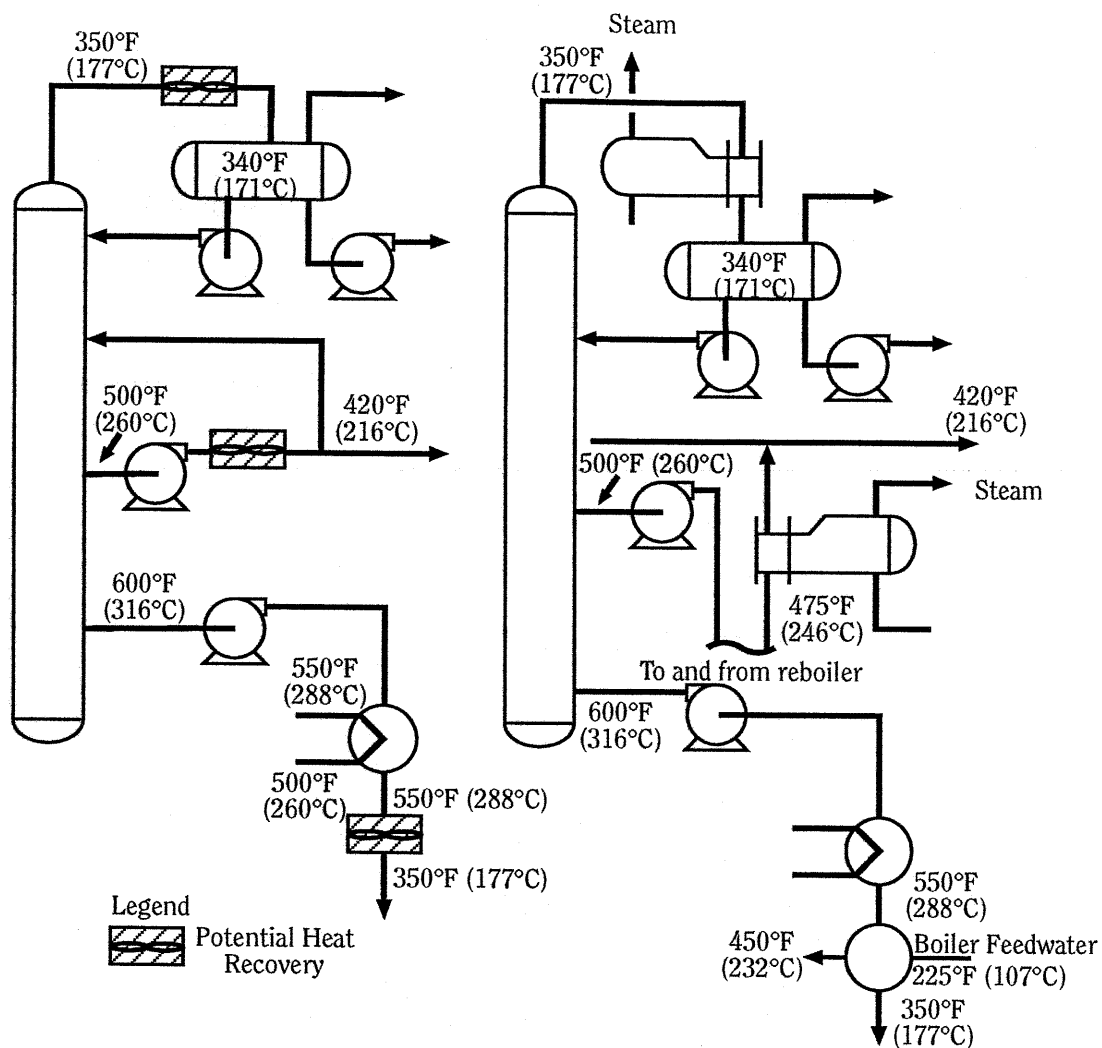
Figure 50



EDS-R00-1268

Figure 51

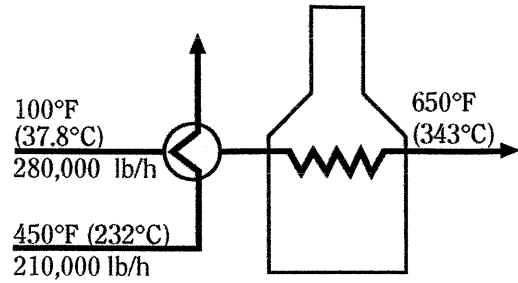
Maximizing steam generation from process heat.



EDS-R00-1222

Figure 52

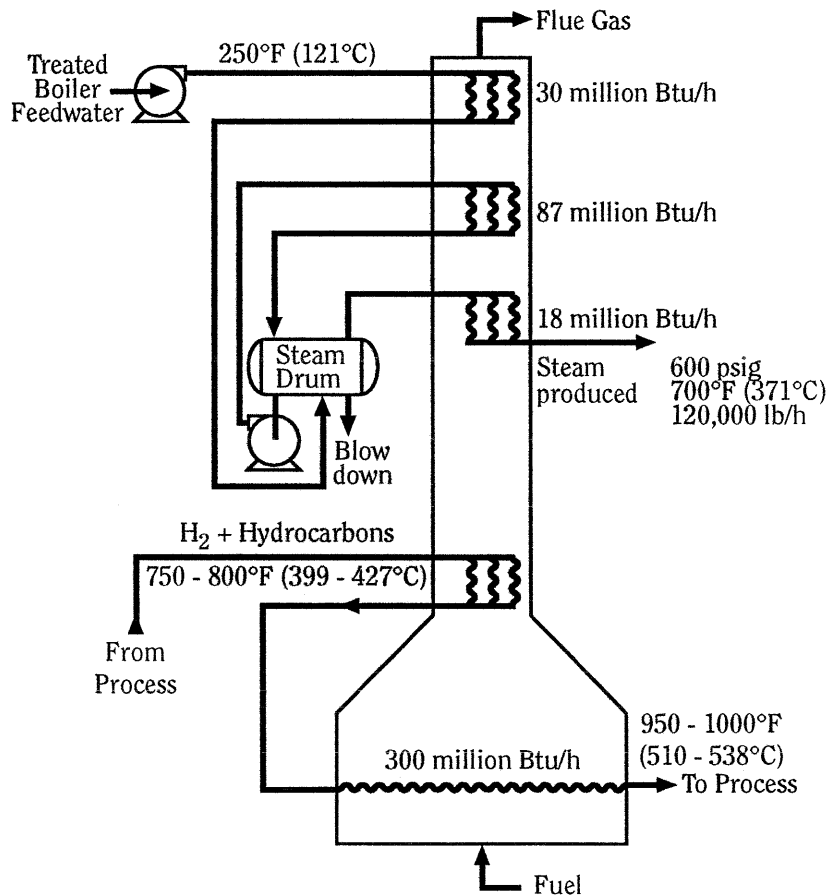
Heat Exchanger Optimization



EDS-R00-1223

Figure 53

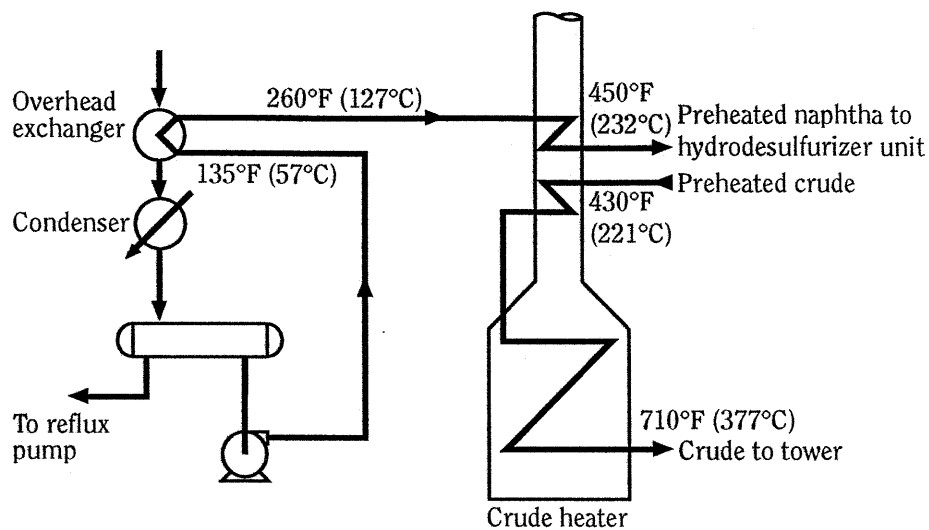
Heat recovery in a catalytic reformer heater.



EDS-R00-1224

Figure 54

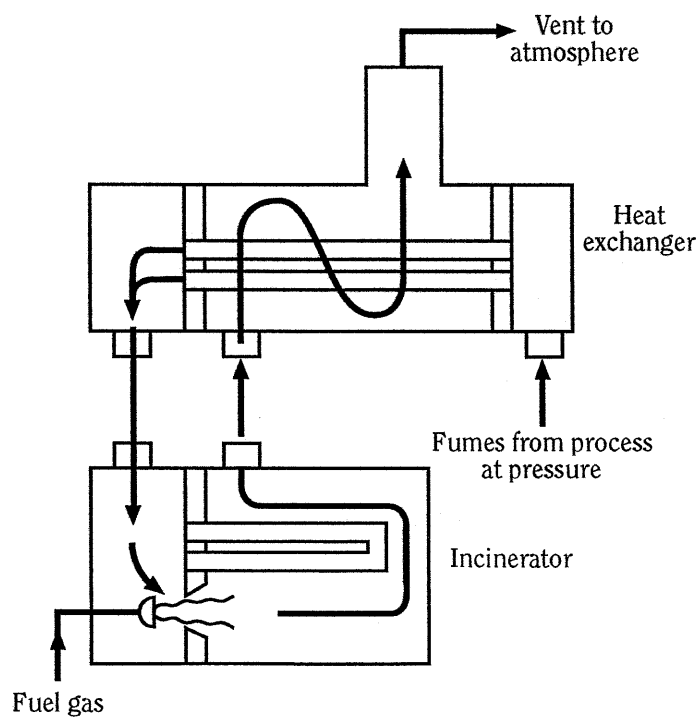
Process heat recovery in a heater



EDS-R00-1225

Figure 55

Heat recovery in a fume incinerator.



EDS-R00-1226