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Date May 28, 1986

ENERGY SAVINGS IN DISTILLATION TRAINS THROUGH THE APPLICATION OF HEAT PUMP SYSTEMS

A Thesis

Presented for the

Master of Science

Degree

The University of Tennessee, Knoxville

Eric W. Forssell

June 1986

DEDICATION

This thesis is dedicated to Mr. Alan G. Forssell whose tireless support and endless patience made this project feasible. It is also dedicated to the memory of Mrs. Gertrude M. Forssell whose scholarly achievements while at Syracuse University and while teaching in the Baldwinsville, N.Y., school system were an inspiration to all she met.

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Several scholars have assisted in this project. Most notably is the assistance provided by my adviser, Dr. John Holmes, and my Master's committee which was headed by my adviser and consisted of Dr. Carl Thomas and Dr. William Johnson. I also wish to acknowledge the assistance of Dr. Duane Bruns who served temporarily on my Master's committee and Mr. C. Crawford, Manager of Power and Engineering Information for the Tennessee Valley Authority, who provided the necessary information on their load forecasts. Without this assistance this project would have been impossible.

ABSTRACT

The purpose of this study is to evaluate the application of heat pump systems to a distillation train. A heat pump removes energy from a low temperature source and provides energy at a higher temperature through the addition of compressor work. A Rankine cycle heat pump consists of an evaporator, condenser, throttling device, trim-cooler, innercooler, and a working fluid.

The economic feasibility of a heat pump system is affected by five major variables. The heat pump system is favored by high reboiler heat duties as the benefits grow faster than the costs of the heat pump system. The heat pump system is favored by heat duty ratios, condenser heat duty to reboiler heat duty, near the no trim-exchanger point. Increases in condenser temperature bring a slight reduction in the required utility ratios above the no trim-exchanger heat duty ratio point and a slight increase below it. The heat pump system is favored by low temperature differences.

The utility ratio (natural gas to electricity on a \$/million BTU basis) was used as a measure of economic performance. A required utility ratio was determined and compared to the actual utility ratio. If the actual utility ratio in effect or projected is higher then the heat pump is favored.

For a specific case analysis, an example column was chosen and several options were considered. These options were compared to a conventional reboiler and condenser system on the basis of the additional

investment required and the return on that investment. The example column was an extractive distillation column used in an ethanol water separation distillation train.

Of the options considered, the use of the column overhead fluid as the heat pump working fluid was the most promising. It resulted in a favorable return on investment of 44 percent. The use of a steam driven compressor showed promise, but failed to achieve a favorable return on investment.

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NOMENCLATURE

A	- heat transfer area, ft^2.
A	- first Antoine equation constant, $\ln(P_v) = A + B/(C + T)$.
A_{ex}	- outside heat transfer area of the exchanger, ft^2.
Af	- heat transfer area of a fin, ft^2.
A_i	- heat transfer area inside tubes, ft^2.
a_o	- coefficient in Nusselt analogy for shell side heat transfer coefficient, equal to 0.33 for staggered tube bundles and 0.26 for in line bundles.
A_o	- heat transfer area outside tubes, ft^2.
A_s	- parameter in Soave-Redlich-Kwong equation of state, $P = RT/(v-B_s) - A_sA_z/(v + B_s)$.
A_z	- Soave-Redlich-Kwong parameter.
A_1	- first parameter in Euler form of the ideal gas heat capacity equation, $C_p = A_1 + B_1{*}T + C_1{*}T^2 + D_1{*}T^3$, with C_p in Cal/gmole °K, T in degrees Kelvin, and A_1, B_1, C_1, and D_1 in appropriate units.
A_2	- first parameter in the Andrae equation for liquid viscosity $u_L = A_2\exp(B_2/T)$, with u_L in centipoise, T in degrees Kelvin, and A_2 and B_2 are constants with appropriate units.
B	- second parameter in the Antoine equation.

B_i	- the correction factor for expansion, contraction and flow reversal used in determining the pressure drop inside tubes with single phase flow.
b_o	- parameter relating the Reynolds number to the modified friction factor for the shell side.
B_o	- correction factor for reversal of flow, recrossing of tubes and variations in cross sectional area, equal to the number of baffles plus one.
B_s	- Soave-Redlich-Kwong equation parameter.
B_1	- second parameter in the Euler form of the ideal gas heat capacity equation.
B_2	- second parameter in the Andrae equation.
c	- speed of sound for a two phase mixture, ft/s.
C	- third parameter in the Antoine equation.
C_{Ao}	- cost per unit outside heat transfer area of a shell and tube heat exchanger, $\$/ft^2$.
C_{cond}	- purchased cost of the condenser, \$.
C_{comp}	- purchased cost of the compressor, \$.
CE	- Chemical Engineering Plant Cost Index, 1984 = 322.77.
C_{ee}	- net present worth of the utilities consumed by the heat pump system during the life span of the system, \$.
C_{eq}	- total capital investment in the heat pump system, \$.
C_{ex}	- purchased cost of a heat exchanger, \$.
C_{hp}	- net present worth of the heat pump system, \$.
C_i	- cost to provide 1 ft lbf to pump fluid through the tubes of a heat exchanger, \$/ft lbf.

C_{in} - purchased cost of the innercooler, \$.

C_o - cost to provide 1 ft lbf to pump fluid through the shell side of a heat exchanger, \$/ft lbf.

COP - coefficient of performance, measure of heat pump cycle efficiency, equal to the ratio of heat delivered to the work done by the compressor.

C_p - constant pressure heat capacity, BTU/lbm °R.

C_{reb} - purchased cost of the reboiler, \$.

C_s - purchased cost of the start-up boiler, \$.

C_t - net present worth of the conventional system minus that of the heat pump system, \$.

C_{trim} - purchased cost of the trim-exchanger, \$.

C_u - cost to provide 1 lbm of the utility fluid, \$.

C_v - constant volume heat capacity, BTU/lbm °R.

C_{wt} - cost of 1 lbm of cooling water, \$/lbm.

D - diameter of tube, ft.

D_c - diameter of clearance between tubes, ft.

dh - enthalpy change of a stream in a heat exchanger, BTU/lbm.

Dh - hydraulic diameter of the outside of a tube, ft.

DH_{comp} - enthalpy difference across the compressor, BTU/lbm.

D_i - inside diameter of tubes, ft.

D_o - outside diameter of tubes, ft.

dT - temperature difference between two streams in a heat exchanger, °R.

DT_{cond} - log mean temperature difference across the condenser, °R.

DT_{ex} - log mean temperature difference across an exchanger, °R.

DT_{in} - log mean temperature difference across the innercooler, °R.

DT_{reb} - log mean temperature difference across the reboiler, °R.

DT_{st} - log mean temperature difference across the start-up boiler, °R.

DT_{trim} - log mean temperature difference across the trim-exchanger, °R.

D_1 - fourth parameter in the Euler form of the ideal gas constant pressure heat capacity equation.

E_r - net present worth of one BTU per year of electricity during the life span of the heat pump system, \$/BTU.

E_i - power loss inside tubes of a heat exchanger per unit heat transfer area, ft lbf/hr ft^2.

E_o - power loss outside tubes of a heat exchanger per unit heat transfer area, ft lbf/hr ft^2.

f' - modified fanning friction factor for the shell side of a heat exchanger.

f_i - fanning friction factor for flow through tubes.

fK - thermal conductivity of a fluid, BTU/hr ft °R.

Fs - factor of safety used to account for bypassing.

F_t - correction factor for noncounter flow in heat exchangers.

g - acceleration due to gravity, ft/hr^2.

G - mass velocity in tubes, lbm/sec ft^2.

g_c - generalized conversion factor, ft lbm/sec^2 lbf.

G_r - net present worth of one BTU/yr of natural gas during the life span of the heat pump system, \$/BTU.

G_s - mass velocity outside tubes based on free area inside shell, lbm/sec ft^2.

ha - enthalpy of fluid entering evaporator, BTU/lbm.

hb - enthalpy of fluid entering compressor, BTU/lbm.

hc - enthalpy of fluid leaving compressor based on isentropic compression, BTU/lbm.

hc' - enthalpy of fluid leaving compressor, BTU/lbm.

hd - enthalpy of fluid entering throttling device, BTU/lbm.

h_{fg} - enthalpy of vaporization, BTU/lbm.

h_i - film heat transfer coefficient of fluid inside tubes, BTU/hr ft^2 °R.

h_o - film heat transfer coefficient of fluid outside tubes, BTU/hr ft^2 °R.

H_p - power requirement of the compressor assuming isentropic behavior, horsepower.

h_{si} - fouling factor for the inside of tubes, BTU/hr ft^2 °R.

h_{so} - fouling factor for the outside of tubes, BTU/hr ft^2 °R.

K_c - constant used to correct pressure drop due to contraction.

K_f - ratio of annual fixed charges to the installed cost of the heat exchanger.

K_i - equilibrium constant.

K_1 - constant used to correct pressure drop for contraction, expansion and reversal of flow.

L - length of tubes, ft.

l_i - unifac molecular length fraction.

L_v - latent heat of vaporization, BTU/lbm.

m - mass flow rate of fluid, lbm/hr.

M - molecular weight, lbm/lbmole.

m_{ic} - mass flow rate of overhead column fluid, lbm/hr.

MS - Marshall and Swift Equipment Index, 1984 = 780.4.

N_b - number of baffles.

N_c - number of clearances.

nf - fin efficiency.

n_k - number of times the functional group k appears in a molecule of component i.

nm - mechanical efficiency of compressor.

N_p - number of passes on the tube side.

N_r - number of rows of tubes.

ns - isentropic efficiency of the compressor.

N_t - number of tubes.

nt_i - thermal efficiency of inside surface.

nt_o - thermal efficiency of outside surface.

Nu - Nusselt number.

nv - volumetric efficiency of the compressor.

Nv - number of tubes in a vertical tier.

Oss - number of hours of operation per year.

p - density of fluid, lbm/ft^3.

P - pressure, atm.

Pa - present worth given annual payment factor.

P_c - critical pressure, atm.

Phi_i - variable relating power loss inside tubes to the 3.5 power of the inside film heat transfer coefficient, ft lbf/[BTU/(hr ft^2 °R)]$^{3.5}$.

Phi_o - variable relating power loss outside tubes to the 4.75 power of the outside film heat transfer coefficient, ft lbf/[BTU/(hr ft^2 °R)]$^{4.75}$.

P_r - Prandtl number, $C_p u_f/fK$.

P_v - vapor pressure of fluid, mm Hg.

P_{vr} - reduced vapor pressure.

P_0 - pressure in the heat pump evaporator, atm.

P_1 - pressure in the heat pump condenser, atm.

Q - heat duty, BTU/hr.

Qc - heat duty of the low temperature source, BTU/hr.

Q_{ex} - heat duty of the heat exchanger, BTU/hr.

Qh - heat duty of the high temperature sink, BTU/hr.

q_i - Unifac molecular surface parameter.

Q_k - Unifac functional group surface parameter.

Q_{ratio} - ratio of the condenser heat duty to the reboiler heat duty.

Q_{reb} - reboiler heat duty, BTU/hr.

Q_s - start-up boiler heat duty, BTU/hr.

R - heat capacity ratio, C_p/C_v.

R_{dw} - resistance to heat transfer at the wall including fouling, hr ft^2 °R/BTU.

Re - Reynolds number, ρVD/u.

r_i - Unifac molecular volume parameter.

R_k - resistance to heat transfer through the wall, hr ft^2 °R/BTU.

R_{main} - net present worth of the maintenance requirements of the heat pump system, \$.

Sh - cross sectional area of the header of a heat exchanger, ft^2.

Si - cross sectional area of the tubes of a heat exchanger, ft^2.

St - number of hours per year of start-up operation.

T - temperature, °R.

T_c - critical temperature, °R.

T_{mk} - temperature adjusted Unifac binary functional group interaction parameter.

Tr - reduced temperature

T_0 - temperature on the refrigerant side of the heat pump evaporator, °R.

T_1 - temperature on the refrigerant side of the heat pump condenser, °R.

T_1' - temperature of the compressor discharge, °R.

T_1' - temperature of the process fluid entering a heat exchanger, °R.

T_2' - temperature of the process fluid leaving a heat exchanger, °R.

u - viscosity of a vluid, lb/hr ft.

U_{cond} - overall heat transfer coefficient of the condenser, BTU/hr ft^2 °R.

U_{ex} - overall heat transfer coefficient of a heat exchanger, BTU/hr ft^2 °R.

U_{in} - overall heat transfer coefficient of the innercooler, BTU/hr ft^2 °R.

U_r - ratio of the natural gas rate to the electric rate on a dollars per million BTU basis.

U_{reb} - overall heat transfer coefficient of the reboiler, BTU/hr ft^2 °R.

U_{st} - overall heat transfer coefficient of the start-up boiler, BTU/hr ft^2 °R.

U_{trim} - overall heat transfer coefficient of the trim-exchanger, BTU/hr ft^2 °R.

v - molecular volume, ft^3/lbmole.

V - fluid velocity in the tube, ft/hr.

v_c - critical molecular volume, ft^3/lbmole.

vf - fraction of fluid vaporized.

vs - suction specific volume, ft^3/lbm.

w - weight rate of flow of condensate per tube on the outside, lbf/hr tube.

w - Pitzer accentric factor.

W	- work done by the compressor assuming isentropic behavior, BTU/hr.
W'	- work done by compressor, BTU/hr.
W_u	- mass flow rate of utility fluid, lbm/hr.
wt	- total weight of fluid condensed inside a tube, lbf/hr.
x_i	- mole fraction of component i in mixture.
Xl	- ratio of parallel pitch to outside tube diameter.
X_m	- Unifac mole fraction of functional group m.
Xt	- ratio of transverse pitch to outside tube diameter.
y_i	- mole fraction of component i in mixture.
Z_c	- critical compressibility of a fluid.
z_1	- mole fraction of component in feed to an increment.
Z_i	- fraction of the temperature difference across a heat exchanger that occurs on the inside of the tubes.
Z_{in}	- compressibility factor of the fluid entering the compressor (Pv/RT) determined from the Soave-Redlich-Kwong equation.
Z_m	- Soave-Redlich-Kwong parameter used to find A_z.
Z_o	- fraction of the temperature difference across a heat exchanger that occurs on the outside of the tubes.

Greek Symbols

γ_{iL}	- liquid activity coefficient of component i.
Γ_k	- functional group residual activity coefficient.
λ	- Lagrange multiplier used in deriving equations for the optimum film heat transfer coefficients.
σ	- surface tension, lbf/ft.

ϕ_i - correction factor for nonisothermal flow used in determining the pressure drop inside the tubes.

ϕ_{iV} - vapor fugacity coefficient of component i.

θ_i - Unifac molecular surface fraction of component i.

θ_m - Unifac functional group surface fraction.

μ_i - Unifac molecular segment fraction.

ν_{iL} - pure species liquid fugacity coefficient.

Superscripts

C - combinatorial portion of the liquid activity coefficient.

o - pure species.

R - residual portion of the liquid activity coefficient.

Subscripts

a - properties of the fluid at the entrance.

b - properties of the fluid at the exit.

b - properties of the fluid at the normal boiling point.

c - properties of the fluid at the critical point.

dis - properties of the fluid at the discharge of the compressor.

f - properties of the fluid at film conditions.

g - properties of the gas.

i - properties of the fluid inside the tubes.

i - properties of component i in the mixture.

in - properties of the fluid at the intake of the compressor.

j - properties of component j in the mixture.

k - properties of the functional group k.

l	- properties of the liquid.
L	- properties of the liquid.
lm	- log mean properties.
m	- properties of the mixture.
max	- maximum value of the property.
n	- properties of the functional group n.
o	- properties of the fluid outside the tubes.
r	- reduced property.
u	- properties of the utility fluid.
V	- properties of the vapor.
w	- properties at the wall of the tubes.
0	- properties of the fluid inside the evaporator.
1	- properties of the fluid inside the condenser.
1	- properties at the entrance.
2	- properties at the exit.

CHAPTER I

INTRODUCTION

Heat pumps are hardly a new invention, but with energy costs rising, heat pumps are receiving more interest.

As one of the more energy inefficient areas of the chemical industry, distillation is also one of the largest consumers of energy. Distillation trains require about 28 percent of the energy consumed by the chemical industry in the United States.[1] Most of this energy is lost through the condensation of the overhead vapor.

The feasibility of the application of a heat pump system to a distillation train depends on many variables. Among these are the temperature difference between the reboiler and the condenser, the temperature levels, and the heat loads of both the reboiler and condenser. It has been shown that heat pumps can decrease the energy consumed by a distillation column by 80 percent while adding between 10 and 20 percent to the capital cost.[2]

CHAPTER II

CONVENTIONAL DISTILLATION COLUMN

The distillation column is a means of separating chemicals. This separation is based on a difference in volatility between these chemicals. It is obtained through the interactions between a liquid phase and a vapor phase, each flowing in opposite directions in the column. Each stage of the column operates close to physical equilibrium. The greatest attribute of the distillation column is the high degree of purity that can be obtained.

Energy is added to the column through a reboiler where a portion of the liquid phase is withdrawn from the bottom of the column, vaporized, and returned to the column. The reboiler is generally heated by steam.

Energy is withdrawn from the column through the condenser where the vapor phase is withdrawn from the top of the column, condensed, and a portion returned to the column for reflux. The condenser is cooled by cooling water, air, or other cooling medium. The condenser operates at a lower temperature than the reboiler.

The temperature difference between the reboiler and condenser frustrates any attempt to recover the heat lost in the condenser and apply it to the reboiler.

CHAPTER III

BASIC HEAT PUMP

A heat pump delivers heat from a lower temperature source to a higher temperature sink as a result of the input of compressor work to the system. It consists of an evaporator, condenser, compressor, throttling device, innercooler, and a trim cooler arranged as in Figure 1. Ideally the evaporator, condenser, and the trim cooler are constant-temperature phase change operations. Ideally the compressor is isentropic. An ideal Rankine cycle is illustrated in Figure 2 on a temperature-enthalpy diagram. A measurement of efficiency of a heat pump is the coefficient of performance (COP) which is the ratio of energy delivered (Qh) to the work required at the compressor (W).[2]

$$COP = Qh/W$$

The governing equation of the Rankine cycle is that the energy delivered at the condenser is equal to the energy absorbed in the evaporator plus the work performed by the compressor.

$$m(hc - hd) = m(hb - ha) + W$$

and

$$W = m(hc - hb)$$

where m is the working fluid mass flow rate, and h is the enthalpy at

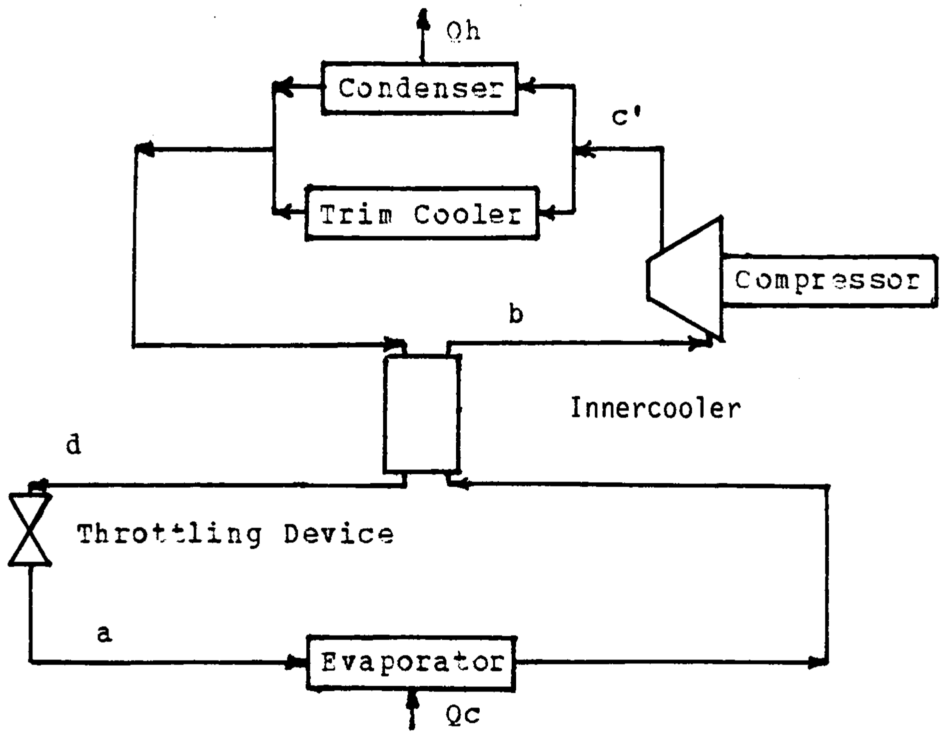

Figure 1. Schematic drawing of a heat pump.

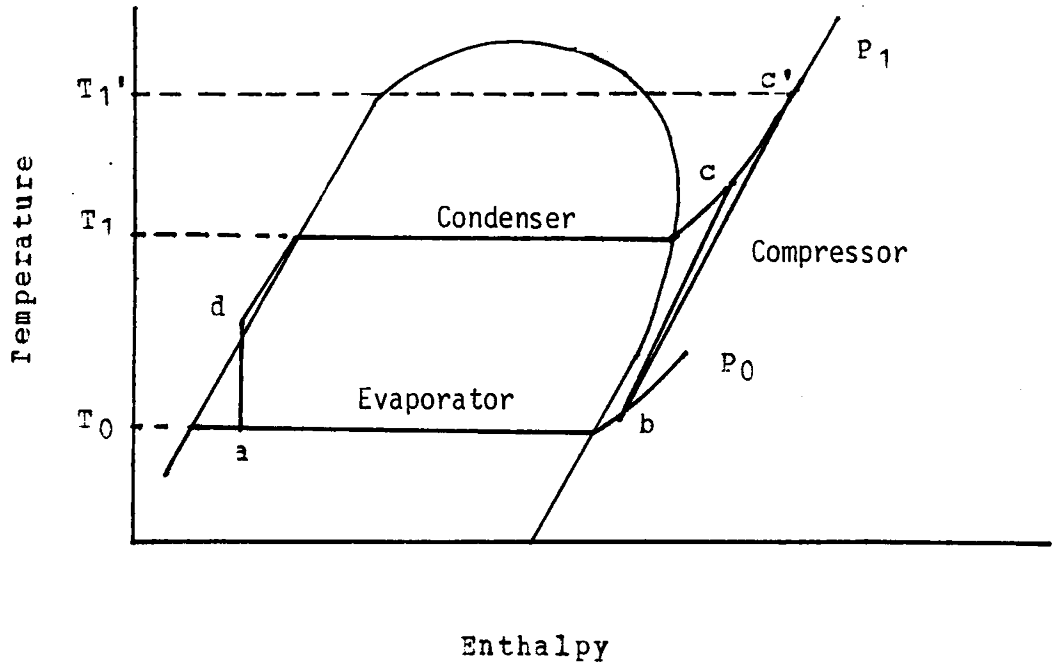

Figure 2. The ideal Rankine cycle.

points a, b, c, or d. Note that hd = ha because the enthalpy does not change across the throttling device.[2,4]

In order to take into account the irreversible nature of the polytropic compressor as well as the heat exchange between the compressor and the working fluid, the point c is replaced by c'. The work required is therefore increased to W'.[4]

$$W' = m(hc' - hb)$$

The ratio of the work ideally required to this increased work is referred to as the isentropic efficiency ns.

$$ns = W/W' = (hc - hb)/(hc' - hb)$$

The compressor does not deliver all of the energy it uses, giving rise to a second efficiency coefficient nm, called the mechanical efficiency.

$$nm = m(hc' - hb)/(\text{energy consumed by compressor})$$

The practical COP is therefore altered to:

$$COP = nm(hc' - hd)/(hc' - hb)$$

A third inefficiency arises in the compressor which is called the volumetric efficiency, nv.

$$nv = mvs/(\text{compressor swept volume})$$

where vs is the suction specific volume. This does not affect the COP but does affect the size of the compressor.[4]

CHAPTER IV

WORKING FLUIDS

The selection of a working fluid is one of the most critical choices involved in the design of a heat pump. The first consideration in the selection process concerns the stability of the fluid when exposed to the temperature at the discharge of the compressor. Good design requires that this temperature be well below the fluid critical temperature (reduced temperature not greater than 0.9).

The absolute value of the slope of the saturated vapor entropy curve, i.e. $(dT/ds)_{vap.}$, should be high to reduce the amount of super heating in the compressor $(T_1' - T_1)$. This is the same as having a low constant pressure heat capacity (c_p) over the temperature range.[5]

$$T_1' - T_1 = T_1(T_0 - T_1)/(c_p(dT/ds)_{vap.})$$

$$(dT/ds)_{vap.} = T/(c_p - h_{fg}/T)$$

The fluid should also have a high latent heat of vaporization (h_{fg}) which will give a lower compressor suction volume flow rate. Unfortunately, fluids with a high latent heat of vaporization require high compression ratios (P_1/P_0).[5]

$$\ln(P_1/P_0) = h_{fg}(T_1 - T_0)/(PT_0T_1)$$

One other consideration is that of the solubility of this compressor lubricant in the working fluid. This can cause loss of lubricant, loss of working fluid, contamination of either fluid, and/or a breakdown of the lubricant.

With these things in mind, the selection process begins by eliminating fluids with a reduced temperature greater than 0.9 at the compressor discharge. Then remove the fluids that pose special problems due to flammability or toxicity. Next, using the ideal Rankine cycle, calculate the compression ratio, the compressor displacement (volume flow rate at the compressor inlet), and the coefficient of performance. This along with the relative cost of the refrigerants, should be enough to narrow the choice down to a few refrigerants.[4,5]

Refrigerants, as working fluids are generally called, are given a shorthand notation called an ASHRAE (American Society of Heating, Refrigerating, and Air-Conditioning Engineers) number.[6] The ASHRAE number is a basic three digit code. The first digit is the number of carbons in the formula of the compound minus one. It is omitted if it is less than one. The second digit is the number of hydrogens in the formula plus one. The last digit is the number of fluorines. The number one is added to the beginning if the compound is unsaturated. A capital C is attached to the beginning if it is a cyclic compound. For brominated compounds a capital B followed by the number of bromines in the formula is added to the end of the ASHRAE number. The number of chlorines in the formula is not a part of the ASHRAE number, but is implied by the ASHRAE number. Isomers are numbered with respect to symmetry. The most symmetrical receives just the normal number, while the rest receive an alphabetic suffix. The second-most symmetrical has the letter a attached; the third, a letter b, etc. For example, R-12 is dichlorodifluoromethane or freon, R-114 is dichlorotetrafluoroethane, and R-114a is the isomer of R-114.[6]

CHAPTER V

COMPRESSORS

There are many types of compressors available for use in heat pumps. The choice of type of compressor is dependent on the inlet volume flow rate, the compression ratio, and the discharge pressure. Two broad operating categories of compressors are wet and dry. In a wet compressor, the compressor lubricant is exposed to the refrigerant causing the lubricant to be present in the refrigerant to a concentration level above 15 ppm. In some cases, the lubricant concentration can reach 20 percent by volume. The lubricant is necessary to enhance compressor seal performance or to cut down on compressor wear. In a dry compressor, lubricant is only present due to leaks in the bearing seals and the concentration level rarely is greater than 5 ppm.

There are three types of casings for a compressor plus its motor drive: hermitic, semi-hermitic, and open. In the open type, the drive shaft emerges through a seal from the casing containing the refrigerant. In the hermitic type, both the compressor and the motor are contained in the same welded casing. This eliminates the seal and the refrigerant leakage therein. The major drawback of this is the increased danger of the motor overheating. Semi-hermitic is the same as hermitic except that the casing is bolted rather than welded for easier servicing.

A reciprocating compressor works in much the same way as an automatic engine. Compression occurs in a cylinder by force applied by a piston. The refrigerant is impinged on the cylinder walls as it enters to help with cooling the cylinder and to improve the isentropic efficiency. Liquid in the intake of the compressor can result in damage to valves and in an extreme case, can damage the cylinder head and pistons. Reciprocating compressors are equipped with a spring loaded cylinder head to prevent damage.

There are two variations of a reciprocating compressor. In one, a thin film of lubricant is maintained on the cylinder walls. In the other, a crosshead is introduced between the piston and the crankshaft. This is done to remove the angular forces on the piston allowing it to glide on a thin film of refrigerant. This creates a dry compressor instead of a wet one, but also makes it more expensive. Single stage reciprocating compressors are limited to a 4 to 1 compression ratio.

Centrifugal compressors utilize the centrifugal force to achieve compression. This is done when the refrigerant vapor is thrashed by the rotating impeller blades. The impeller blade rotates at a speed between 2500 rpm to 30000 rpm. The upper value is limited by the stress on the impeller.

Capacity is controlled by either varying speed, suction guide vane angles, or a blast gate opening. Suction guide vanes are the most common. They work by rotating to a more favorable or less favorable position. Speed variance is the most efficient method but is both difficult and expensive when used with constant speed electric motors. Blast gates are the least efficient method.

The prime problem with a centrifugal compressor is that of surging. This problem is caused by momentarily higher pressure in discharge than the pressure developed by the compressor, thus the flow reverses. This is an oscillatory flow which is unstable. Surging may be predicted by the shape of the head capacity curves which reach a maximum at approximately half capacity and then plummet to zero.[4,7]

The type of compressor required can be determined from the following application ranges of compression ratio, adiabatic head, and suction volume flow rate. A reciprocating compressor is limited by a maximum suction volume flow rate of 1000 ft^3/min where the maximum compression ratio has fallen to 4.0 per stage from 10 per stage. A centrifugal compressor is used for flow rates above 1000 cubic ft/min with a limiting adiabatic head of 10,000 ft of gas.[7]

Compressor lubricants are generally mineral oils and the selection of the lubricant depends on the choice of working fluid.

It should be noted that the compressor of a heat pump is exposed to higher temperatures than a similar compressor used in a refrigerating application. The danger of the bearings drying out is therefore increased and larger bearings are required.[8] If the refrigerant is allowed to accumulate at the bearings, lubricants can be leached out.

CHAPTER VI

HEAT EXCHANGERS

The design equation for heat exchangers is:

$$Q = U_{ex}AdT = mdh$$

where Q is the heat duty of the heat exchanger, A is the exchange area, U_{ex} is the overall heat transfer coefficient, dT is the temperature difference between the two streams, m is the mass flow rate of a stream, and dh is the change in enthalpy of that stream as it passes through the heat exchanger.

Except when dT is a constant across the heat exchanger (both streams change phase), dT is replaced by $F_t dT_{lm}$.[9]

$$F_t dT_{lm} = F_t(dT_a - dT_b)/\ln(dT_a/dT_b)$$

where dT_{lm} is the log mean temperature difference, dT_a is the temperature difference between the two streams at the inlet, dT_b is the temperature difference between the two streams at the outlet, and F_t is a correction factor which is equal to 1.0 for an ideal counter-flow heat exchanger.

The overall heat transfer coefficient can be found from the following equation:[10]

$$U_{ex} = ((1/(nt_o hs_o)) + (1/(nt_o h_o)) + Rk + (A_o/(A_i nt_i hs_i)) + (A_o/(A_i nt_i h_i)))^{-1}$$

where hs is the fouling coefficient of the surface, Rk is the thermal resistance of the wall, a is the total surface area, h is the average unit conductance of the surface, and the subscripts i and o refer to the inside and outside surfaces, respectively.

The total efficiency of a surface (nt) is equal to 1.0 if no fins are present. It is determined from the following equation otherwise:[10]

$$nt = 1.0 - Af(1.0 - nf)/A$$

where Af is the heat transfer area of the fin, nf is the fin efficiency and A is the heat transfer area of the surface if no fins were present.

The average unit conductance of the outside surface can be determined from the Nusselt analogy.[10]

$$h_o D/fK_f = Nu = a(Re_{max})^{0.6}(Pr_f)^{0.3}/Fs$$

$$Re_{max} = (pV_{max}D)/u_f$$

$$Pr_f = (C_p u_f)/fK_f$$

where the subscript f refers to film properties which are defined as having a temperature equal to the arithmetic average of the bulk stream temperature and the temperature at the surface. The coefficient a is 0.33 for staggered tube bundles, and 0.26 for in line tube bundles. Fs is the factor of safety to account for by passing and varies from 1.0 to 1.8 with an average value of 1.6. fK is the thermal conductivity of the fluid, D is the outside diameter of the tube, p is the density of the fluid, u is the absolute viscosity of the fluid, and V_{max} is the maximum fluid velocity in the tube. This equation is valid for Reynolds numbers greater than 6000, Prandtl numbers between 0.7 and 300, with no phase change.

The average unit conductance of the inside surface is found similarly with the following equation:[10]

$$Nu_{Dh} = 1.86(Re_{Dh}PrDh/L)^{0.33}(u_b/u_s)^{0.14}$$

where Dh is the hydraulic diameter, and the subscripts b and s refer to bulk and surface conditions, respectively. L is the length of the tube. Nu_{Dh} and Re_{Dh} are the Nusselt and Reynolds numbers based on the hydraulic diameter. This equation is valid for Reynolds number less than 2100, Prandtl number greater than 0.7, and no phase change. For the situation that the Reynolds number is greater than 6000 but all other conditions are the same, the following equation is valid:[10]

$$Nu_{Dh} = 0.023(Re_{Dh})^{0.8}(Pr)^{0.33}$$

For condensing vapors outside horizontal tubes:[10]

$$h_o = 0.725(K_f{}^3P_f{}^2gh_{fg}/(NvD_ou_fdT))^{0.25}$$
$$= 0.95(K_f{}^3P_f{}^2gL/(wu_f))^{1/3}$$

where Nv is the number of tubes in a vertical tier, h_{fg} is the latent heat of condensation [BTU/lbm], L is the heated length of the tube [ft], w is weight rate of flow of condensate per tube from the lowest point on the condensing surface [lbf/hr], and dT is the temperature difference across the liquid film [°F]. D is in ft, u is in lbf/(hrft), h is in BTU/(hrft^2°F), fK is in BTU/(hrft°F), g is in ft/hr^2, and p is in lbm/ft^3. This is valid for Reynolds number greater than 2000, with all physical properties being for the condensate.

For condensing vapor outside vertical tubes:[10]

$$h_o = 1.47(\pi D_o fK_f{}^3 p_f{}^2 g/(4wu_f))^{1/3}$$

For condensing vapors inside horizontal tubes at low flow rates:[10]

$$h_i = 0.761(LfK_l{}^3 p_l(p_l - p_v)g/(wtu_l))^{1/3}$$

$$= 0.612(fK_l{}^3 p_l(p_l - p_v)gh_{fg}/(u_l D_i dT_f))^{0.25}$$

where wt is the total vapor condensed in one tube [lbm/hr], and the Reynolds number is less than 2100.

At a high condensing load with vapor shear dominating the Boyko-Kruzhilin Correlation applies:[7]

$$Nu = 0.024(D_i G/u_l)^{0.8}(pr_l)^{0.43}((p/p_m)_i + (p/p_m)_o)/2$$

$$(p/p_m)_i = 1 + (p_l - p_v)x_i/p_v$$

$$(p/p_m)_o = 1 + (p_l - p_v)x_o/p_v$$

where G is the mass velocity in the tube [lbm/(hrft^2)], and x is the mass vapor fraction.

For boiling either the Mostinski Equation[7,12]

$$hb = 0.00658(P_c)^{0.69}(Q/A)^{0.7}(1.8(P/P_c)^{0.17} + 4(P/P_c)^{1.2} + 10(P/P_c)^{10})$$

or the McNelly Equation[7]

$$hb = 0.225(gc_p/(Ah_{fg})^{0.69}((p_l/p_v - 1)^{0.33}$$

can be used for the nucleated boiling regime. P is in psia, Q is in BTU/hr, hb is the average unit conductance for boiling [BTU/(hrft^2°F)], and σ is the surface tension of the fluid [lbf/ft]. In a heat pump, the boiling regime is assumed to be nucleated, because of the low temperature difference between the wall and the bulk liquid. The subscript c refers to a critical property.

One other important factor in designing a heat exchanger is the pressure drop experienced by each stream. For single phase flow in tubes, the pressure drop is given by:[10]

$$dP_i = B_i 2f_i G^2 LN_p/(g_c p_i D_i \phi_i)$$

where the subscript i refers to properties inside the tube at bulk temperature, G is the mass velocity in the tube in lbm/sec ft^2, f is the fanning friction factor, ϕ is the correction factor for nonisothermal flow, g_c is 32.17 ftlbm/(sec^{2}lbf), P is in lbf/ft^2, N_p is the number of passes, and B is the correction factor for expansion, contraction and flow reversal. ϕ is determined from the following equation if the Reynolds number is less than 2100:

$$\phi_i = 1.1(u_i/u_w)^{0.25}$$

where the subscript w refers to wall conditions. It is determined from this equation otherwise:

$$\phi_i = 1.02(u_i/u_w)^{0.14}$$

B is determined from:

$$B_i = 1 + K_1 D_i \phi_i/(4f_i L)$$

The parameter K_1 is:

$$K_1 = (1 - (Si/Sh))^2 + K_c + 0.5(N_p - 1)/N_p$$

K_c is:

$$K_c = 0.4(1.25 - Si/Sh)$$

where Si is the total inside tube cross sectional area and Sh is the header cross sectional area, and the ratio of Si to Sh is less than 0.715. If this ratio is greater than 0.715, then K_c is found from the following equation:

$$K_c = 0.75(1 - Si/Sh)$$

The fanning friction factor can be obtained from graphs or from the following relationships:[10]

For Reynolds number less than 2100,

$$f_i = 16/Re$$

For Reynolds number greater than 6000 in smooth pipe,

$$f_i = 0.046/Re^{0.2}$$

For Reynolds number greater than 6000 in steel or new iron,

$$f_i = 0.04/Re^{0.16}$$

Shell side pressure drop can be calculated from the following equation for no change of phase:[10]

$$dP_o = -B_o 2f' N_r G_s^2/(g_c p_o)$$

where G_s is the mass velocity based on free area in shell in lbm/sec ft^2, N_r is the number of rows crossed by flow, and B is a correction factor for reversal of flow, recrossing of tubes, variations in cross sectional area. B is roughly equal to the number of baffles plus one, and f' is the modified friction factor determined by the following equations:

$$f' = b_o Re^{-0.15}$$

For staggered tube arrangements,

$$b_o = 0.23 + 0.11/(Xt - 1)^{1.08}$$

For in line tube arrangements,

$$b_o = 0.044 + 0.08Xl/(Xt - 1)^{(0.43+1.13/Xl)}$$

where Xt is the ratio of transverse pitch to tube diameter, and Xl is the ratio of parallel pitch to tube diameter. These equations are valid for Reynolds numbers between 2000 and 40,000.

There are several types of reboilers that can be applied to the situation where a condensing fluid gives up heat to a boiling fluid. The first type is the flooded bundle or kettle reboiler. Boiling occurs on the shell side, with the condensing fluid inside horizontal tubes. The pressure drop on the shell side due to friction is negligible provided that liquid flow paths are adequate to prevent vapor binding. This is the normal type of the heat pump evaporator because of the requirement of a vapor fraction of 1.0 for the refrigerant at the exit.

In kettle reboilers, the heat flux is generally limited to 70 percent of the maximum defined by the Palen and Small modification of the Foster-Zuber Equation.[7]

$$(Q/A)_{max} = 61.6 P p_v h_{fg} (g\sigma(p_l - p_v)/p_v{}^2)^{0.25}/(D_o N^{0.5})$$

where N_t is the number of tubes, σ is the surface tension [lbf/ft], p is the density [lbm/ft^3], P is the pressure [psia], and the subscripts l and v refer to the liquid and vapor phase, respectively. Note that any desuperheating of the condensing vapor is done by a single phase mechanism.

Horizontal thermosiphon, vertical thermosiphon, forced recirculation, and natural convection reboilers can be designed by a method developed by Dr. James R. Fair.[13] In all of these types of reboilers, boiling occurs on the tube side (forced recirculation reboilers can accommodate boiling on the shell side as well) and deliver a fluid that is only partially vaporized. This is done to avoid the mist flow region with the accompanying low heat transfer coefficients. Dr. Fair's method was developed for vertical thermosiphon reboilers but is easily modified for use with the other types.

CHAPTER VII

THROTTLING DEVICES

There are two predominant types of throttling devices, capillary tubes (inside diameters between 0.026, and 0.1 inches) and expansion valves. The latter is generally preferred as it has better control capabilities, especially in startup and shutdown situations. Flashing flow occurs in either situation.

The design equation of a capillary tube is:[14]

$$vdP + G^2(vdv + 2f_i v^2 dL/D)/g_c = 0$$

where v is the specific volume of the fluid. Numerical solutions of this equation are more easily found from this form of the same equation:[14]

$$dL = -D(g_c dP/(G^2 v) + dv/v)/(2f_i)$$

where v is the average specific volume over the increment, and f_i is the fanning friction factor for liquid flow until two phase flow begins. The apparent friction factor for two phase flow is given by Edward P. Mikol as 95 percent of the value for single phase liquid flow. The required length of the tube being found by taking small increments of the pressure drop and summing up the lengths.

One problem with designing a capillary tube is that the speed of sound can be obtained in the tube, and the pressure will not decrease any further. This is indicated when the incremental length calculated is equal to zero (if this length is less than zero, then the increment is too large). If the pressure at this point is still too high, then the assumed diameter must be increased. The speed of sound for a two phase mixture is given by Mikol as:[14]

$$c = (g_c(dP/d\rho)_S)^{0.5}$$

One type of expansion valve is the constant pressure expansion valve. In this valve, two opposing springs act against each other. The closing spring is attached to the valve pin while the opening spring acts on a diaphragm that rests on top of the push rod. The evaporator pressure is admitted beneath the diaphragm either through an opening just beyond the valve seal called an internal equalizer or through a capillary tube attached to a point downstream of the valve called an external equalizer. The outlet pressure (evaporator pressure) is adjusted by either tightening or loosening the opening spring.

This type of valve is best suited for a constant heat load application. If the heat load is increased the valve will starve the evaporator; and if the heat load decreases, it will flood it. In cases where the motor drive of the compressor has a low starting torque, a bleed hole is added to the valve seat to allow the pressure to equalize the valve.

In a thermostatic expansion valve, the opening spring of the constant pressure expansion valve is replaced by a capillary line to a bulb containing a fluid that expands as the temperature rises. The bulb is attached to the evaporator outlet as is the external equilizer. The temperature in the bulb together with the pressure from the equilizer adjust the heat duty of the evaporator to meet present operating conditions.

The fluid in the bulb can be the same fluid as the refrigerant in two forms, a gas charge or a liquid charge. In a gas charge, at a set point all the liquid in the bulb has vaporized and above this point the pressure exerted on the diaphragm does not change significantly, causing the valve to act like a constant pressure expansion valve. If the capillary or diaphragm chamber is colder than the bulb, condensation may form lowering the set point. In liquid charge, the bulb is made to contain enough liquid such that it does not all vaporize.

In liquid cross charges, a fluid other than the refrigerant is used. The fluid used has a less steep pressure versus temperature curve to moderate valve response.

One control problem associated with a thermostatic expansion valve is that of hunting. Hunting is caused by the valve alternatively starving and flooding the evaporator. This is avoided by using liquid cross charges, proper sizing of equipment, a short refrigerant path through the evaporator, and proper locations for the bulb and equalizer.

Thermostatic expansion valves are rarely used with flooded bundle evaporators because of the low superheat of the vapor leaving the evaporator.

The two types of float valves are applicable to controlling the flow rate to a flooded bundle evaporator. In a high side float valve, refrigerant enters the float chamber before entering the evaporator. The main drawback of this is that the amount of refrigerant in the heat pump is a critical factor. The liquid level is controlled by matching the flow rate through the compressor.

A low side float valve controls the liquid level in the evaporator more directly. The float chamber is connected to the evaporator by two lines, a vapor line and a liquid line. This is done to insure the same liquid level in the float chamber as in the evaporator. The valve overcomes the drawback of the high side float valve, but must be manually closed when the heat pump is not operating to prevent it from overfilling the evaporator.

The last type is a pilot operated expansion valve. This is generally a piston type, pilot operated regulator, with the pilot valve being one of the types mentioned previously. It is used when the flow rate is too large for a single direct acting valve to be used. This valve takes on the operating characteristics of the pilot valve, and also allows secondary control valves to act on the pilot line alone, to control the main flow.

It should be noted that all valves require adequate protection from corrosion and foreign material, through the application of strainers and drying systems.[15]

CHAPTER VIII

APPLICATION TO DISTILLATION

There are four basic configurations in which a heat pump can be applied to a distillation column which are illustrated in Figures 3 through 6. The first way is the most obvious, that is, with a refrigerant in the heat pump different from the column fluid. The column condenser is replaced by the heat pump evaporator, and the column reboiler is replaced by the heat pump condenser. There is no alteration of the process fluid path through the column and therefore no danger of degrading that fluid, either through contamination, or exposure to high temperature of the compressor discharge.[16]

A second way is to use the column overhead vapor as the heat pump working fluid. This has the main advantage of eliminating the need for the condenser/reboiler. It also reduces the temperature rise required across the compressor. How well this fluid acts as a working fluid determines the usefulness of this method. A flash drum is added to recycle the uncondensed vapor leaving the expansion valve. A dry compressor is used to minimize the contamination of the process fluid by compressor lubricants.[16]

A third method is to use the bottoms liquid as the heat pump working fluid in a similar manner.[16]

A fourth configuration may be applied in which high and low temperature reservoirs are set up. Heat is supplied to the high temperature reservoir by circulating fluid from the low temperature

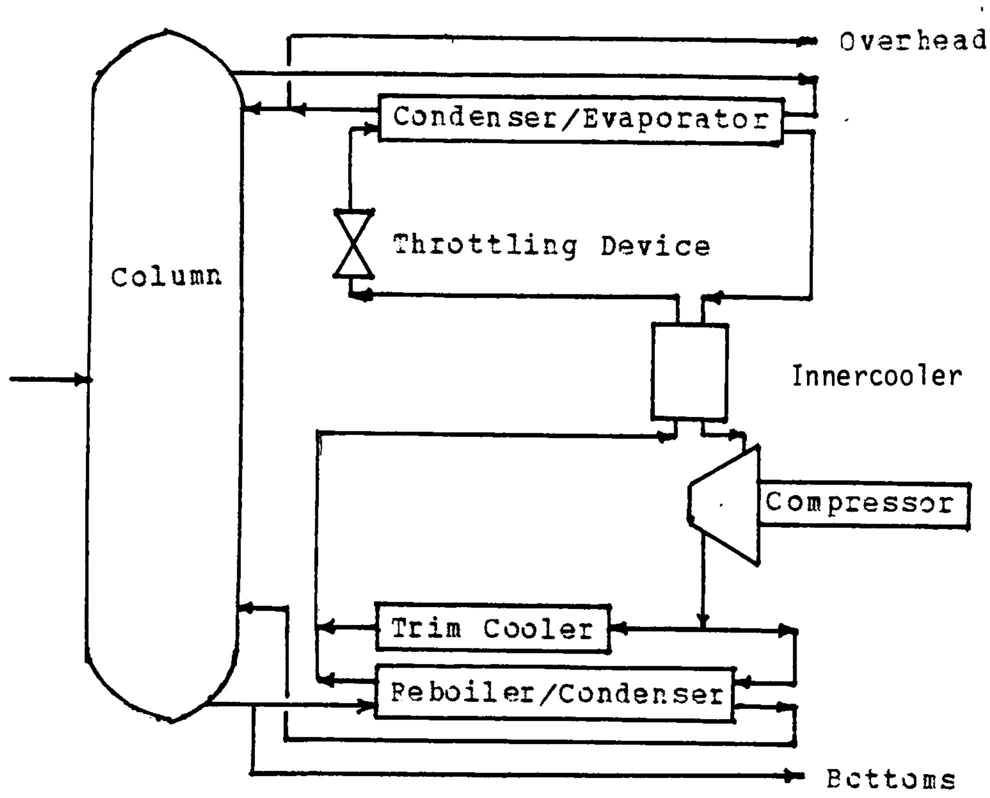

Figure 3. Refrigerant in the heat pump different from the column fluid.

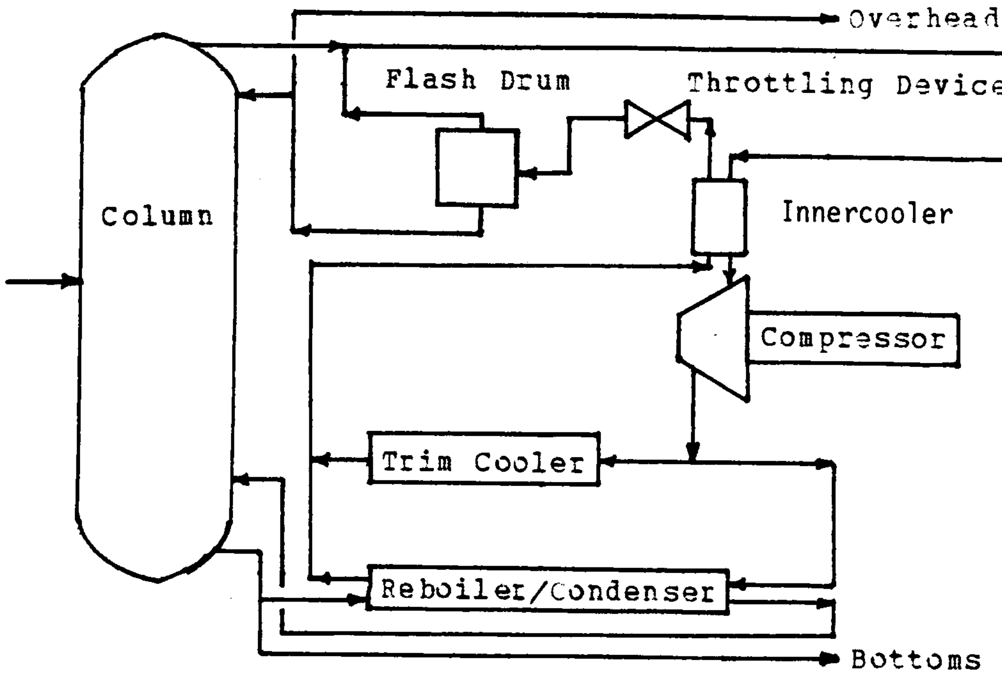

Figure 4. Column overhead as refrigerant in heat pump.

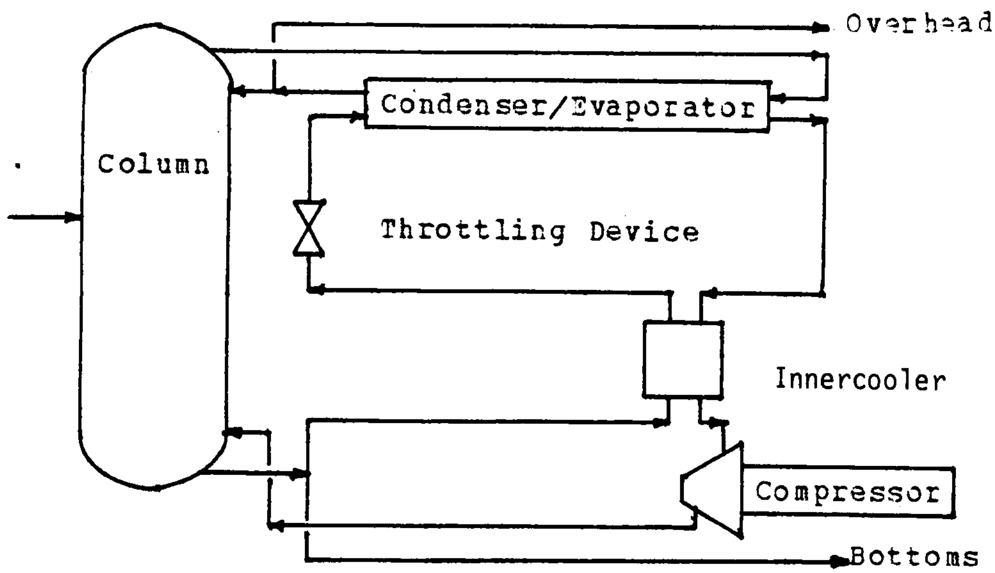

Figure 5. Column bottoms as refrigerant in heat pump.

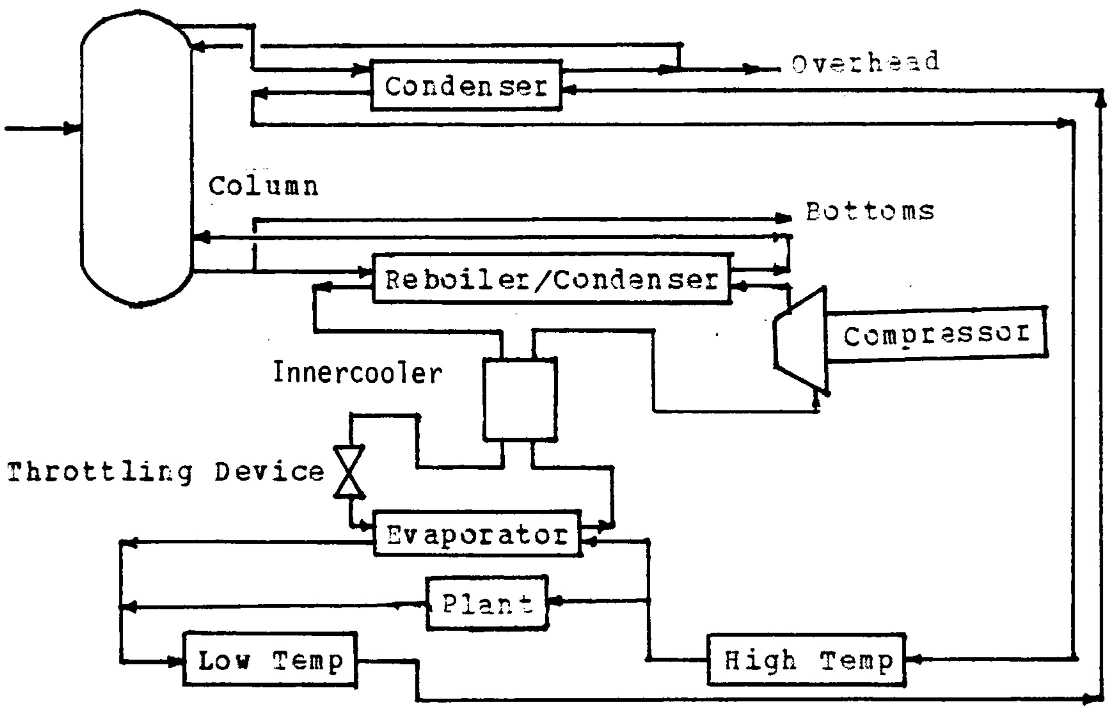

Figure 6. Heat pump system utilizing temperature reservoirs.

reservoir through the column condenser. Fluid from the high temperature reservoir provides the heat source for the heat pump supplying the reboiler, then flowing to the low temperature reservoir. This system is applicable when the column condenser is capable of supplying more heat than is needed in the column reboiler and the temperature in the high temperature reservoir is sufficient to supply the heat needed elsewhere in the plant.[17]

In general, many of the limitations on heat pump applications in distillation columns are centered around the heating requirements of the rest of the plant. If the heat generated at the condenser is providing the heating needs of the rest of the plant with little waste, a switch to a heat pump system is rarely justified.[16]

It should be noted that the amount of heat available at the heat pump condenser is greater than the amount of heat absorbed at the evaporator. If this is more than that required in the column reboiler, this is an added bonus if it can be applied elsewhere in the plant, otherwise it is an added burden.

CHAPTER IX

ANALYSIS METHODOLOGY

Introduction

The project is divided into two parts. In the first part, feasibility quidelines, the effects of seven variables on the economic feasibility of the heat pump system are studied. In this part, the heat pump system is limited to the configuration with an external refrigerant which is illustrated in Figure 3 (p. 28). The process fluids involved in the heat pump system are unknown and estimated heat transfer coefficients are used. The utility ratio is used as a measure of economic feasibility in that the actual utility ratio must be greater than that determined from the graphs developed for the heat pump system to be feasible.

In the second part, specific case analysis, the process fluids are known so it is not necessary to make the rough estimates of the heat transfer coefficients required in the first part. The heat pump system is freed to use other configurations and options to improve its economic potential. The return on investment replaces the utility ratio as the measure of economic feasibility.

Feasibility Guidelines

The purpose of this section is to determine the conditions required in a distillation column to make the application of a heat pump system economically feasible. To this end a number of assumptions were made. The heat pump system was limited in configuration to the case of an external refrigerant cycle (Figure 3, p. 28). This was done to limit the number of assumptions regarding the physical properties of the process fluids.

Process Fluid Heat Transfer Coefficients

The film resistances to heat transfer on the process side in both the reboiler and condenser are assumed to be noncontrolling with respect to overall heat transfer coefficient. A value of 200 BTU/hr sq ft°R was used as the heat transfer coefficient of the condensing overhead column fluid. A value of 300 BTU/hr sq ft°R was used as the heat transfer coefficient of the boiling column bottom fluid. Both of these values represent the lower end of the range given by Peters and Timmerhaus,[10] and are used because the temperature difference across the heat exchanger is small.

The overall heat transfer coefficients for the conventional reboiler and condenser were approximated by 700 BTU/hr sq ft°R and 500 BTU/hr sq ft °R, respectively.[10]

Physical Properties of Heat Pump Fluids

Physical properties of the refrigerant and other known fluids (cooling water and steam) were estimated from a number of equations. The vapor pressure was estimated using the Antoine equation:

$$\ln(P_v) = A - B/(C + T)$$

where P_v is the vapor pressure in mm Hg, T is the temperature in degrees Kelvin and A, B, and C are constants with appropriate units.[18]

The Soave-Redlich-Kwong equation was used to determine the state properties of the fluid:

$$P = RT/(v - B_s) - A_s A_z / v(v + B_s)$$

where P is the pressure, R is the ideal gas constant, T is the temperature, v is the molar volume, and A_s, A_z, and B_s are constants with consistent units. A_z is determined by the following equations:

$$A_z = [1 + Z_m(1 - T_r^{0.5})]^2$$

where T_r is the reduced temperature (T/T_c), and Z_m is determined by the following equation:

$$Z_m = 0.48 + 1.574w - 0.176w^2$$

where w is the Pitzer accentric factor. The constants A_s and B_s are determined from properties at the critical point by the following equations:

$$A_s = 0.4278R^2T_c^{\,2}/P_c$$

$$B_s = 0.0867RT_c/P_c$$

where T_c is the critical temperature and P_c is the critical pressure.[19]

The Euler form of the ideal gas heat capacity equation was used to estimate the constant pressure heat capacity of gases:

$$C_p = A_1 + B_1T + C_1T^2 + D_1T^3$$

where C_p is in Cal/gmole°K, T is the temperature in degrees Kelvin, and A_1, B_1, C_1, and D_1 are constants with appropriate units.[18] To this ideal value for C_p is added the constant pressure derivative of the enthalpy departure function with respect to temperature to correct for situations where the pressure is too high for the ideal gas assumption to hold. For liquids, the equation of Sternling and Brown was used:

$$(C_{pL} - C_p)/R = [3.67 + 11.64(1 - T_r)^4 + 0.634/(1 - T_r)]$$
$$(0.5 + 2.2w)$$

where C_{pL} is the constant pressure heat capacity of the liquid, C_p is the ideal gas heat capacity, R is the ideal gas constant, T_r is the reduced temperature, and w is the Pitzer accentric factor.[18]

The viscosity of a liquid was estimated using the Andrae equation:

$$u_L = A_2\exp(B_2/T)$$

where u_L is the liquid viscosity in centipoise, T is the temperature in degrees Kelvin, and A_2 and B_2 are constants with appropriate units.[7]

The viscosity for gases was estimated using Arnold's correlation:

$$u_g = 0.00027M^{0.5}T^{1.5}/v_b{}^{2/3}(T + 1.47T_b)$$

where u_g is the viscosity in centipoise, M is the molecular weight, T is the temperature in degrees Kelvin, v_b is the molar liquid volume at the normal boiling point in cc/gmole, and T_b is the normal boiling temperature in degrees Kelvin.[7]

The thermal conductivity of gases was estimated using Eucken's approximation:

$$fK = u_g(C_p + 2.48/M)$$

where fK is the thermal conductivity in btu/hr ft°F, u_g is the vapor viscosity in lb/hr ft, C_p is the constant pressure heat capacity in btu/lb°F, and M is the molecular weight in lb/lbmole.[7]

The thermal conductivity of liquids was estimated using the following equation:

$$fK_L = 0.00264(C_p/C_{pb})(p/p_b)^{4/3}(T/T_b)/M^{1/2}$$

where the subscript b refers to conditions at the normal boiling point, p is the molar liquid density, T is the temperature, p is the molar liquid density, T is the temperature, and fK_L is the thermal conductivity in Cal/cm sec°K.[18]

The latent heat of vaporization, L_v, was estimated using Chen's equation:

$$L_v = T(7.9T_r - 7.82 - 3.09\mathrm{Log}(P_{vr}))/(1.07 - T_r)$$

where P_{vr} is the reduced vapor pressure (P_v/P_c), T is the temperature in degrees Kelvin, T_r is the reduced temperature, and L_v is in Cal/gmole.[18]

Utility Consumption

Steam used in the start-up boiler and conventional reboiler was assumed to be delivered at 904.4 degrees Rankine (400 psia). The cost of steam was limited to the cost of the natural gas required to generate it. No credit was taken for reducing the size of the steam generator required because a reasonable estimate could not be made without more information on the heat balance for the plant.

Cooling water, used in the trim-cooler and the conventional condenser, was assumed to be delivered at 520 degrees Rankine. A cost of $0.12 per thousand gallons in 1979 dollars was assumed.[10] No credit was taken for reducing the size of the cooling tower because an estimate could not be made without more information.

Equipment Size and Costs

The purchased cost of heat exchangers was estimated by the following equation:

$$C_{ex} = 190.35(Q_{ex}/U_{ex}DT_{ex})^{0.632}MS/561$$

where C_{ex} is the purchased cost of the heat exchanger, Q_{ex} is the heat duty of the exchanger in btu/hr, U_{ex} is the overall heat transfer coefficient in btu/hr sq ft°R, DT_{ex} is the temperature difference across the exchanger in degrees°R, and MS is the value of the Marshall and Swift Equipment Index (1984 = 780.4). This value is multiplied by 1.2 if the exchanger is a kettle type heat exchanger.[20]

The cost of the centrifugal compressor was estimated by the following equation

$$C_{comp} = 150.48(H_p/0.7)^{0.82}CE/119$$

where C_{comp} is the purchased cost of the compressor, H_p is the power requirement of the compressor in horsepower (note that 0.7 is to account for the isentropic efficiency of the compressor), and CE is the value of the Chemical Engineering Plant Cost Index (1984 = 322.7).[21]

The isentropic horsepower requirement of the compressor was determined by the following equation:

$$H_p = Z_{in}(1.987*1.415/3600)(CrT_{in}/M)(R/(R - 1))$$
$$((P_{dis}/P_{in})^{((R-1)/R)} - 1)$$

where Cr is the circulation rate of the refrigerant in lb/hr, R is the heat capacity ratio (C_p/C_v) determined using the Soave-Redlich-Kwong equation, P_{dis} is the compressor discharge pressure in atm, P_{in} is the intake pressure in atm, M is the molecular weight, Z_{in} is the compressibility of the fluid (Pv/RT) at the compressor intake determined

determined using the Soave-Redlich-Kwong equation, and T_{in} is the intake temperature in degrees Rankine.[7]

The total capital investment was estimated as 5.69 times the total purchased cost of the equipment required. This includes installation, instrumentation, engineering, construction, etc.[10]

Economic Parameters

A 10 percent discount rate was used in all net present worth calculations.

The plant was assumed to be operated 330 days per year with 2 days start-up time. The operating life of the plant was assumed to be 10 years.

Maintenance was estimated as 20 percent of the purchased equipment cost.[10]

All prices were escalated to 1984 dollars using either the Marshall and Swift Equipment Cost Index, or the Chemical Engineering Plant Cost Index.

No salvage values for the equipment were included.

Optimization of the Heat Pump System

The two most critical variables to the design of a heat pump system are the temperature difference across the reboiler (DT_{reb}) and the temperature difference across the condenser (DT_{cond}). The best values for these variables are determined by setting the derivatives of the net present worth equation of the heat pump system, with respect to each one, equal to zero and solving simultaneously. The net present worth equation is as follows:

$$C_{hp} = C_{ee} + C_{eq} + R_{main}$$

where C_{hp} is the net present worth of the heat pump system, C_{ee} is the net present worth of the utilities consumed during the life-span of the heat pump system, C_{eq} is the total capital investment, and R_{main} is the net present worth of the required maintenance during the life-span of the heat pump system.

C_{ee} was estimated by the following equation:

$$C_{ee} = OssE_rCrDH_{comp}/(0.95*0.7) + StE_rU_rQ_s/0.8$$
$$+ C_{wt}W_uOssPa$$

where Oss is the number of hours of operation per year, E_r is the net present worth of one BTU/yr of electricity during the life-span of the heat pump system, Cr is the refrigerant circulation rate in lb/hr, and DH_{comp} is the change in enthalpy of the refrigerant in the compressor assuming isentropic behavior in BTU/lb. The factor (0.7*0.95) represents the combined mechanical and isentropic efficiencies, St is the number of hours per year of start-up operation, Q_s is the heat duty of the start-up boiler in BTU/hr, 0.8 is the boiler efficiency, C_{wt} is the cost per pound of cooling water, W_u is the number of pounds of cooling water required per hour, Pa is the present worth given annual payment factor, and U_r is the utility ratio (natural gas rate/electric rate).

DH_{comp} was determined with the following equation:

$$DH_{comp} = Z_{in}\ 1.987T_{in}R/(M(R - 1))(P_{dis}/P_{in})^{(R-1)/R} - 1)$$

where T_{in} is the temperature at the intake of the compressor in Rankine, R is the ratio of the constant pressure heat capacity, C_p, to the constant volume heat capacity, C_v, M is the molecular weight, P_{dis} is the pressure at the discharge of the compressor, and P_{in} is the pressure at the intake of the compressor.[7]

C_{eq} was determined from the following equation:

$$C_{eq} = 5.69(C_{cond} + C_{reb} + C_{comp} + C_{trim} + C_{in} + C_s)$$

where C_{cond} is the purchased cost of the condenser, C_{reb} is the purchased cost of the reboiler, C_{comp} is the purchased cost of the compressor, C_{trim} is the purchased cost of the trim-heater or trim-cooler, C_{in} is the purchased cost of the innercooler, C_s is the purchased cost of the start-up boiler, and 5.69 is the factor for estimating the total capital investment from the total purchased equipment cost.

The complex equation for C_{hp} is then differentiated with respect to DT_{reb} and with respect to DT_{cond} to obtain the following equations:

$$dC_{hp}/dDT_{reb} = dC_{ee}/dDT_{reb} + dC_{eq}/dDT_{reb} + dR_{main}/dDT_{reb}$$

$$dC_{hp}/dDT_{cond} = dC_{ee}/dDT_{cond} + dC_{eq}/dDT_{cond} + dR_{main}/dDT_{cond}$$

In evaluating these two equations, DH_{comp} and Cr are the only variables in C_{ee} that are dependent upon either DT_{cond} or DT_{reb}. DH_{comp} is dependent upon both DT_{reb} and DT_{cond} through the compression ratio (P_{dis}/P_{in}). This dependence forces a simultaneous solution. C_{eq} is dependent upon both DT_{reb} and DT_{cond}. C_{cond} is dependent upon DT_{cond} both directly and through U_{cond}. C_{reb} is dependent upon DT_{reb} directly, through U_{reb} and through the size of the desuperheating section of the reboiler. C_{comp} is dependent upon both DT_{reb} and DT_{cond} through H_p. If a trim-heater is present, then C_{trim} is dependent upon DT_{cond} through DT_{trim} which is the temperature difference across the trim-heater in degrees Rankine. If a trim-cooler is present, then C_{trim} is dependent upon DT_{reb} through DT_{trim}. The dependence of U_{trim}, the overall heat transfer coefficient of the trim heat exchanger in btu/hr sq ft R, on either DT_{reb} or DT_{cond}, is ignored. C_{in} is dependent upon both DT_{reb} and DT_{cond} through DT_{in}, the log mean temperature difference across the innercooler in degrees Rankine, while the overall heat transfer coefficient, U_{in}, is considered a constant. C_s is dependent upon DT_{cond} through DT_{st}, the temperature difference across the start-up boiler, while the overall heat transfer coefficient, U_{st}, is considered a constant. R_{main} is dependent upon DT_{reb} and DT_{cond} in much the same way as C_{eq}.

These two equations are then set equal to zero and are solved simultaneously for DT_{reb} and DT_{cond}. This is done by assuming a value for DT_{cond} and solving for DT_{reb}. The value for DT_{reb} is then used to find the next value for DT_{cond} as in Figure 7. Once DT_{reb} and DT_{cond} are determined, the refrigerant circulation rate, the cost of the reboiler, the cost of the condenser, and the cost of the compressor are all easily found.

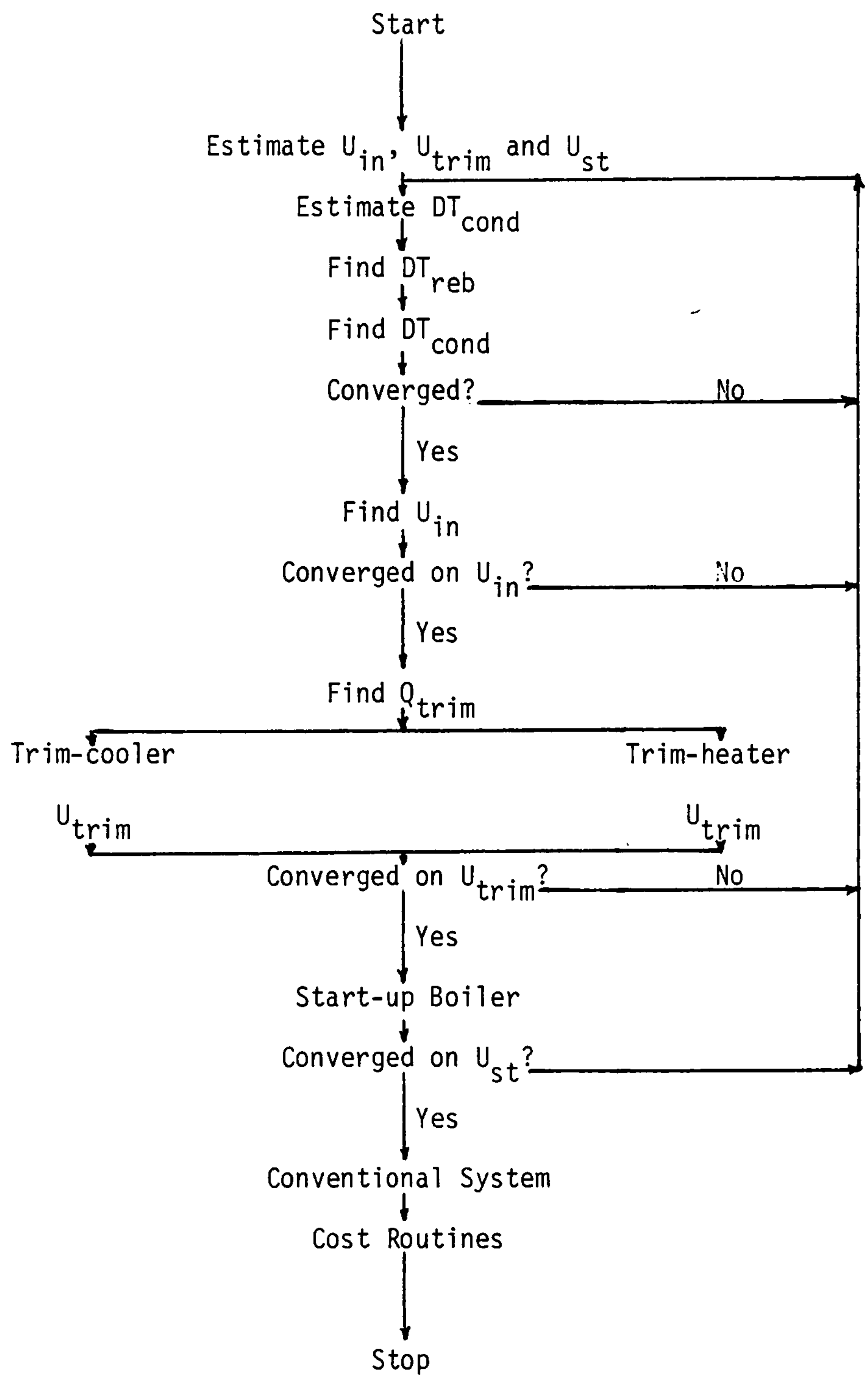

Figure 7. Outline for heat pump model.

Overall Heat Transfer Coefficient

Heat exchangers are broken into three groups: no phase change on either side, phase change on both sides, and phase change on one side.

The heat exchangers with phase changes on both the tube and shell sides are the easiest to specify. With the inside tube diameter set to 0.87 inches, the outside diameter set to 1 inch, and the tube length set to 16 feet, the only variables left undetermined are Z_i and Z_o. Z_i and Z_o are the fraction of the temperature difference across the heat exchanger that occurs on the inside of the tubes and on the outside of the tubes. Z_i and Z_o can be determined from the relative magnitudes of the resistances to heat flow on each side with respect to the total resistance to heat flow in the heat exchanger:

$$Z_o = U_{ex}/h_o$$

$$Z_i = U_{ex}/h_i$$

where h_i is the film heat transfer coefficient on the tube side in BTU/hr sq ft°R and h_o is the heat transfer coefficient on the shell side in BTU/hr sq ft°R. The determination of Z_i and Z_o is an iterative one, because H_i and H_o depend on Z_i and Z_o. The start-up boiler, the condenser, the reboiler, and the trim-heater are of this type. The trim-cooler may be of this type when this heat is to be used somewhere else in the plant.[10]

Heat exchangers without a phase change on either side are more complicated. This is because the heat transfer coefficients and the pressure drops are dependent upon the heat transfer area. The development of an optimal value for the heat transfer area begins with an equation for the total annual cost of the heat exchanger and its operation:

$$C_t = A_o K_f C_{Ao} + W_u Oss C_u + A_o E_i Oss C_i + A_o E_o Oss C_o$$

where C_t is the total annual cost, A_o is the heat transfer area (outside tubes), K_f is the ratio of annual fixed charges (including maintenance, depreciation, and labor) to the installed cost of the heat exchanger, C_{Ao} is the installed cost of the heat exchanger per square foot of heat transfer area, W_u is the mass flow rate of the utility fluid in lbm/hr, Oss is the number of hours of operation per year, C_u is the cost per pound of the utility fluid, E_i is the power loss on the inside of the tubes per square foot of heat transfer area in ft lbf/hr sq ft, C_i is the cost to pump the fluid through the inside of the tubes in $/ft lbf (includes cost of pump and efficiency effects), E_o is the power loss on the shell side, and C_o is the cost to pump the fluid through the shell side. All the cost factors is linearized to simplify this equation. However, this makes all calculations iterative.[10]

The following expressions are then substituted in for E_i and E_o:

$$E_i = Phi_i h_i^{3.5}$$

$$E_o = Phi_o h_o^{4.75}$$

where Phi_i and Phi_o are determined from the following equations:

$$Phi_i = B_i[12200D_i^{1.5}u_i^{1.83}(u_{wi}/u_i)^{0.63}$$

$$/(g_cD_op_i^2fk_1^{2.33}C_{pi}^{1.17})]$$

$$Phi_o = (B_o/N_b)(N_rN_c/N_t)[2b_oD_cD_o^{0.75}F_s^{4.75}u_{fo}^{1.42}$$

$$/(\pi a_o^{4.75}g_cp_o^2fK_{fo}^{3.17}C_{pfo}^{1.58})]$$

where a_o is 0.26 since the tubes are assumed to be in line, D_c is the diameter of the clearances between tubes and is equal to 0.25 inch (1.25 inch square pitch), D_o is 1 inch, D_i is 0.87 inch, B_i and B_o are the pressure drop correction factors for contraction, expansion, etc., b_o is the constant used to evaluate shell side friction, p is the density of the fluid in lbm/ft^3, N_r is the number of rows, N_c is the number of clearances, N_t is the number of tubes, N_b is the number of baffle spaces, F_s is a correction factor to account for bypassing and a value of 1.6 was used, g_c is 32.174 $(3600)^2$ ft lbm/hr^2 lbf, the subscript i refers to conditions on the inside of the tubes, the subscript o refers to conditions on the shell side, the subscript f refers to film properties, and the subscript w refers to conditions at the wall of the tube.[10]

The total annual cost equation was then subjected to the normal constraint of the design equation:

$$Q_{ex} = U_{ex}A_{ex}DT_{ex}$$

When the method of Lagrange multipliers is applied, the following object equation is arrived at:

$$C_t = \lambda[F_t(DT_2 - DT_1)/(Q_{ex}\ln(DT_2/DT_1))$$

$$- (D_o/(D_i h_i) + 1/h_o + R_{dw})/A_o]$$

$$+ A_o Phi_i h_i^{3.5} OssC_i + A_o Phi_o h_o^{4.75} OssC_o$$

$$+ Q_{ex} OssC_u/(C_{pu}(DT_1 - DT_2 + T_1' - T_2')) + A_o K_f C_{Ao}$$

where λ is the Lagrange multiplier, F_t is the correction factor applied to the log mean temperature difference to correct for flows other than counterflow in the exchanger, R_{dw} is the combined resistances to heat transfer due to the wall and fouling on both sides of the wall, DT_1 is the temperature difference at the inlet to the exchanger in Rankine, DT_2 is the temperature difference at the exit, T_1' is the temperature of the process fluid at the inlet in Rankine, and T_2' is the temperature of the process fluid at the exit in Rankine. The equation is now simplified in accordance with the case in hand, differentiated, and solved simultaneously for h_i and h_o.[10]

In the case of the innercooler, the compressor feed is superheated by the fluid leaving the reboiler. The mass flow rate on the tube side is the same as on the shell side, making the temperatures in the object function constants. This also makes the heat transfer coefficient a constant. The cost to pump the liquid refrigerant through the innercooler is negligible because an increase in the pressure drop on this fluid only reduces the allowable drop in the throttling device ($C_o = 0$). The object function is then differentiated with respect to the heat transfer area and with respect to the inside heat transfer coefficient. The two resulting equations are then combined to eliminate the Lagrange multiplier and the heat transfer area to obtain an equation for finding h_i:

$$K_f C_{Ao} = [3.5 Phi_i OssC_i D_i/D_o(1/h_o + R_{dw})h_i + 2.5 Phi_i OssC_i]h_i^{3.5}$$

This is solved in an iterative manner. Z_i and Z_o are solved for in the same manner as in the case with phase change on both sides.[10]

The trim-cooler has a phase change on only one side of the exchanger. The change in temperature and the pressure drop on the process or refrigerant side is negligible, making C_o equal to zero. The same equation for the innercooler can be used for the trim-cooler. The temperature drop across the exit of the exchanger is solved for with the following equation:[10]

$$F_t U_{ex} OssC_u/(C_{pu}(K_f C_{Ao} + E_i OssC_i)) = \ln(DT_2/DT_1) - 1 + DT_1/DT_2$$

Variables Investigated

Several column variables were looked at to determine their effect on the application of the heat pump system. The seven variables investigated were the electric rate forecast, the utility ratio, the reboiler heat duty, the heat duty ratio, the process side condenser temperature, the process side temperature difference between the reboiler and the condenser, and the heat pump working fluid.

The electric rates, E_r, used in this analysis were those predicted by TVA. The three forecasts are probabilistic models with the high forecast having a 90 percent probability of the actual rate being lower than the rate predicted, the medium forecast having a 50 percent change, and the low forecast having a 10 percent chance. A graph of the three forecasts is presented in Figure 8.[21]

The utility ratio, U_r, is the ratio of the natural gas rate to the electric rate (each in dollars per million BTU). This variable was used as a measure of feasibility, in that the utility ratio must be higher than those obtained from the graphs for the application of the heat pump to be feasible. The utility ratios predicted from TVA data are presented in Figure 9.[21]

The reboiler heat duty, Q_{reb}, determines the size of the heat pump required. It was allowed to vary from 200,000 BTU/hr to 200,000,000 BTU/hr by increments of 10. The application of the heat pump system is more favorable at high reboiler heat duties.

The heat duty ratio, Q_{ratio}, is the ratio of the condenser heat duty to the reboiler heat duty. This variable determines if a trim-cooler or trim-heater is required. Trim-heaters are required for Q_{ratio} values below the no trim-exchanger value and trim-coolers being required above that value. The no trim-exchanger value is below 1.0 because of the heat added by the compressor. The Q_{ratio} is allowed

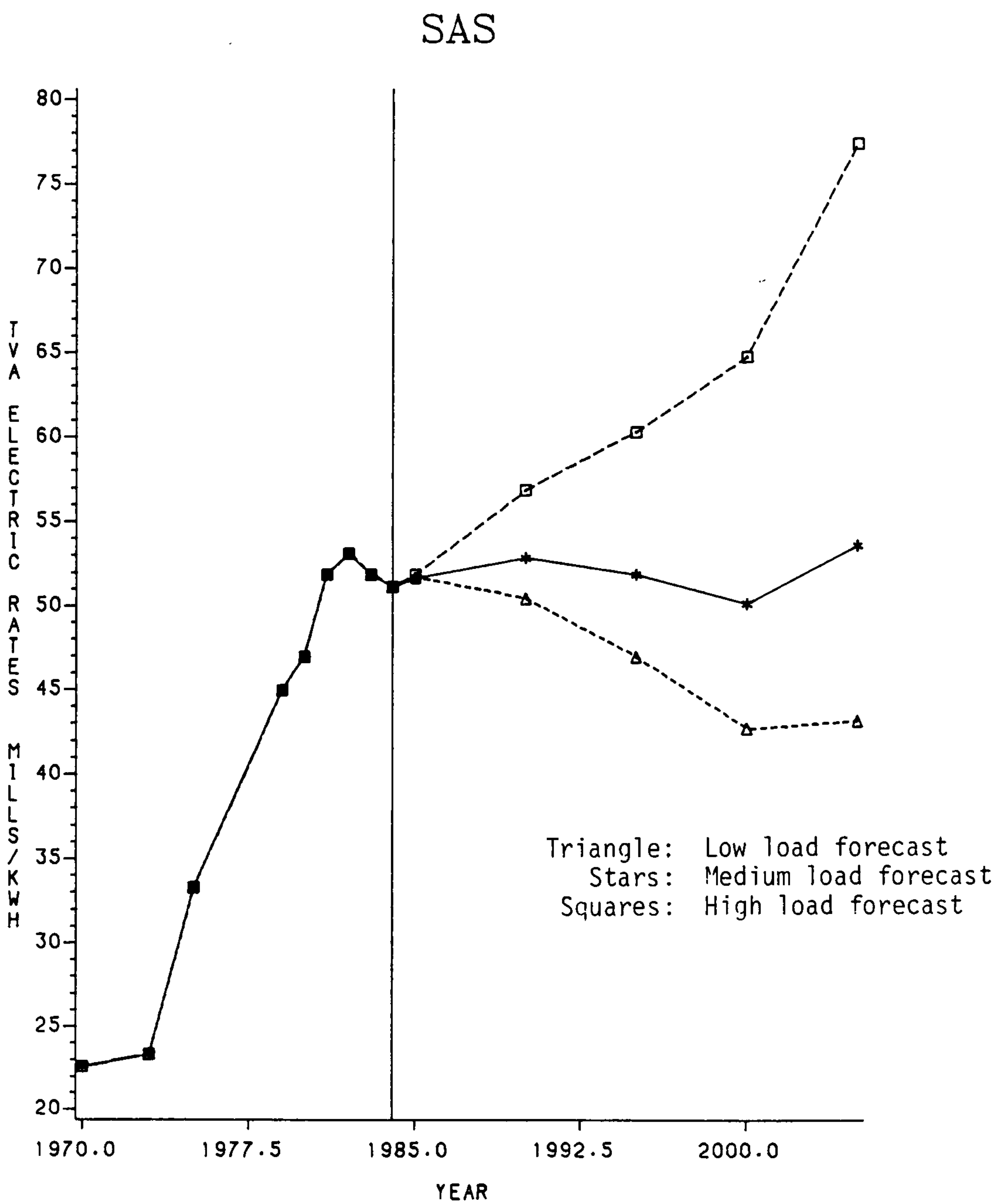

Figure 8. Industrial electric rates vs. year TVA load forecast for 1986. (Based on 1984 mills/kwh.)

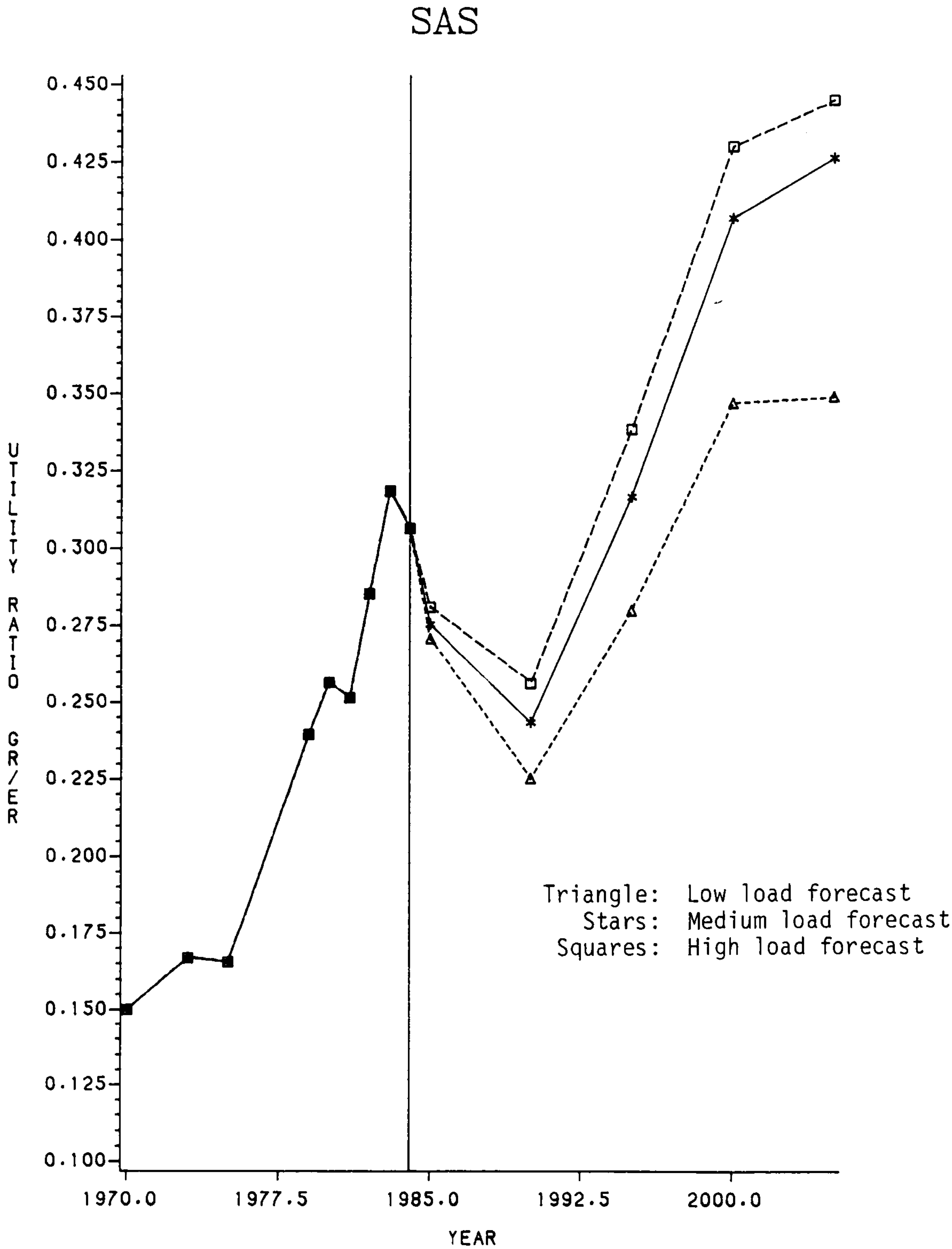

Figure 9. Industrial utility ratio vs. year TVA load forecast for 1986. (Based on 1984 dollars.)

to vary from 0.5 to 1.5. The heat pump system is favored by columns where the need for trim-coolers or trim-heaters is small.

The condenser temperature, T, is the bulk average temperature on the process side. This temperature is allowed to vary from 560 R to the lower of 710 R or the point where it makes the compressor discharge temperature exceed 90 percent of the refrigerant critical temperature.

The temperature difference, DT, between the reboiler and the condenser determines the compression ratio required in the compressor. The application of the heat pump system is favored by low temperature differences. DT is allowed to vary from 10 R to the lower of 90 R or where it makes the compressor discharge temperature exceed 90 percent of the refrigerant critical temperature.

The working fluids used in this analysis are R-114 and R-113. Both fluids are commonly used refrigerants and have high critical temperatures.

Specific Case Analysis

In analysis of a specific case, several issues must be considered. First, consider the assumptions made previously regarding the process fluids. These assumptions can be replaced in the analysis by what is known about the process fluids. Second, there are numerous options that need to be investigated that potentially could improve the heat pump economics. Of these options, configuration of the heat pump system appears to be the most promising.

Process Fluids

The fluids examined previously were all assumed to be pure liquids and gases, while process fluids are generally multicomponent mixtures. The physical properties of multicomponent mixtures are estimated by an average of the properties of pure components.

Critical properties of mixtures (pseudocritical) are estimated using Kay's rule:

$$P_{cm} = \sum y_i P_{ci}$$

$$T_{cm} = \sum y_i T_{ci}$$

$$Z_{cm} = \sum y_i Z_{ci}$$

$$v_{cm} = Z_{cm} R T_{cm} / P_{cm}$$

where y_i is the mole fraction of component i, P_c is the critical pressure, T_c is the critical temperature, Z_c is the critical compressibility, v_c is the critical molar volume, and R is the ideal gas constant in consistent units. The subscripts m and i refer to mixture and pure component properties, respectively.[22]

The Soave-Redlich-Kwong equation is modified by the following mixing rules:

$$B_{sm} = \sum y_i B_{si}$$

$$A_{sm} A_{zm} = \sum \sum y_i y_j (A_{si} A_{sj} A_{zi} A_{zj})^{(1/2)}$$

where y is the mole fraction of a component.[18,19]

The constant pressure heat capacity of gas mixtures was estimated by:

$$C_{pm} = \sum y_i C_{pi}$$

where y_i is the mole fraction of component i. For liquid mixtures, the constant pressure heat capacity is estimated by:

$$C_{pm} = \sum x_i C_{pi}$$

where x_i is the weight fraction of component i.[7]

The viscosity of gas mixtures was estimated using the following equation:

$$u_m = (\sum y_i u_i M_i^{(1/2)}) / \sum y_i M_i^{(1/2)}$$

where y_i is the mole fraction of component i and M_i is the molecular weight of component i.[7] For liquid mixtures, the Kendall-Monroe equation was used:

$$u_m^{(1/3)} = \sum x_i u_i^{(1/3)}$$

where x_i is the mole fraction of component i.[7]

The thermal conductivity of gas mixtures was estimated by:

$$fK_m = (\sum y_i K_i M_i^{(1/3)}) / \sum y_i M_i^{(1/3)}$$

where y_i is the mole fraction of component i. For liquid mixtures:

$$fk_m = \sum x_i fK_i$$

where x_i is the mole fraction of component i if M_1 is greater than M_2 and fK_1 is greater than fK_2; otherwise x_i is the weight fraction of component i.[7]

The latent heat of vaporization of a mixture where the temperature is allowed to vary while the pressure and overall composition remains constant is determined as follows:

$$L_{vm} = x_j L_{vj}$$

where x_j is the mole fraction of component j.[7]

Film Heat Transfer Coefficients

The film heat transfer coefficient of a multicomponent mixture is an average of the film heat transfer coefficients of the pure components:

$$h_m = \sum x_j h_j$$

where x_j is the mole fraction of component j. However, if a change of phase occurs, then the composition is not a constant.[23]

In condensation, the temperature varies along with the composition of each phase. An incremental approach is therefore used. The exchanger is broken into increments of temperature and the area is determined for each.

The first time through the process of determining the liquid and vapor compositions of an increment, the equilibrium constant, K_i, is set equal to the pure species liquid fugacity coefficient:

$$K_i = \nu_{iL}^0$$

where ν_{iL}^0 is the pure species liquid fugacity coefficient.[10]

The pure species liquid fugacity coefficient is determined from the following equation which was derived from the Soave-Redlich-Kwong equation:

$$\ln\nu_{iL}^0 = Z_{iL} - 1.0 - \ln Z_{iL} + \ln(v_{iL}/(v_{iL} - B_{si}))$$
$$- A_{si}A_{zi}/(RTB_{si})\ln((v_{iL} + B_{si})/v_{iL})$$

where Z_{iL} is the liquid compressibility of component i and v_{iL} is the liquid molar volume of component i in ft^3/lb mole.[18,19]

Thereafter the equilibrium constants are determined from the following equation:

$$K_i = \gamma_{iL}\nu_{iL}^0/\phi_{iV}$$

where K_i is the equilibrium constant, ϕ_{iV} is the vapor fugacity coefficient of component i, and γ_{iL} is the liquid activity coefficient of component i.[19]

The liquid activity coefficient was determined using the UNIFAC equation:

$$\ln\gamma_{iL} = \ln\gamma_{iL}^C + \ln\gamma_{iL}^R$$

where the superscripts R and C refer to the combinatorial portion and to the residual portion, respectively. The UNIFAC parameters are based on functional groups rather than molecules.[19]

The combinatorial portion was determined from the following equation:

$$\ln\gamma_{iL}^{C} = \ln(\mu_i/x_i) + 5q_i\ln(\theta_i/\mu_i) + l_i - \mu_i/x_i \sum(x_j l_j)$$

where μ_i is the molecular segment fraction, x_i is the mole fraction of component i, q_i is the molecular surface parameter, θ_i is the area fraction, and l_i is defined as follows:

$$l_i = 5(r_i - q_i) - (r_i - 1.0)$$

where r_i is the molecular volume parameter and q_i is the molecular surface area parameter.[19]

μ_i was determined from the following equation:

$$\mu_i = x_i r_i/\sum x_j r_j$$

where the subscripts i and j refer to components i and j.[19]

θ_i was determined from the following equation:[19]

$$\theta_i = x_i q_i/\sum x_j q_j$$

The two molecular parameters r_i and q_i are determined from their functional group counterparts as follows:

$$r_i = \sum n_k R_k$$

$$q_i = \sum n_k Q_k$$

where n_k is the number of times the functional group k appears in a molecule of component i.[19]

The residual portion was determined in the following manner:

$$\ln\gamma_{iL}^{R} = \sum n_k(\ln\Gamma_k - \ln\Gamma_k^{o})$$

where Γ_k is the functional group residual activity coefficient and Γ_k^{o} is the pure component functional group residual activity coefficient.[19]

Both Γ_k and Γ_k^{o} are determined from the following equation with appropriate composition terms:

$$\ln\Gamma_k = Q_k(1.0 - \ln(\sum\theta_m T_{mk}) - \sum\theta_m T_{km}/\sum\theta_n T_{nm}$$

where θ_m is the area fraction of the functional group m and T_{mk} is the functional group interaction parameter.[19]

θ_m is defined by the following:

$$\theta_m = X_m Q_m/\sum X_n Q_n$$

where X_m is the functional group mole fraction and is defined by:

$$X_m = \sum n_m x_j/\sum \sum n_k x_i$$

where x_j is the mole fraction of component j.[19]

T_{mk} is determined from:

$$T_{mk} = \exp(-A_{mk}/T)$$

where T is the temperature in consistent units with A_{mk}.[19]

Q_m, R_m, and A_{mk} are found in tables of UNIFAC parameters.[24]

ϕ_{iV} was determined from the following equation derived from the Soave-Redlich-Kwong equation:

$$\ln\phi_{iV} = (B_{si}/B_{sm} - 2\sum(A_{si}A_{sj}A_{zi}A_{zj})^{0.5}y_j/A_{sm}A_{zm})$$

$$(A_{sm}A_{zm}/B_{sm}RT)\ln((v_g + B_{sm})/v_g)$$

$$+ B_{si}(Z_g - 1)/B_{sm} - \ln Z_q + \ln(v_g/(v_g - B_{sm}))$$

where Z_g is the compressibility of the vapor phase ($P\underline{V}/RT$), V_g is the molar volume of the vapor phase, the subscripts i and j refer to the corresponding components, and the subscript m refers to the mixture.[18,19]

Once the equilibrium constants are found, the fraction of the feed that remains a vapor is found from the following equation:

$$0.0 = \sum z_i(1 - K_i)/(1 + vf(K_i - 1))$$

where z_i is the mole fraction of component i in the feed to the increment and vf is the fraction of the feed that remains a vapor at the end of the increment.[19]

New values for x_i and y_i are determined from the following:

$$x_i = z_i/(1 + vf(K_i - 1))$$

$$y_i = K_i x_i$$

and the process repeats itself until there is no significant change in x_i or y_i.[19]

The amount of heat transferred in the increment is determined from the amount of vapor condensed, and the heat transfer area of the increment is determined.

This procedure is done for each increment and the total heat transfer area of the condenser is the sum of the incremental areas. This procedure is outlined in Figure 10.

Boiling mixtures in a kettle reboiler can be assumed to be well mixed. Therefore, the composition of the liquid mixture is a constant in the reboiler and is equal to the composition of the liquid mixture leaving the reboiler.

Options

The first set of options are considered with configuration. These options were previously outlined in Chapter VIII. In using one of the column fluids as the heat pump working fluid (Figures 4 and 5, pp. 29 and 30), the analysis is easier because one variable to be optimized is removed, either DT_{reb} or DT_{cond}. The circulation rate is set at the flow rate of the column fluid used as the heat pump refrigerant. If the column overhead fluid is used, the vapor produced at the throttling device is recycled. The circulation rate is then determined from the following:

$$C_r = \frac{\dot{m}_{ic}}{(1 - vf)}$$

where $\dot{m}_{ic}$ is the flow rate of the column fluid and vf is the fraction vaporized at the throttling device. If the temperature configuration of Figure 6 (p. 31) is used, the analysis is harder. The temperatures of the high and low temperature reservoirs must be determined from the heating requirements of the rest of the plant. The circulation rate

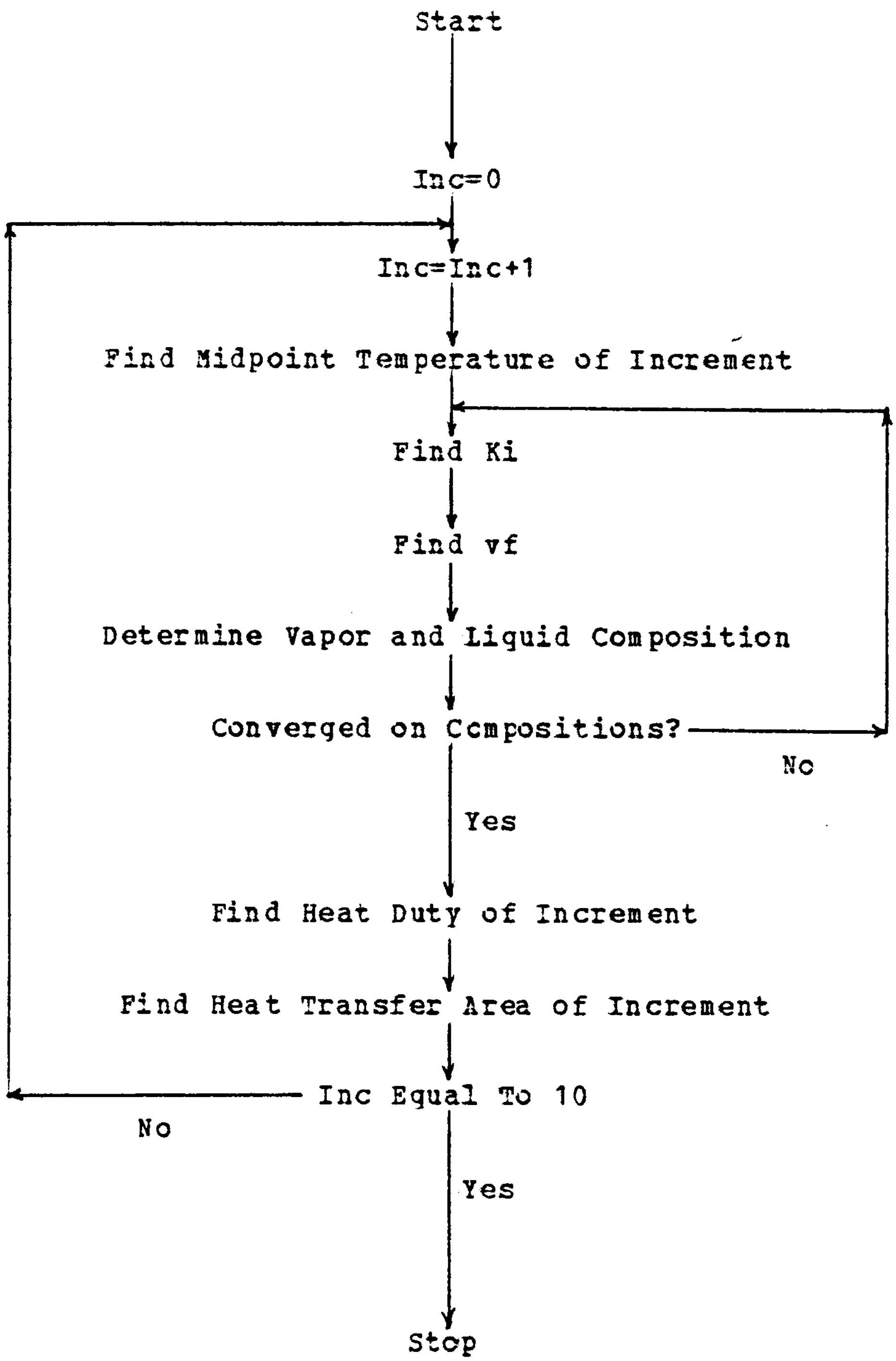

Figure 10. Determination of vapor-liquid composition.

of the second working fluid is then determined from the enthalpy difference between the two reservoirs and the cooling requirement of the column condenser and the heating requirement of the column reboiler.

Another option is to use a steam turbine to drive the compressor. This increases the cost of the compressor and drive by a factor of 1.15, but the energy cost is reduced.

Economic Parameters

The return on investment for each option was determined using a discounted cash flow routine. The annual revenue normally included in a discounted cash flow analysis was replaced with the utility savings from the use of the heat pump system. The annual cash expense was replaced with the additional annual maintenance cost. A 10 percent investment tax credit was taken with a federal tax rate of 46 percent. The Accelerated Cost Recovery System (ACRS) was used in determining the annual depreciation.[25]

CHAPTER X

FEASIBILITY GUIDELINES

In this chapter the effects of six column variables are investigated. The six variables are the reboiler heat duty (Q_{reb}), the ratio of the condenser heat duty to that of the reboiler (Q_{ratio}), the condenser temperature on the process side (T), the temperature difference on the process side between the reboiler and the condenser (DT), the electric rate (Er) forecast assumed either high, medium, or low, and the utility ratio which is the ratio of the natural gas rate to the electric rate on a dollars per million btu basis (Ur).

The utility ratio was used as a measure of economic feasibility. The utility ratio was allowed to vary until the net present worth of the heat pump system including maintenance and utility expenses over the lifespan of the system was equal to that of the conventional (no heat pump) system. Therefore, if the actual utility ratio is greater than that obtained, then the application of the heat pump system is economically favorable (see Figure 11).

The results of this study are shown in Figures 12 through 53. Two different working fluids were used to show to a limited extent the effects of the choice of the working fluid. The two fluids are R-114 and R-113.

The effects of the reboiler heat duty is shown in Figures 12 through 17. The benefits of the heat pump system increase with

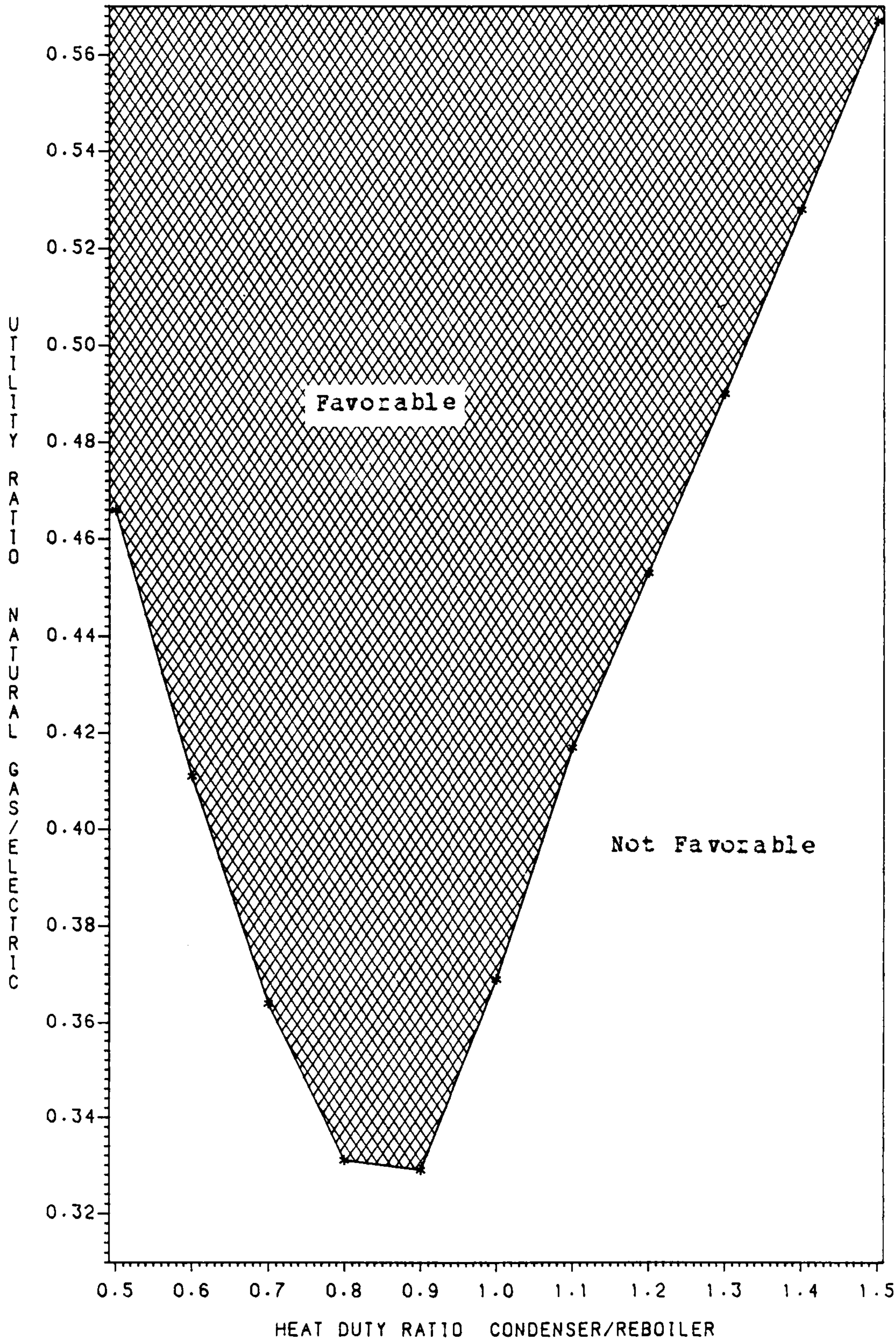

Figure 11. Example graph.

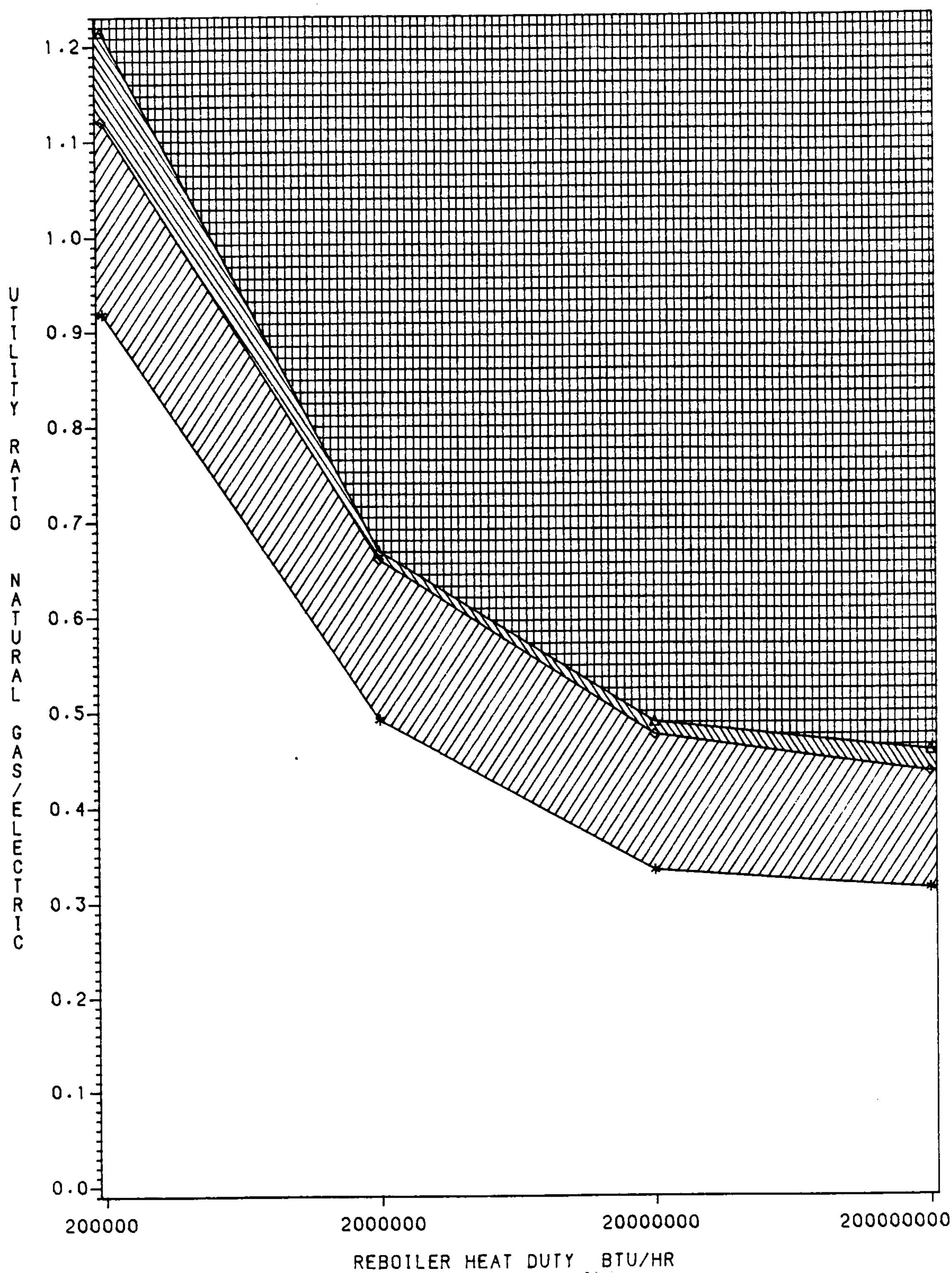

Figure 12. Reboiler heat duty vs. utility ratio for R-114 with INT = 10.0%, DT = 10 R, and T = 560 R; ER = 1986 low TVA load forecast.

Diamonds: Q ratio = 0.5; Stars: Q ratio = 1.0; Triangles: Q ratio = 1.5.

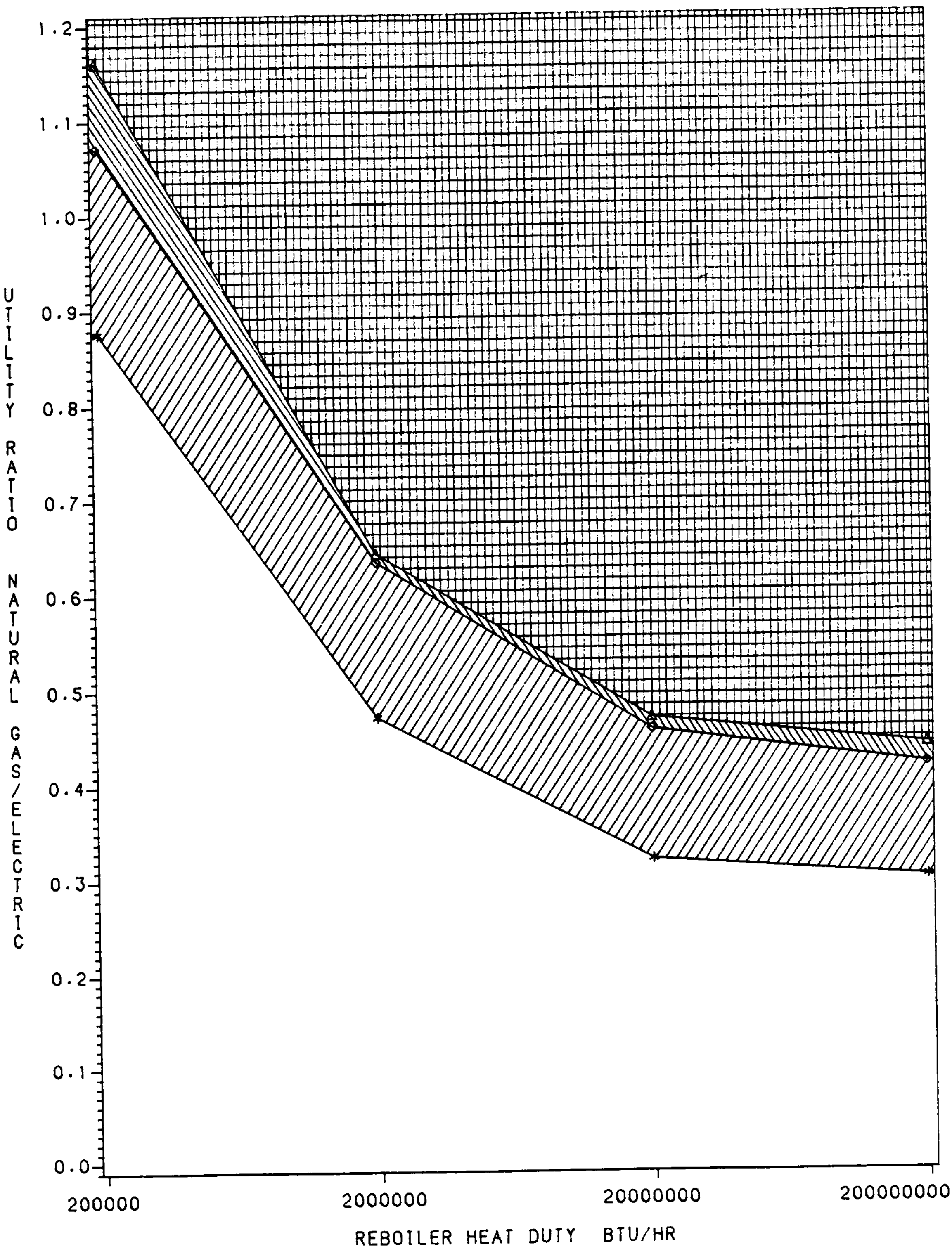

Figure 13. Reboiler heat duty vs. utility ratio for R-114 with INT - 10.0%, DT = 10 R, and T = 560 R; ER = 1986 medium TVA load forecast.

Diamonds: Q ratio = 0.5; Stars: Q ratio = 1.0; Triangles: Q ratio = 1.5.

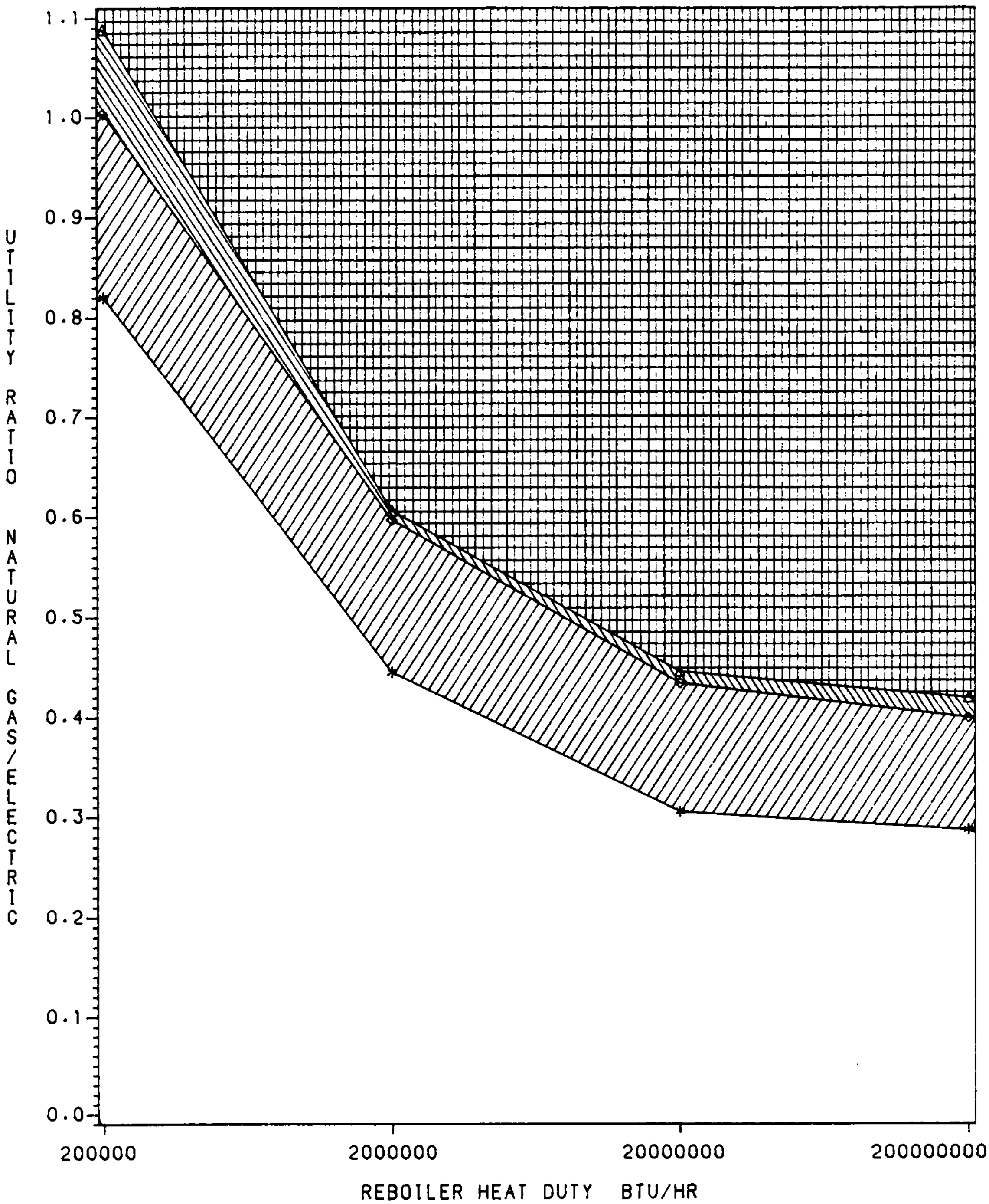

Figure 14. Reboiler heat duty vs. utility ratio for R-114 with INT = 10.0%, DT = 10 R, and T = 560 R; ER = 1986 high TVA load forecast.

Diamonds: Q ratio = 0.5; Stars: Q ratio = 1.0; Triangles: Q ratio = 1.5.

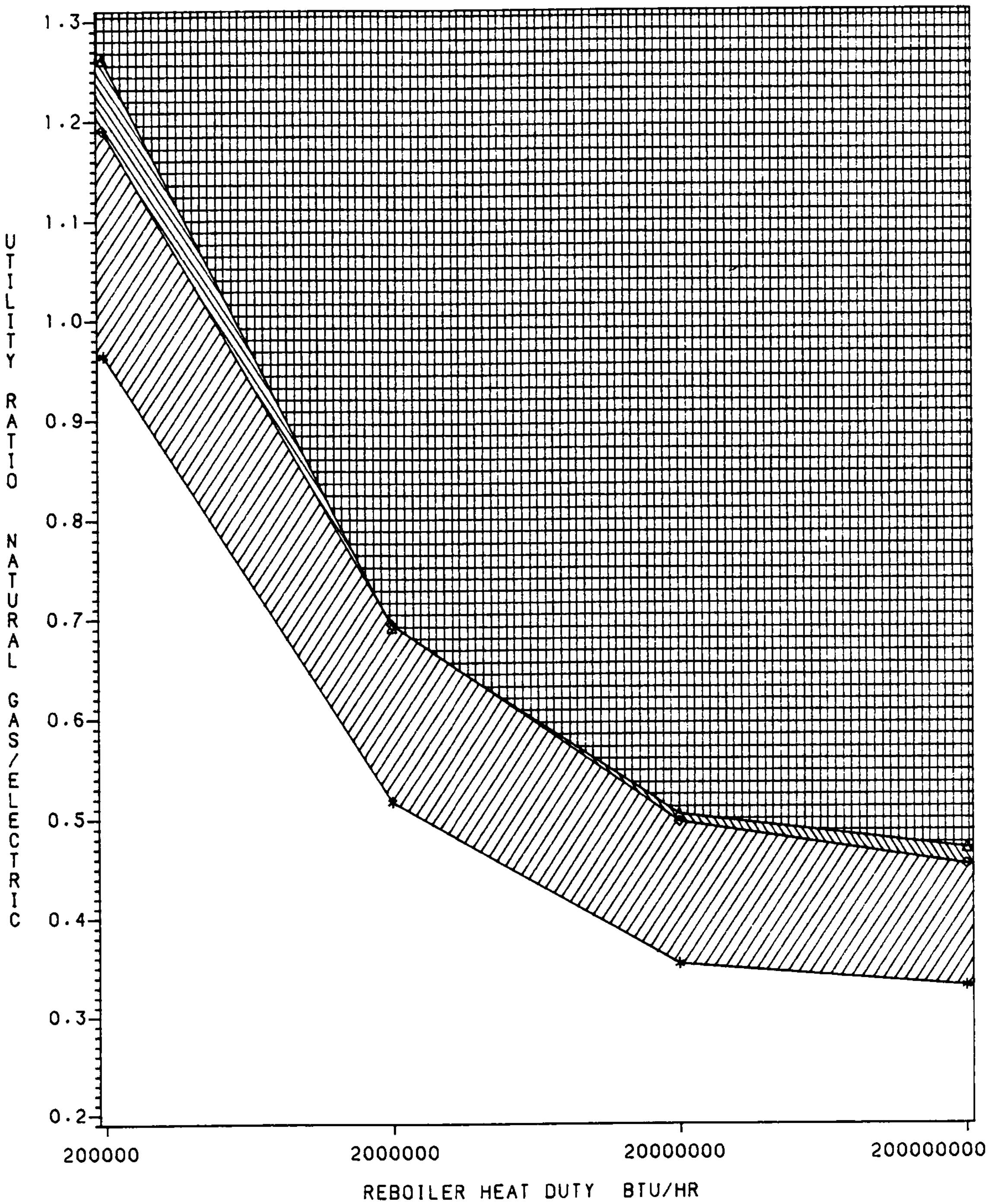

Figure 15. Reboiler heat duty vs. utility ratio for R-113 with INT = 10.0%, DT = 10 R, and T = 610 R; ER = 1986 low TVA load forecast.

Diamonds: Q ratio = 0.5; Stars: Q ratio = 1.0; Triangles: Q ratio = 1.5.

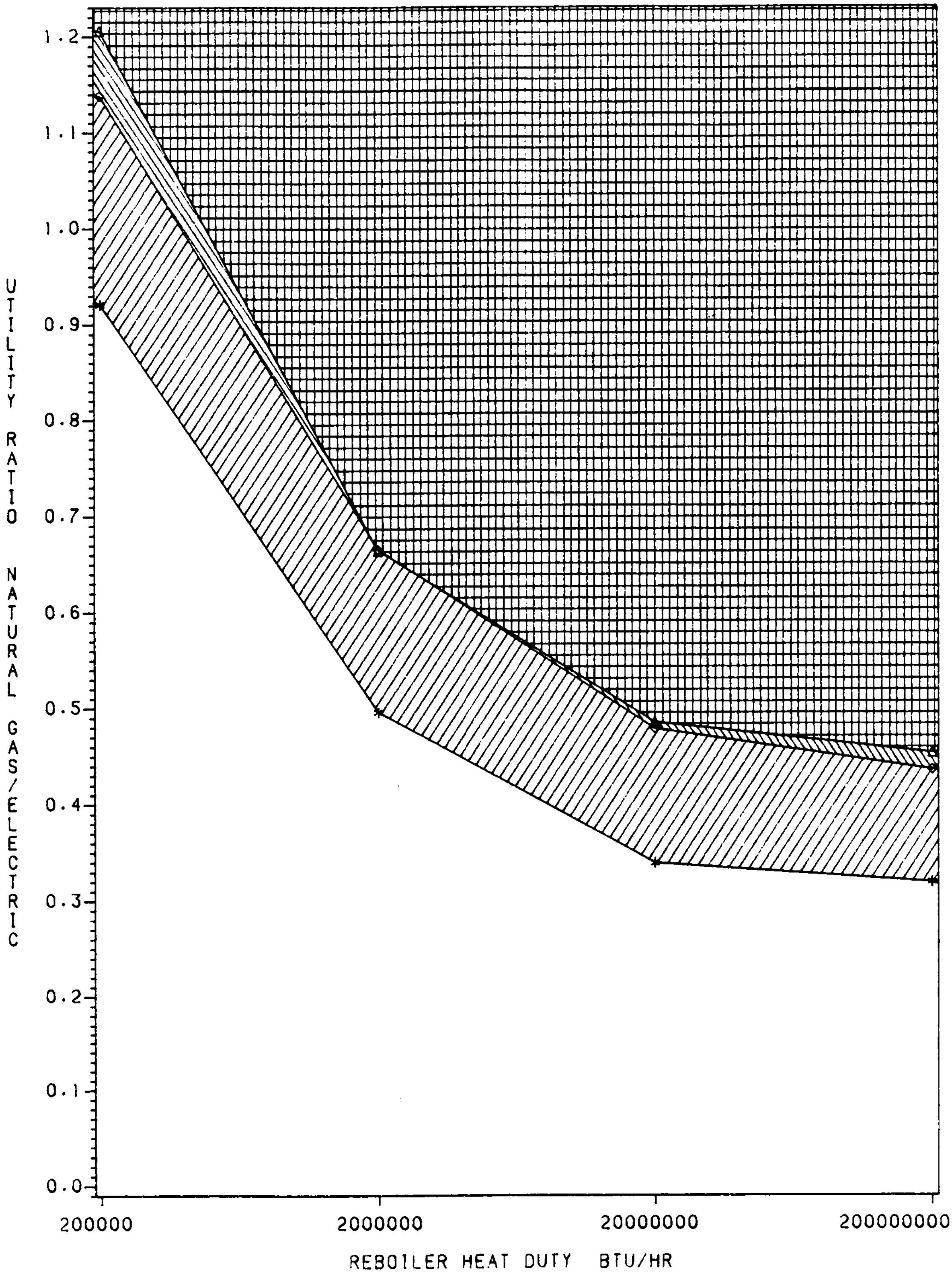

Figure 16. Reboiler heat duty vs. utility ratio for R-113 with INT = 10.0%, DT = 10 R, and T = 610 R; ER = 1986 medium TVA load forecast.

Diamonds: Q ratio = 0.5; Stars: Q ratio = 1.0; Triangles: Q ratio = 1.5.

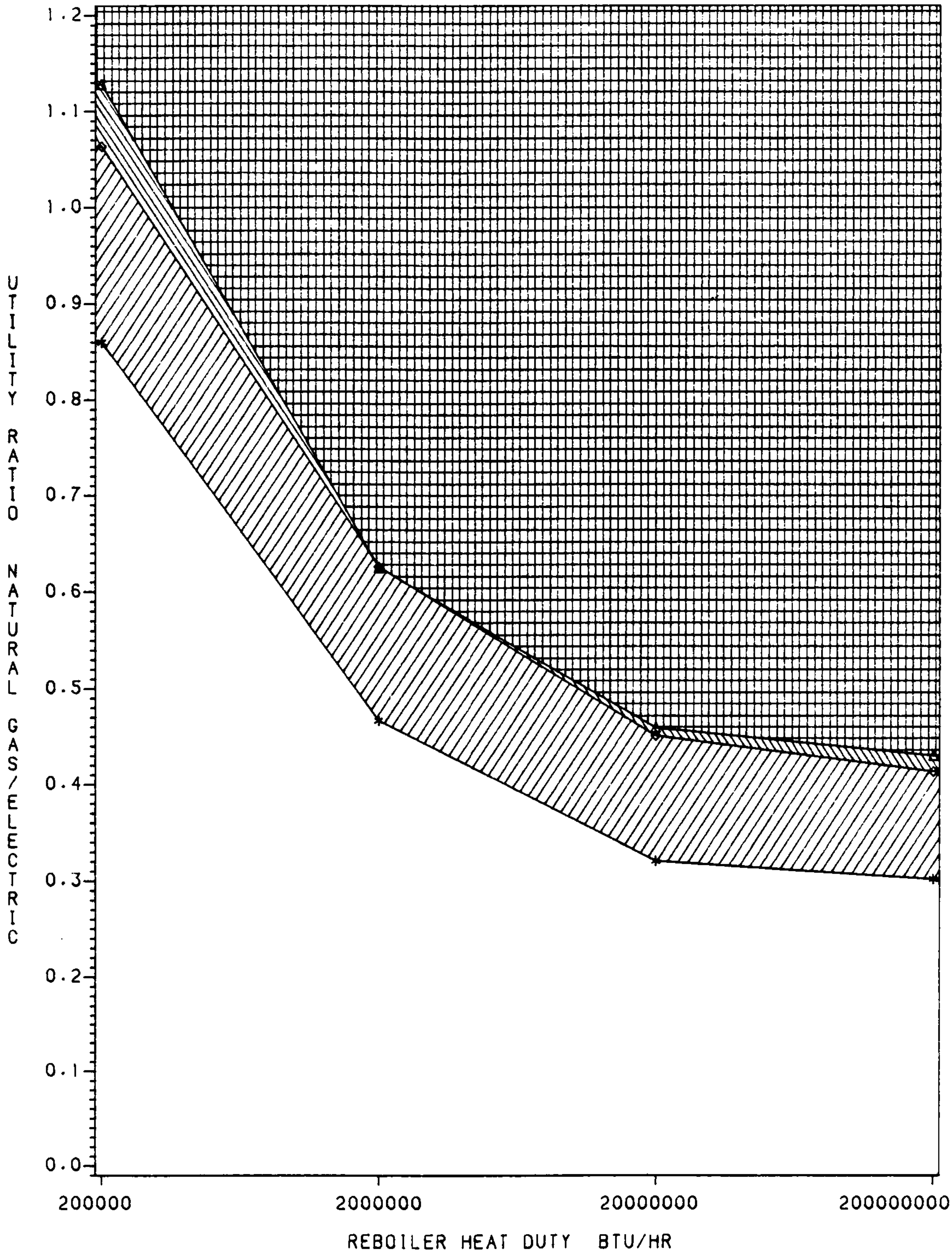

Figure 17. Reboiler heat duty vs. utility ratio for R-113 with INT = 10.0%, DT = 10 R, and T = 610 R; ER = 1986 high TVA load forecast.

Diamonds: Q ratio = 0.5; Stars: Q ratio = 1.0; Triangles: Q ratio = 1.5.

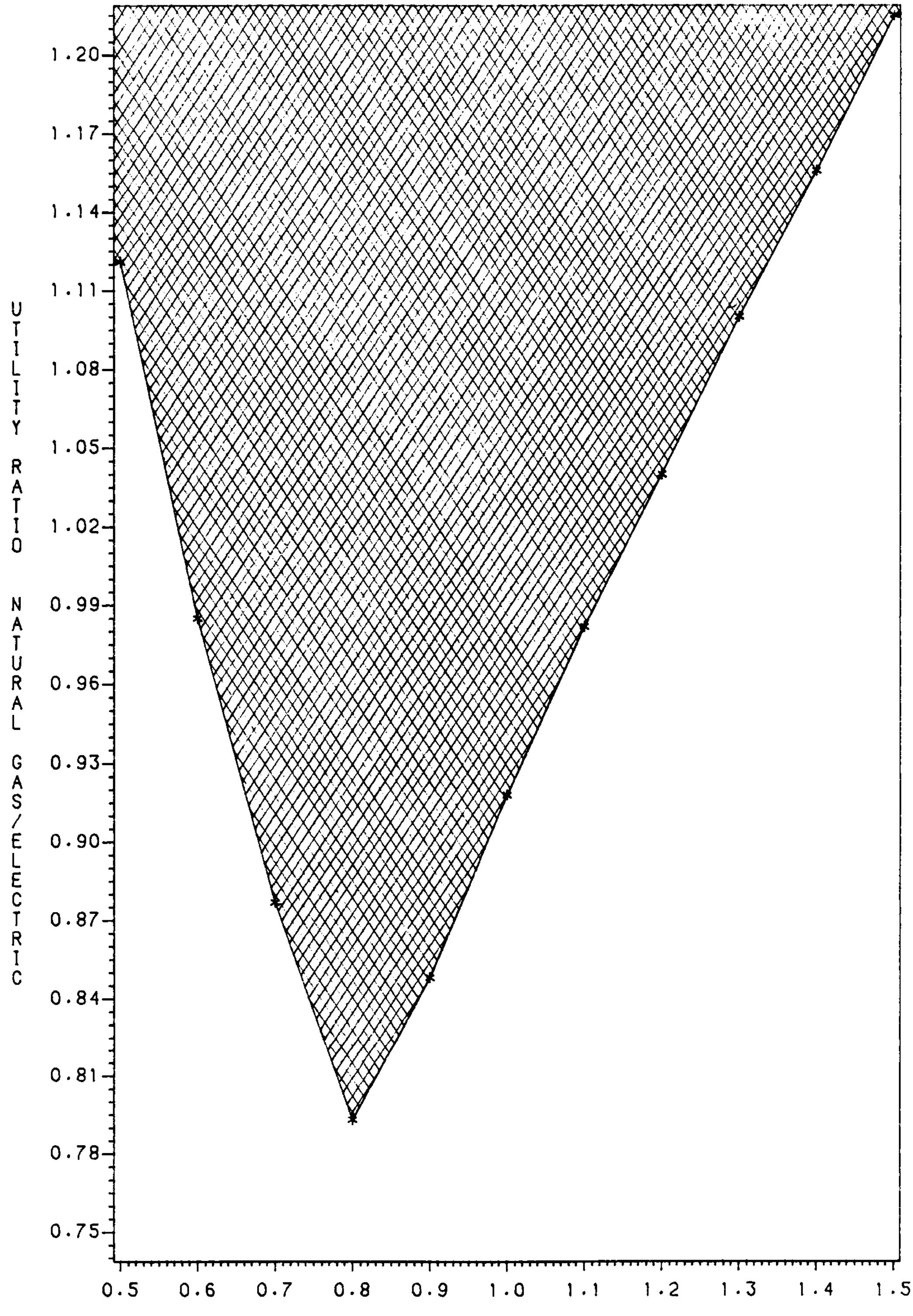

Figure 18. Heat duty ratio vs. utility ratio for R-114 with INT = 10.0%, DT = 10 R, T = 560 R, and QREB = 200,000 BTU/hr; ER = 1986 low TVA load forecast.

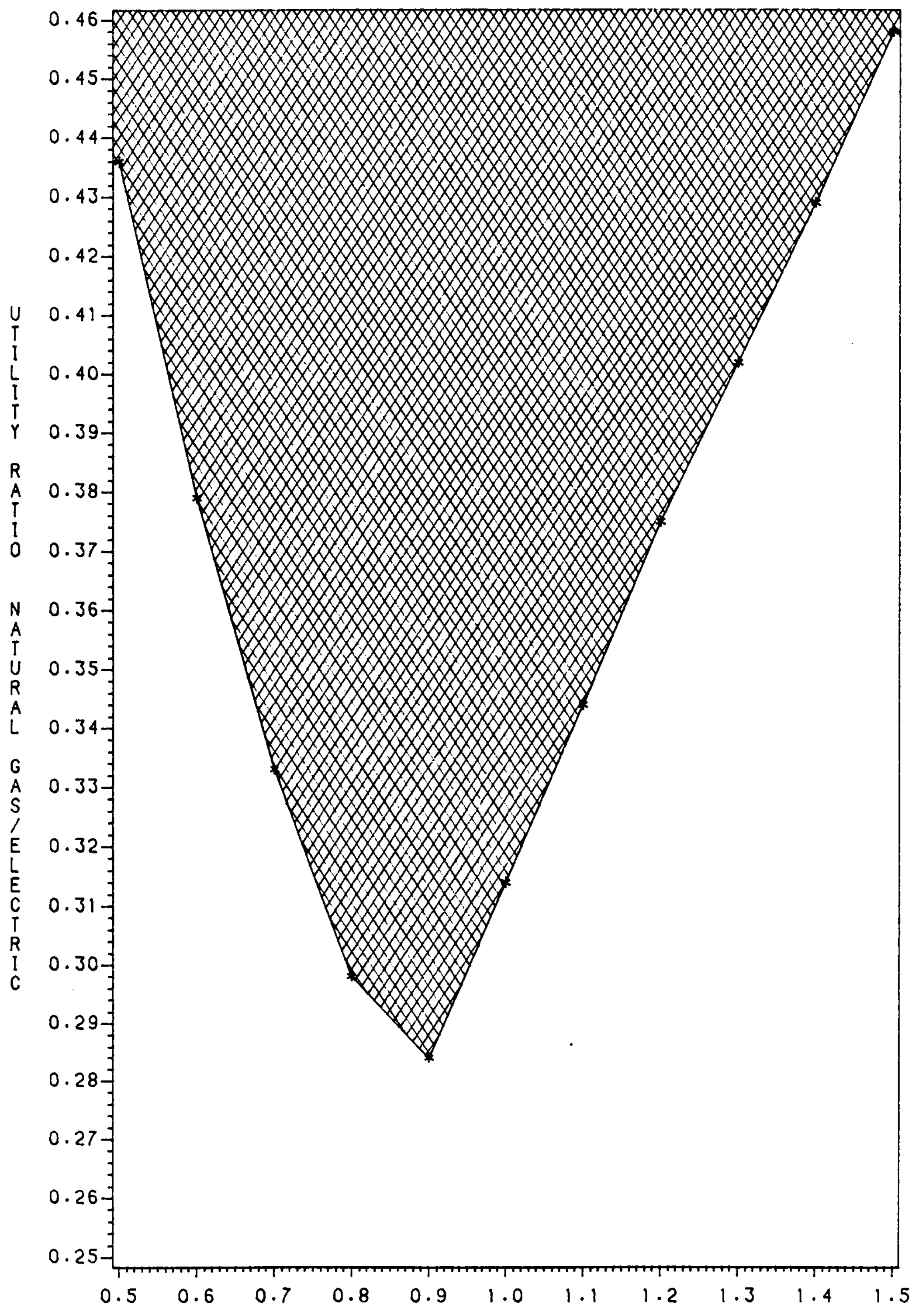

Figure 19. Heat duty ratio vs. utility ratio for R-114 with INT = 10.0%, DT = 10 R, T = 560 R, and QREB = 200,000,000 BTU/hr; ER = 1986 low TVA load forecast.

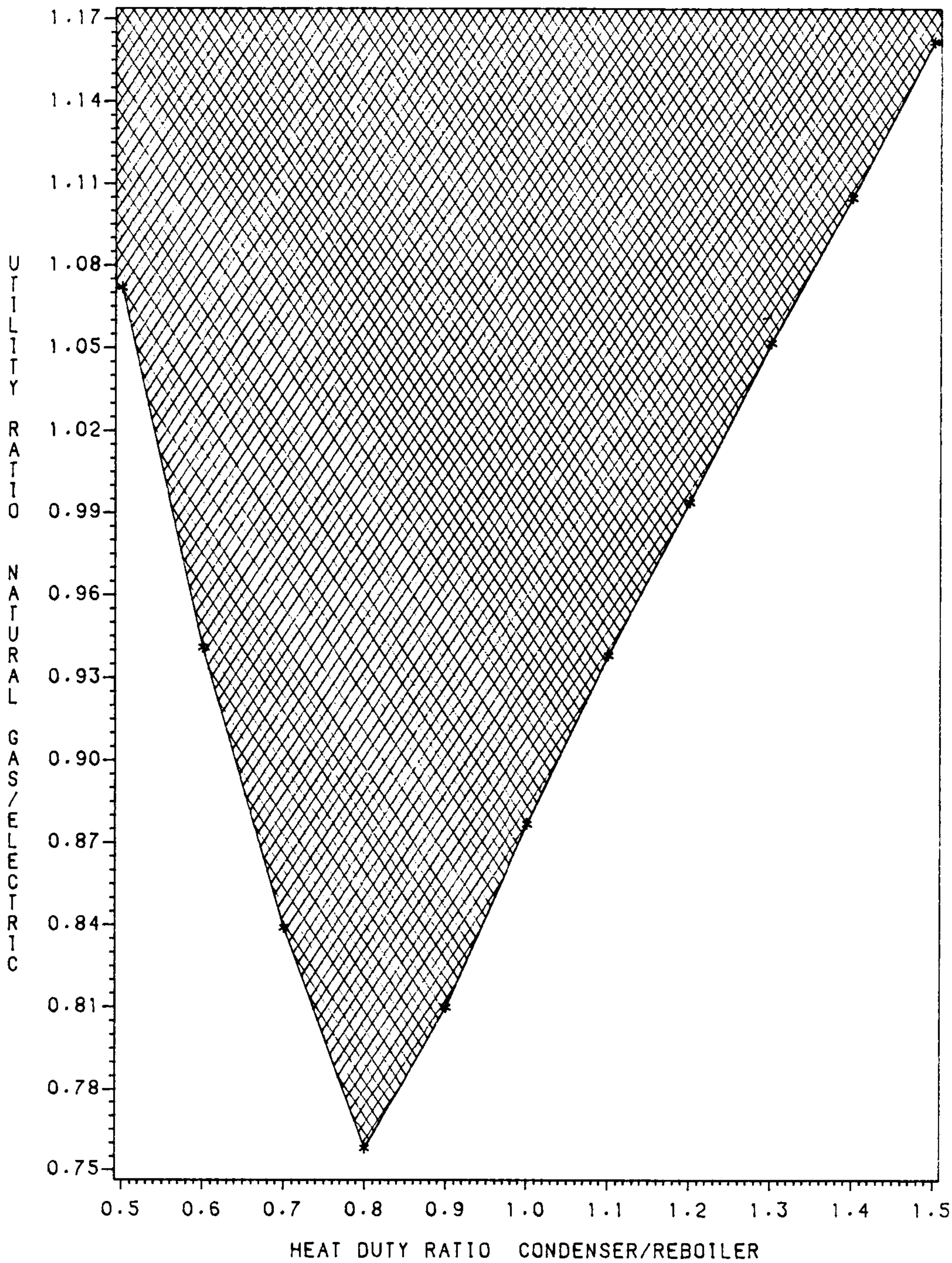

Figure 20. Heat duty ratio vs. utility ratio for R-114 with INT = 10.0%, DT = 10 R, T = 560 R, and QREB = 200,000 BTU/hr; ER = 1986 medium TVA load forecast.

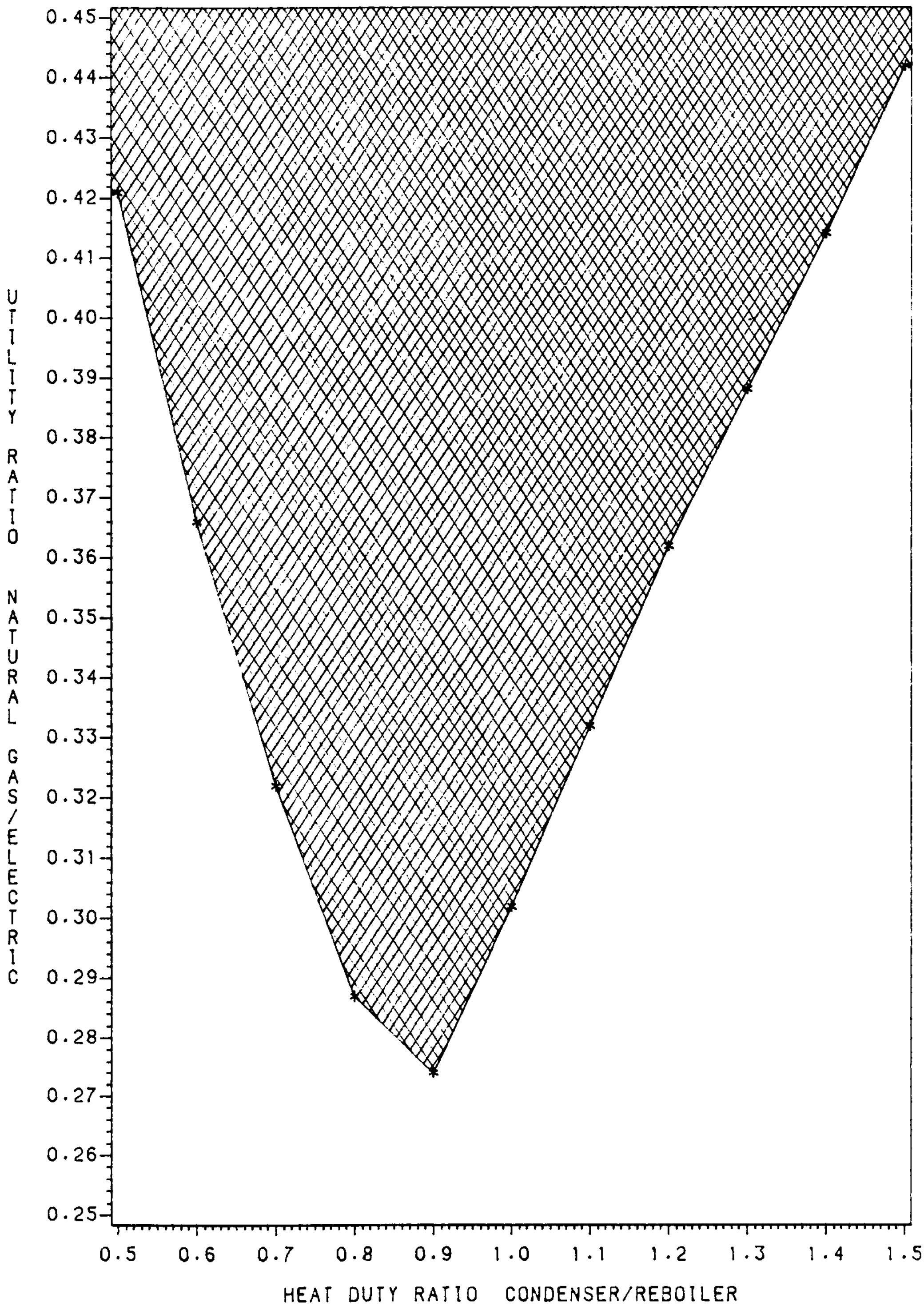

Figure 21. Heat duty ratio vs. utility ratio for R-114 with INT = 10.0%, DT = 10 R, T = 560 R, and QREB = 200,000,000 BTU/hr; ER = 1986 medium TVA load forecast.

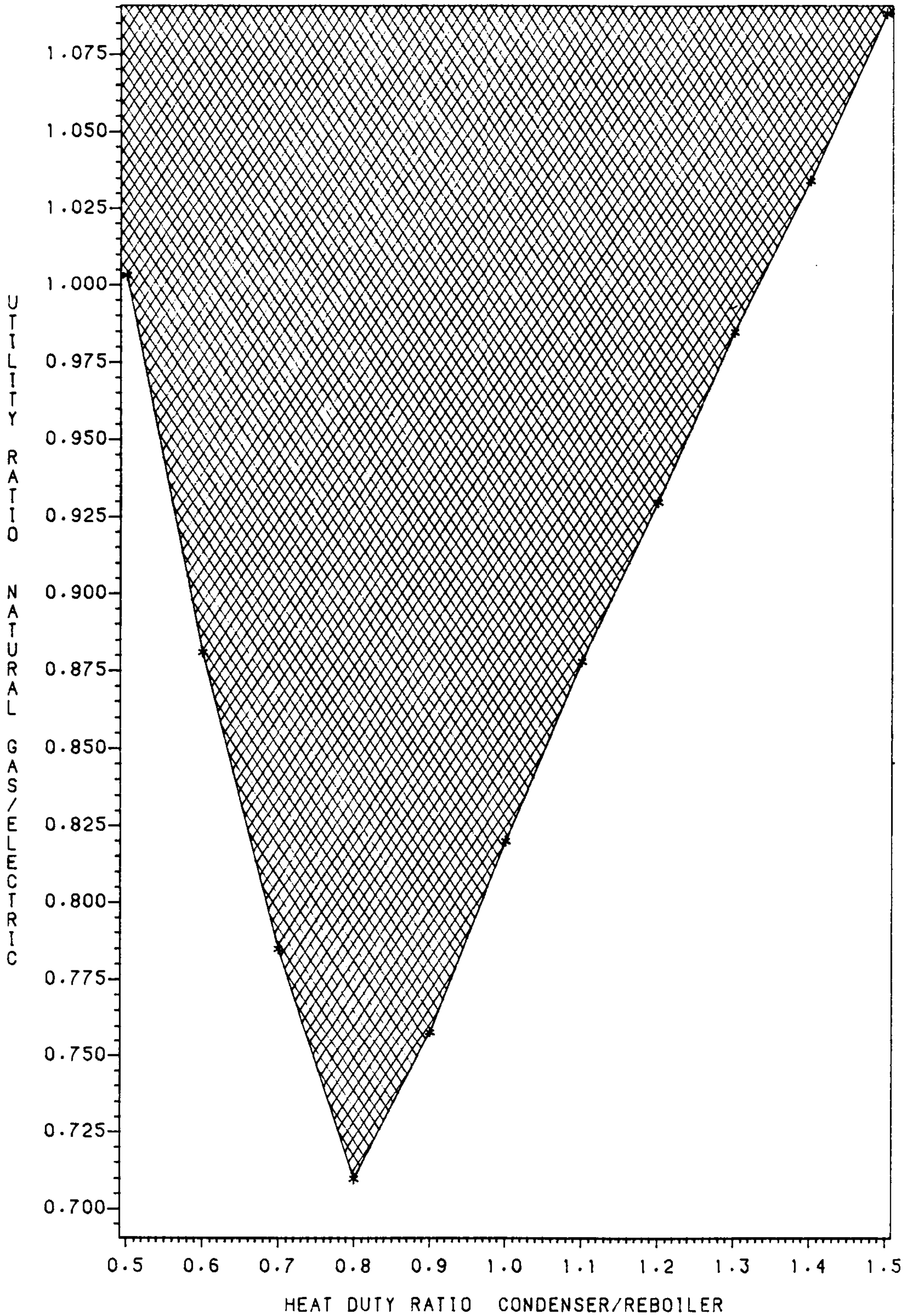

Figure 22. Heat duty ratio vs. utility ratio for R-114 with INT = 10.0%, DT = 10 R, T = 560 R, and QREB = 200,000 BTU/hr; ER = 1986 high TVA load forecast.

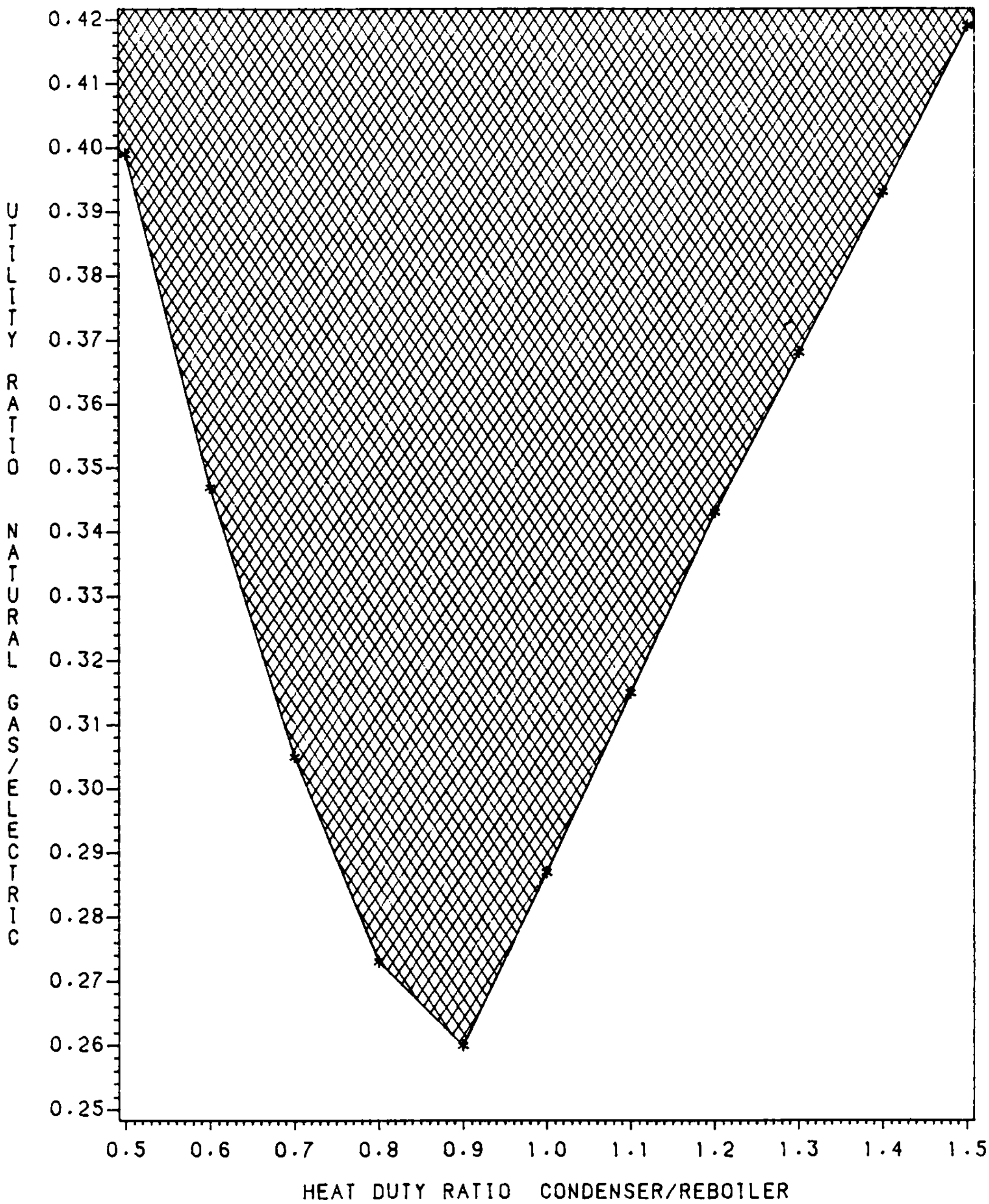

Figure 23. Heat duty ratio vs. utility ratio for R-114 with INT = 10.0%, DT = 10 R, T = 560 R, and QREB = 200,000,000 BTU/hr; ER = 1986 high TVA load forecast.

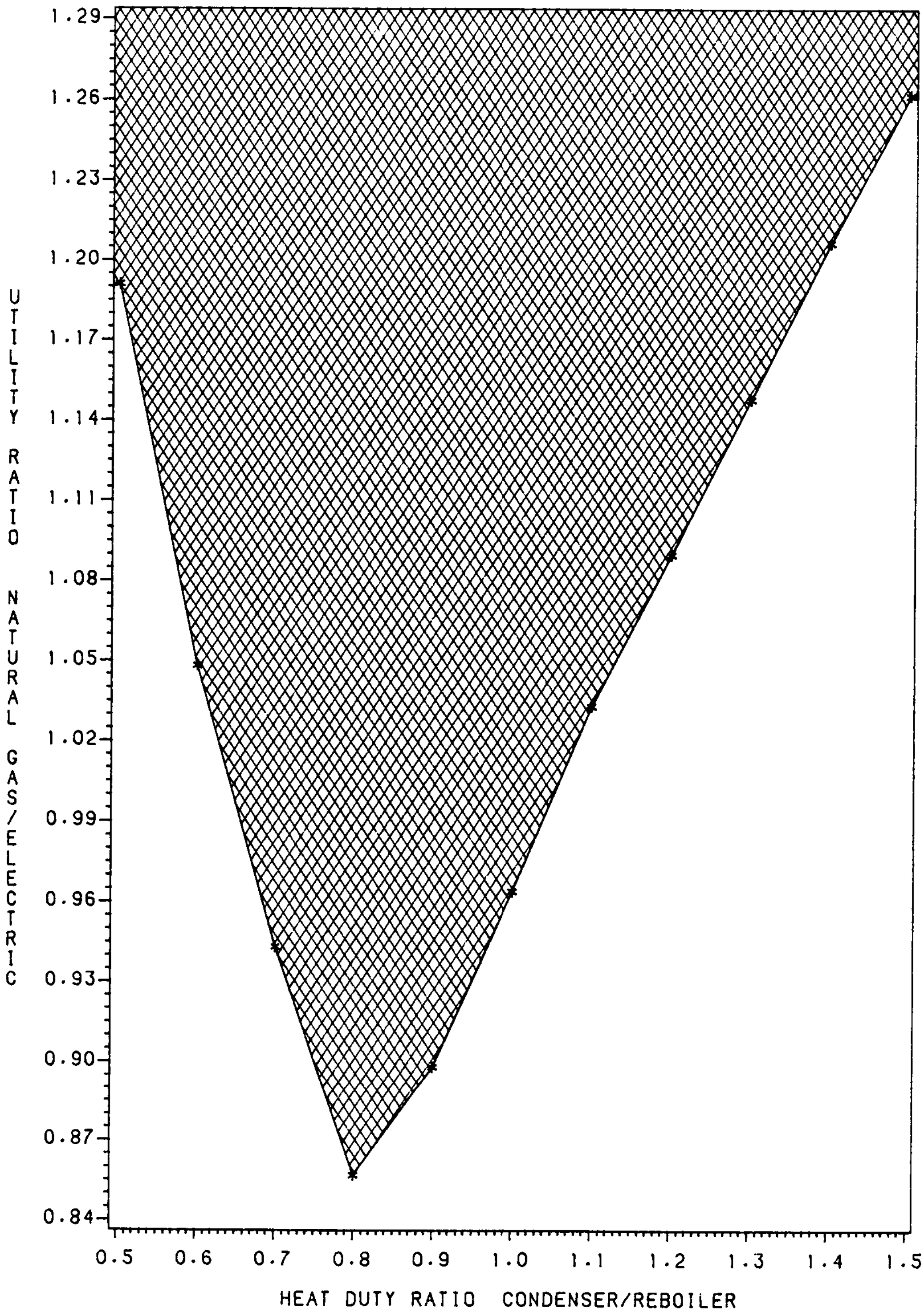

Figure 24. Heat duty ratio vs. utility ratio for R-113 with INT = 10.0%, DT = 10 R, T = 610 R, and QREB = 200,000 BTU/hr; ER = 1986 low TVA load forecast.

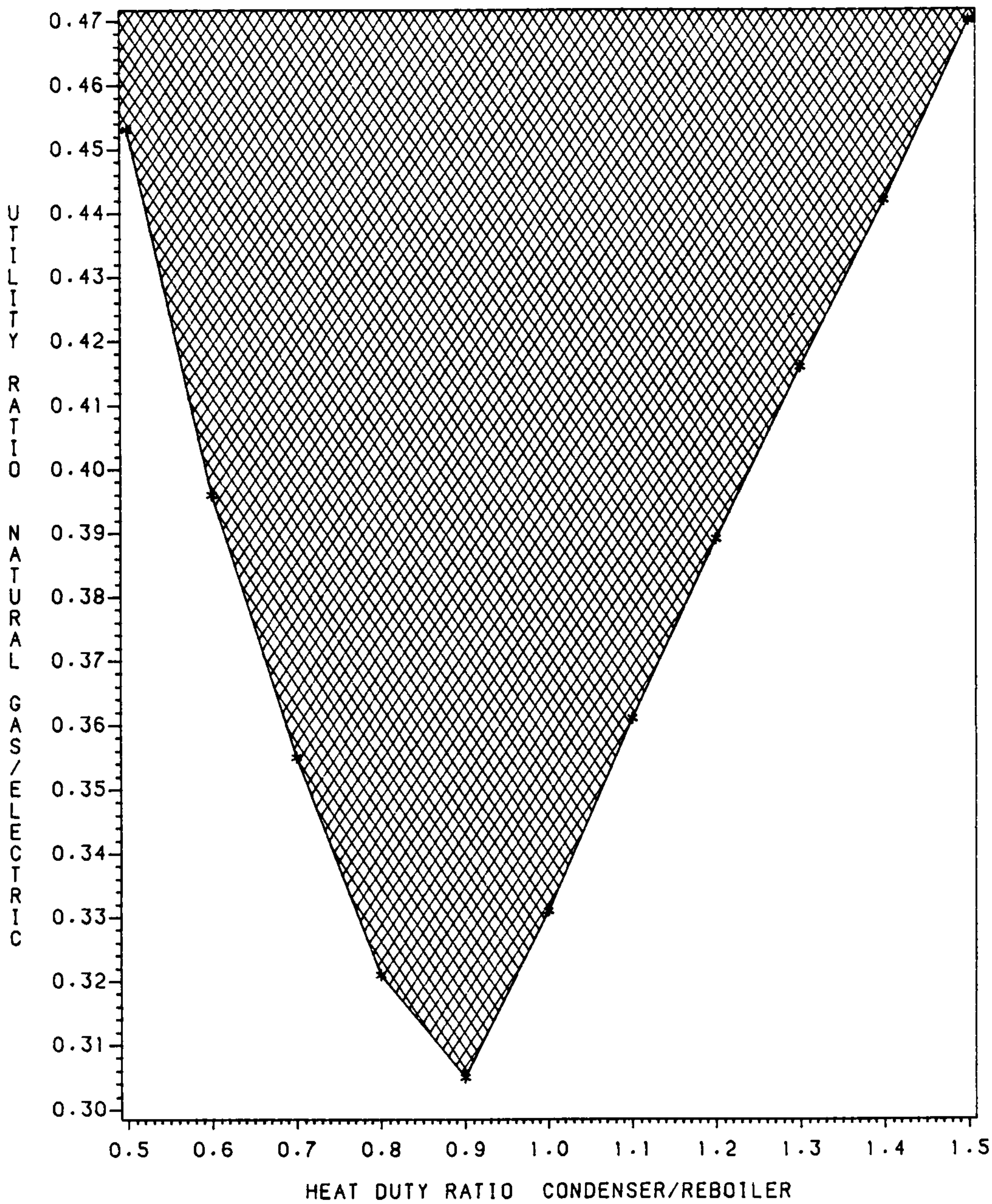

Figure 25. Heat duty ratio vs. utility ratio for R-113 with INT = 10.0%, DT = 10 R, T = 610 R, and QREB = 200,000,000 BTU/hr; ER = 1986 low TVA load forecast.

Star: without energy recovery; Square: with energy recovery.

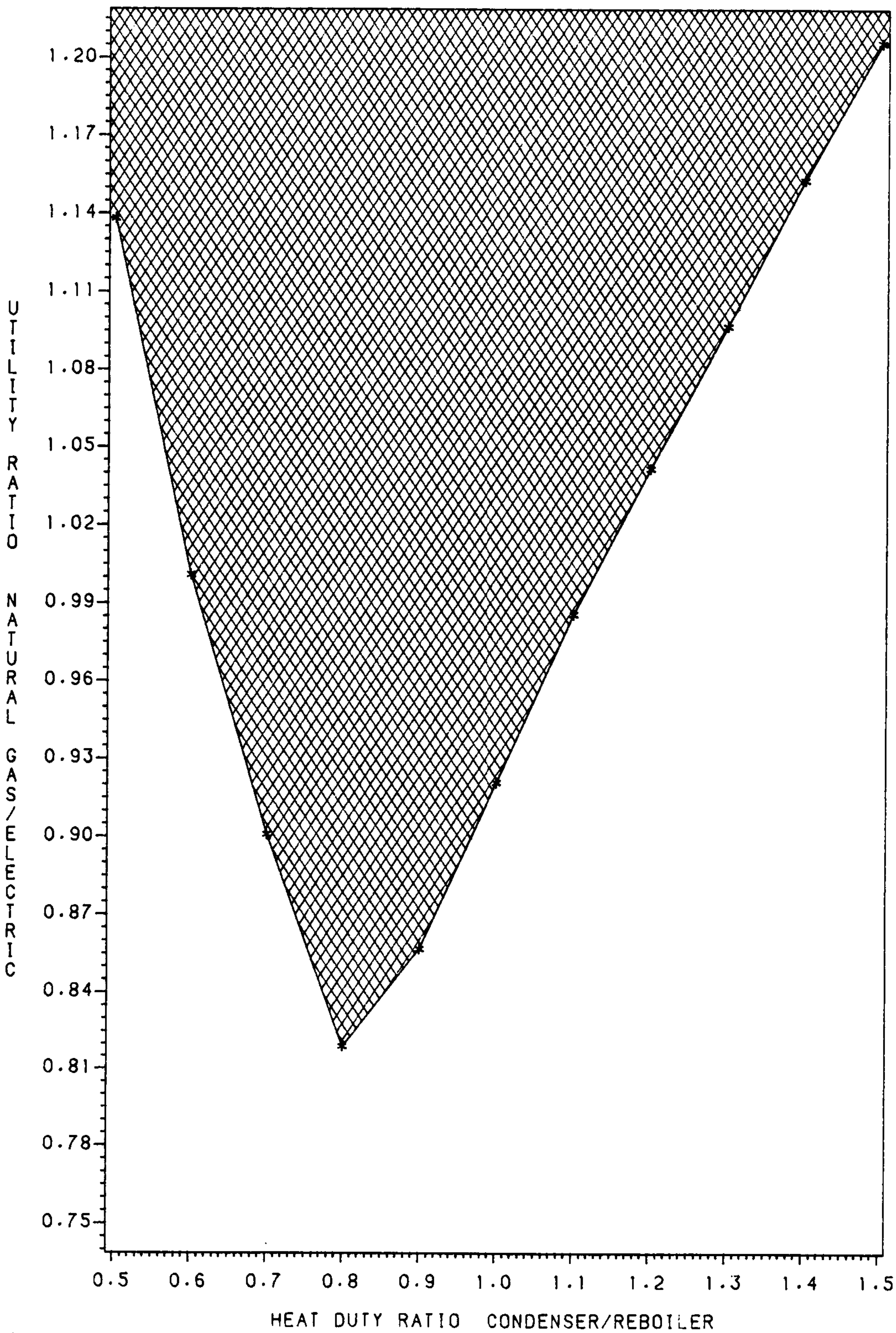

Figure 26. Heat duty ratio vs. utility ratio for R-113 with INT = 10.0%, DT = 10 R, T = 610 R, and QREB = 200,000 BTU/hr; ER = 1986 medium TVA load forecast.

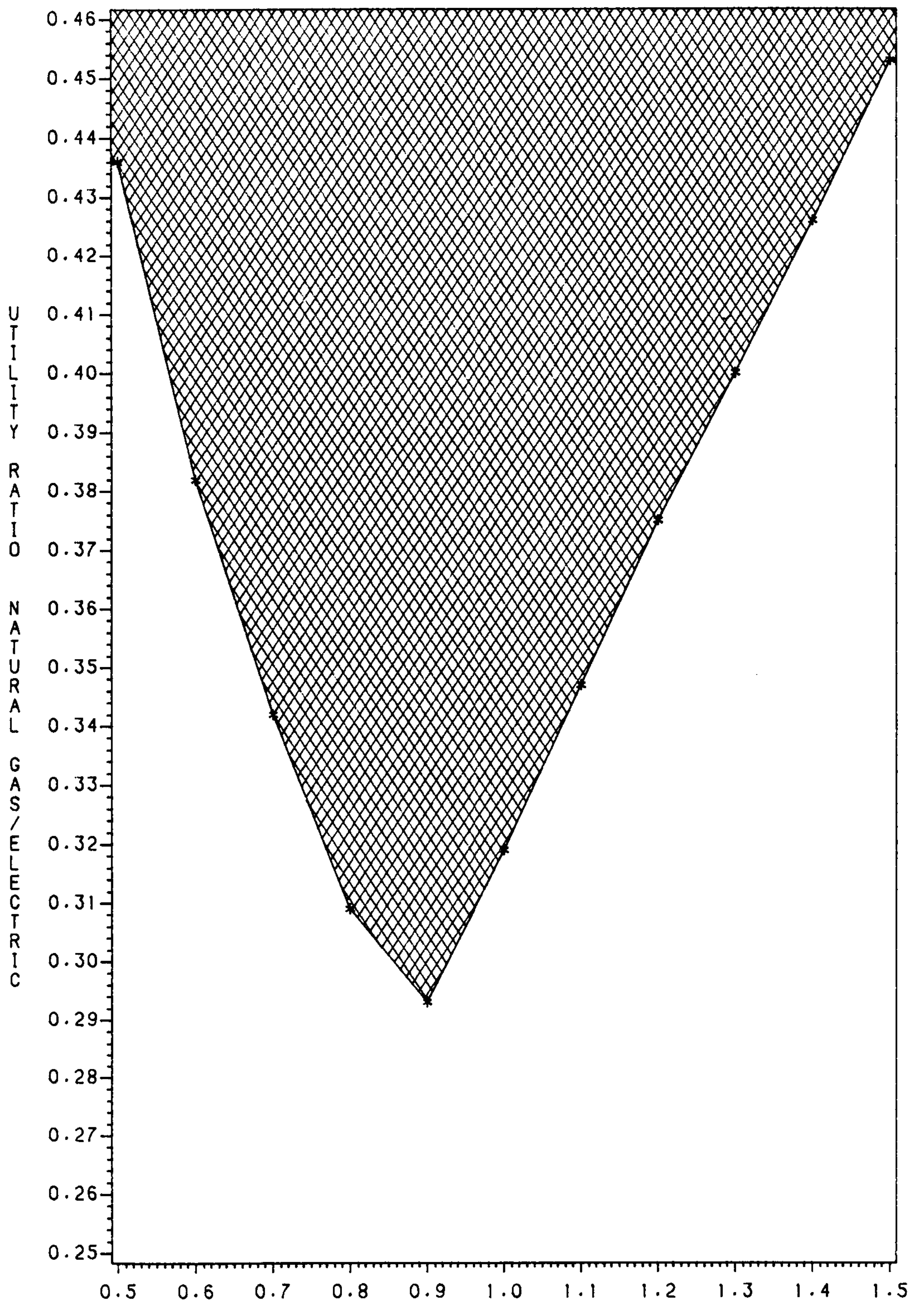

Figure 27. Heat duty ratio vs. utility ratio for R-113 with INT = 10.0%, DT = 10 R, T = 610 R, and QREB = 200,000,000 BTU/hr; ER = 1986 medium TVA load forecast.

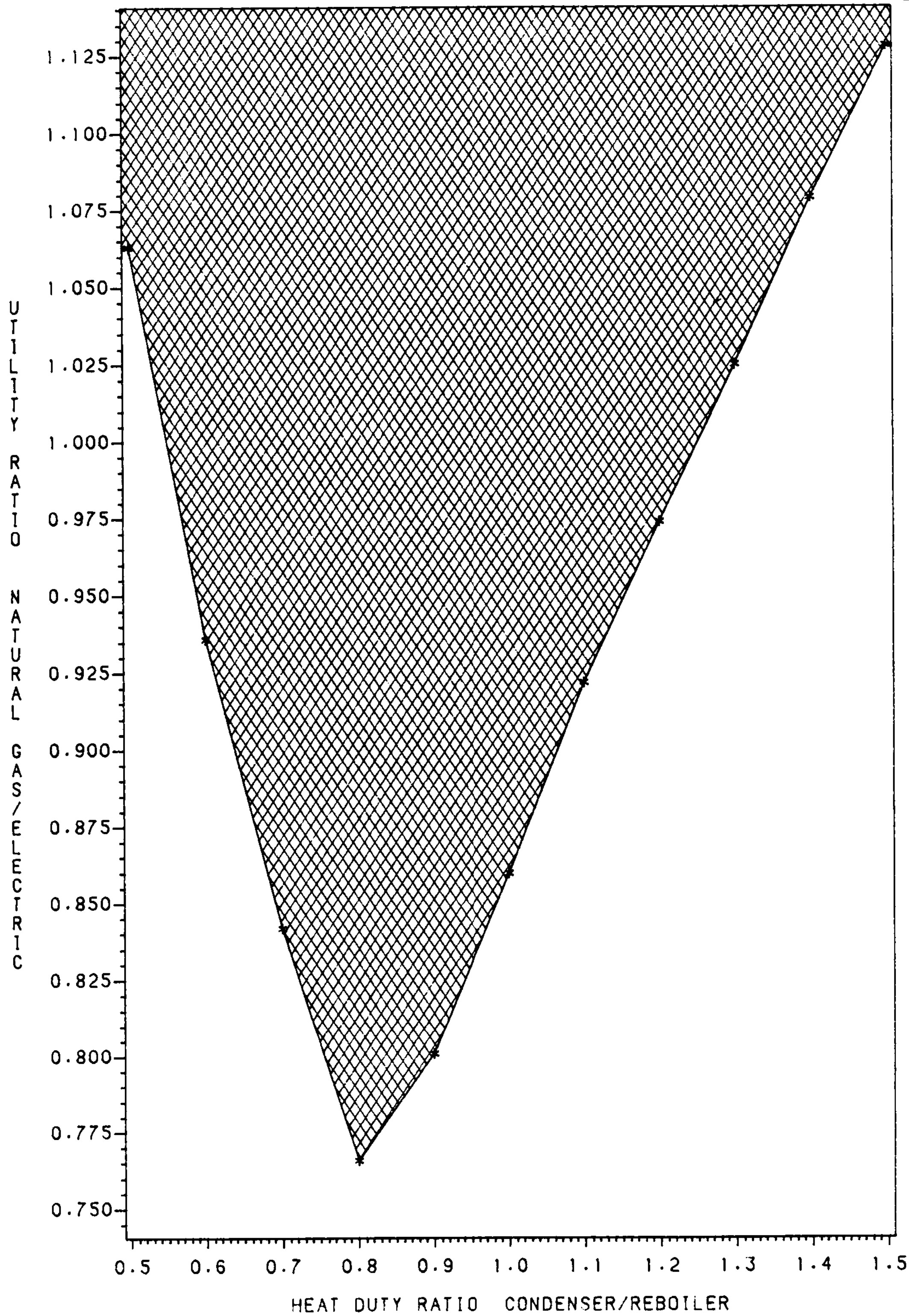

Figure 28. Heat duty ratio vs. utility ratio for R-113 with INT = 10.0%, DT = 10 R, T = 610 R, and QREB = 200,000 BTU/hr; ER = 1986 high TVA load forecast.

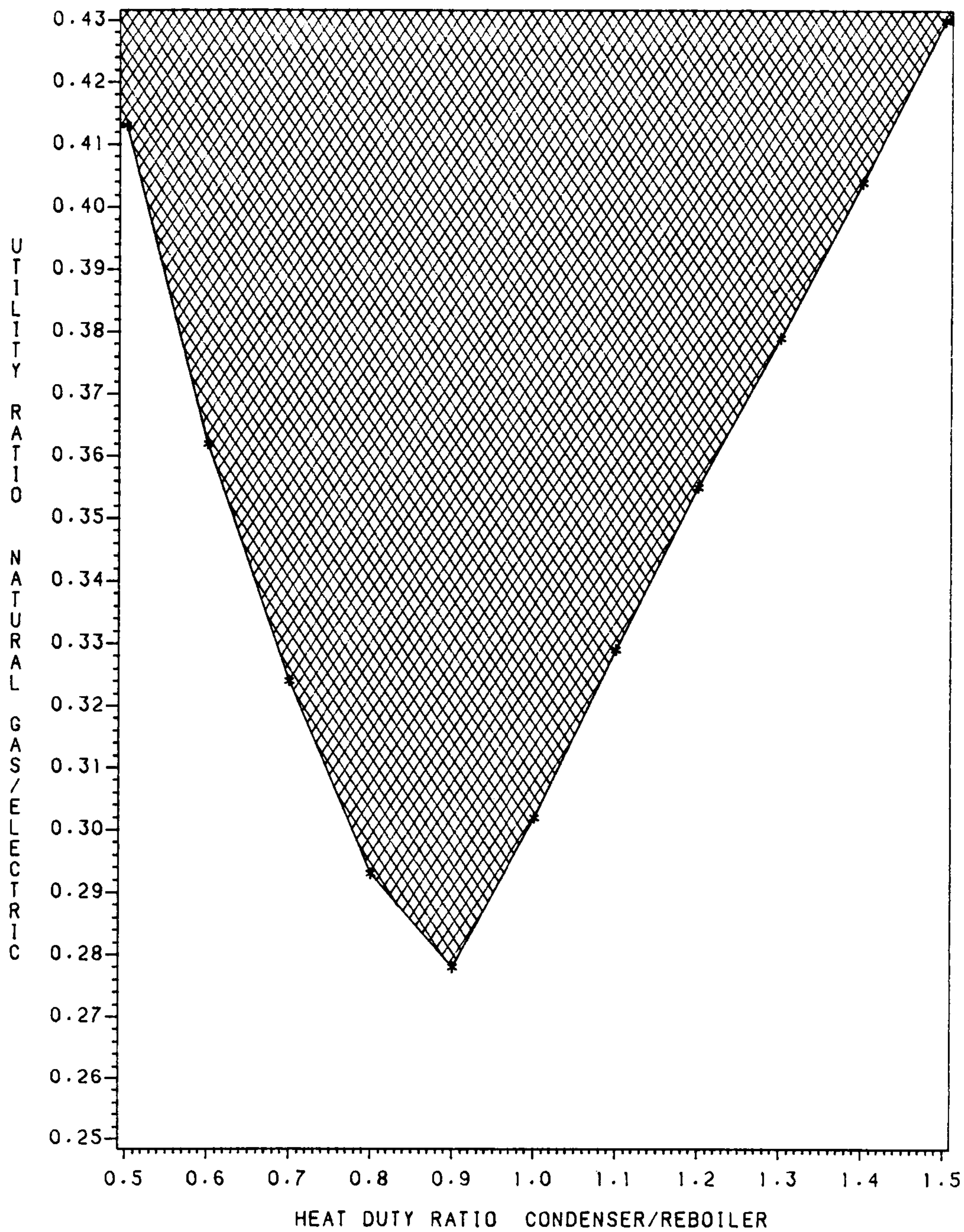

Figure 29. Heat duty ratio vs. utility ratio for R-113 with INT = 10.0%, DT = 10 R, T = 610 R, and QREB = 200,000,000 BTU/hr; ER = 1986 high TVA load forecast.

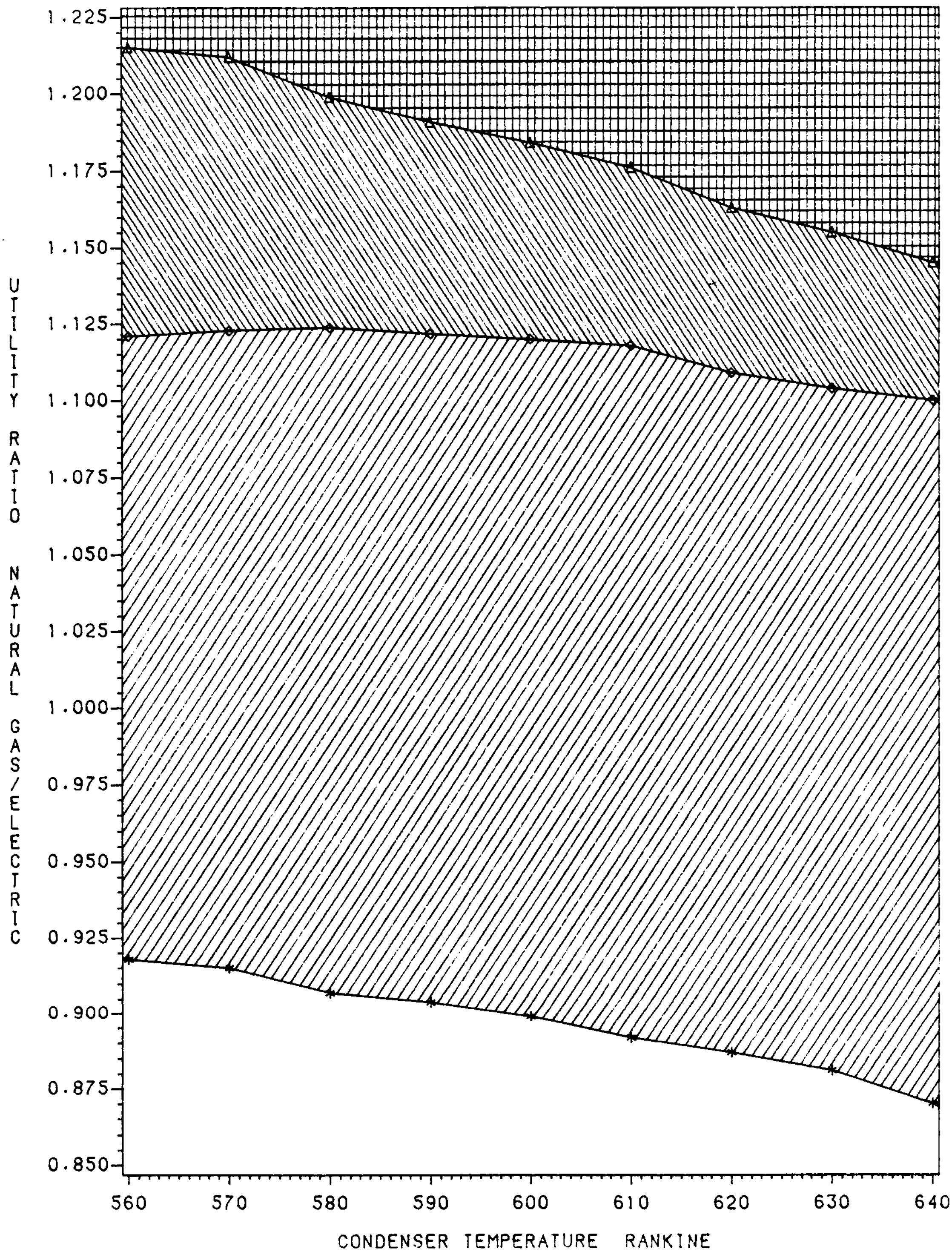

Figure 30. Condenser temperature vs. utility ratio for R-114 with INT = 10.0%, DT = 10 R, and QREB = 200,000 BTU/hr; ER = 1986 low TVA load forecast.

Diamonds: Q ratio = 0.5; Stars: Q ratio = 1.0; Triangles: Q ratio = 1.5.

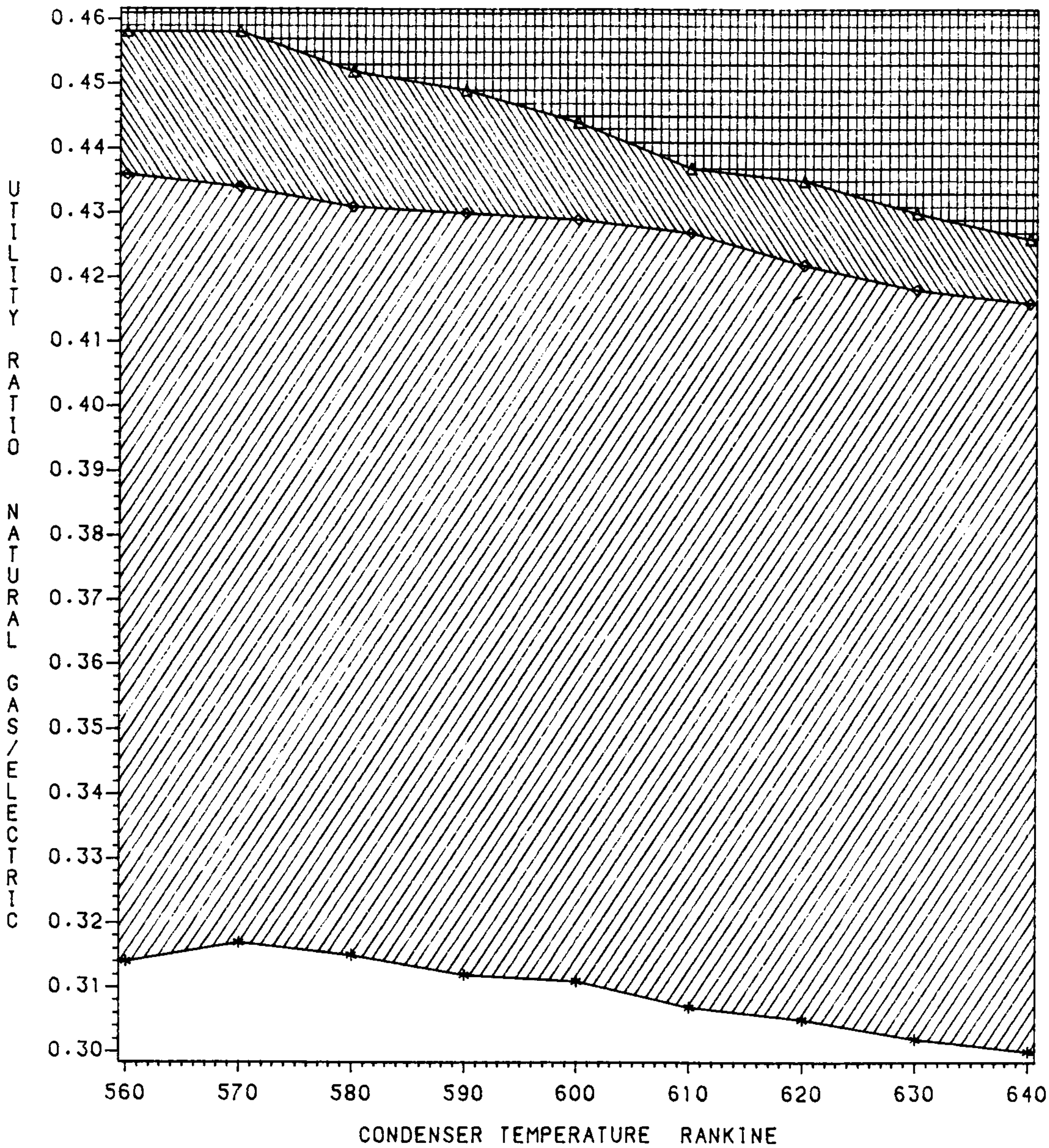

Figure 31. Condenser temperature vs. utility ratio for R-113 with INT = 10.0%, DT = 10 R, and QREB = 200,000,000 BTU/hr; ER = 1986 low TVA load forecast.

Diamonds: Q ratio = 0.5; Stars: Q ratio = 1.0; Triangles: Q ratio = 1.5.

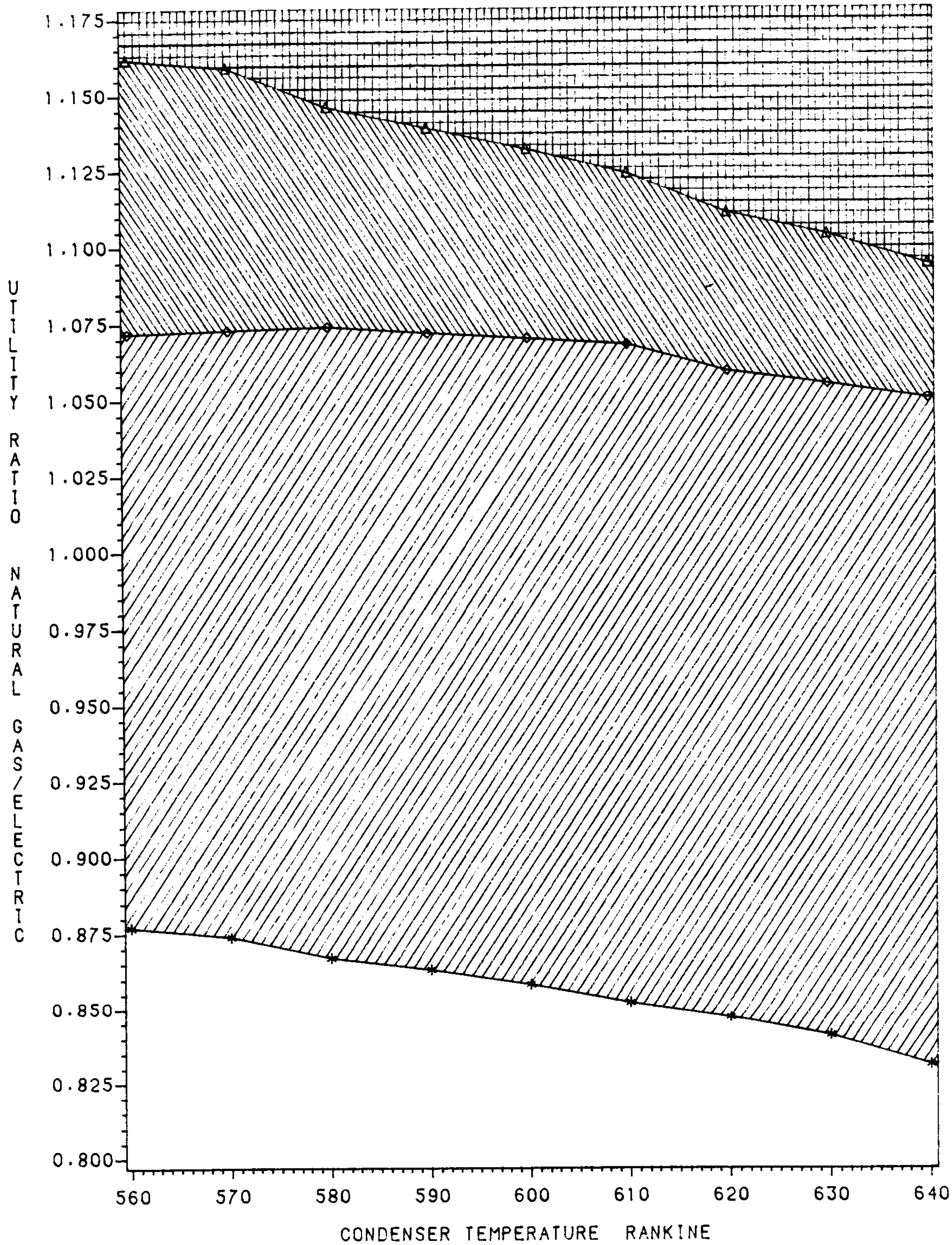

Figure 32. Condenser temperature vs. utility ratio for R-114 with INT = 10.0%, DT = 10 R, and QREB = 200,000 BTU/hr; ER = 1986 medium TVA load forecast.

Diamonds: Q ratio = 0.5; Stars: Q ratio = 1.0; Triangles: Q ratio = 1.5.

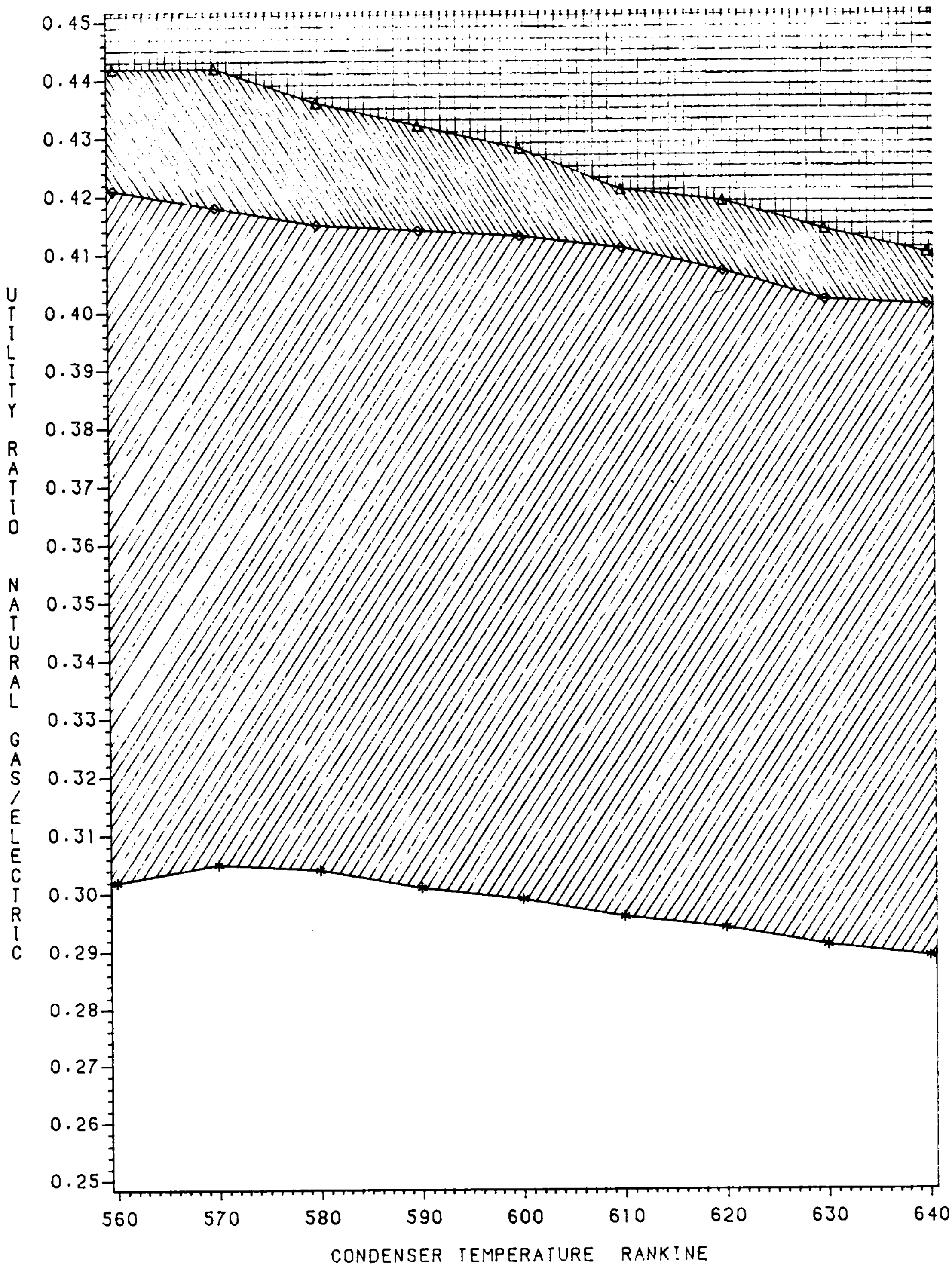

Figure 33. Condenser temperature vs. utility ratio for R-114 with INT = 10.0%, DT = 10 R, and QREB = 200,000,000 BTU/hr; ER = 1986 medium TVA load forecast.

Diamonds: Q ratio = 0.5; Stars: Q ratio = 1.0; Triangles: Q ratio = 1.5.

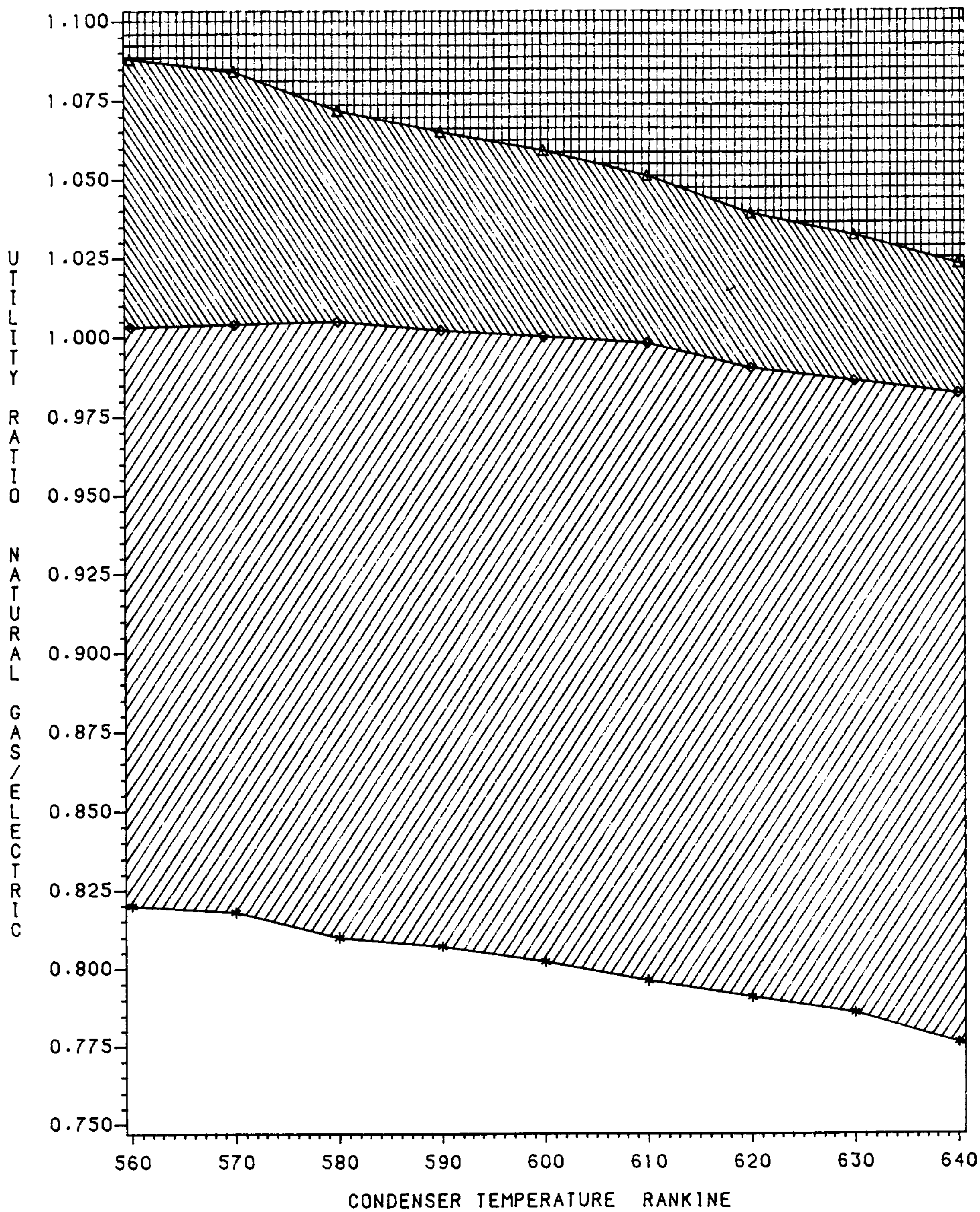

Figure 34. Condenser temperature vs. utility ratio for R-114 with INT = 10.0%, DT = 10 R, and QREB = 200,000 BTU/hr; ER = 1986 high TVA load forecast.

Diamonds: Q ratio = 0.5; Stars: Q ratio = 1.0; Triangles: Q ratio = 1.5.

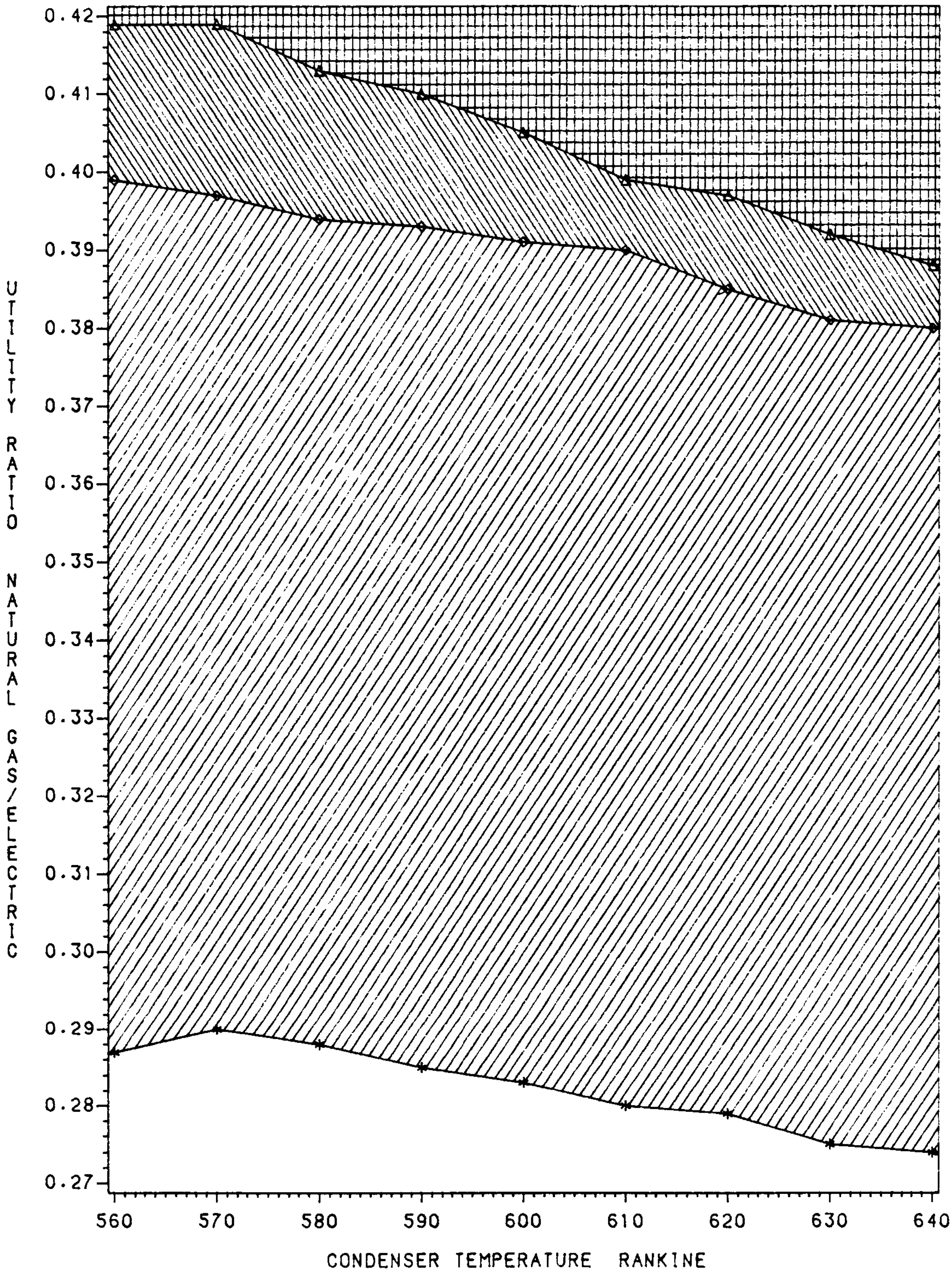

Figure 35. Condenser temperature vs. utility ratio for R-114 with INT = 10.0%, DT = 10 R, and QREB = 200,000,000 BTU/hr; ER = 1986 high TVA load forecast.

Diamonds: Q ratio = 0.5; Stars: Q ratio = 1.0; Triangles: Q ratio = 1.5.

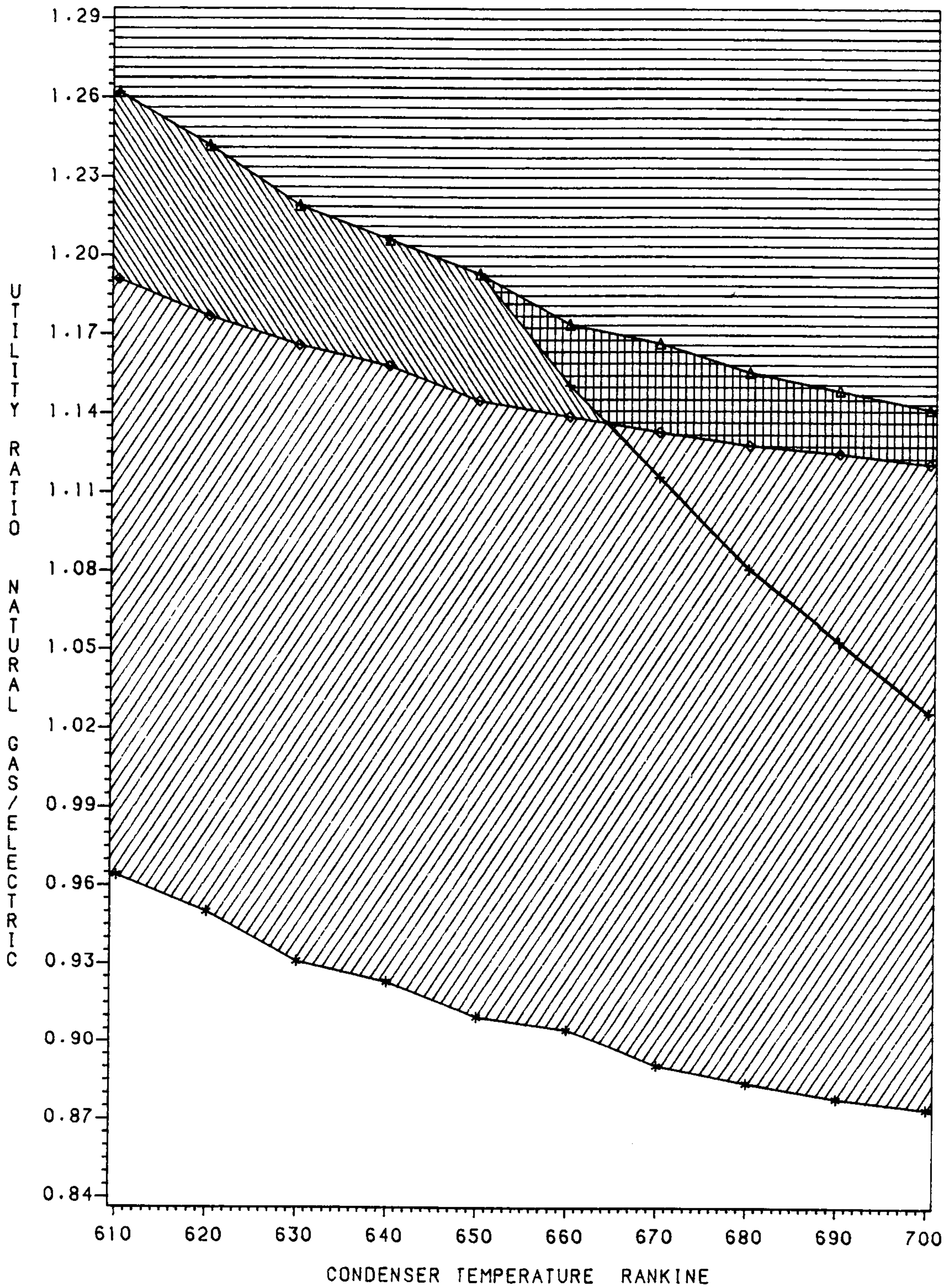

Figure 36. Condenser temperature vs. utility ratio for R-113 with INT = 10.0%, DT = 10 R, and QREB = 200,000 BTU/hr; ER = 1986 low TVA load forecast.

Diamonds: Q ratio = 0.5; Stars: Q ratio = 1.0; Triangles: Q ratio = 1.5; Pluses: with energy recovery.

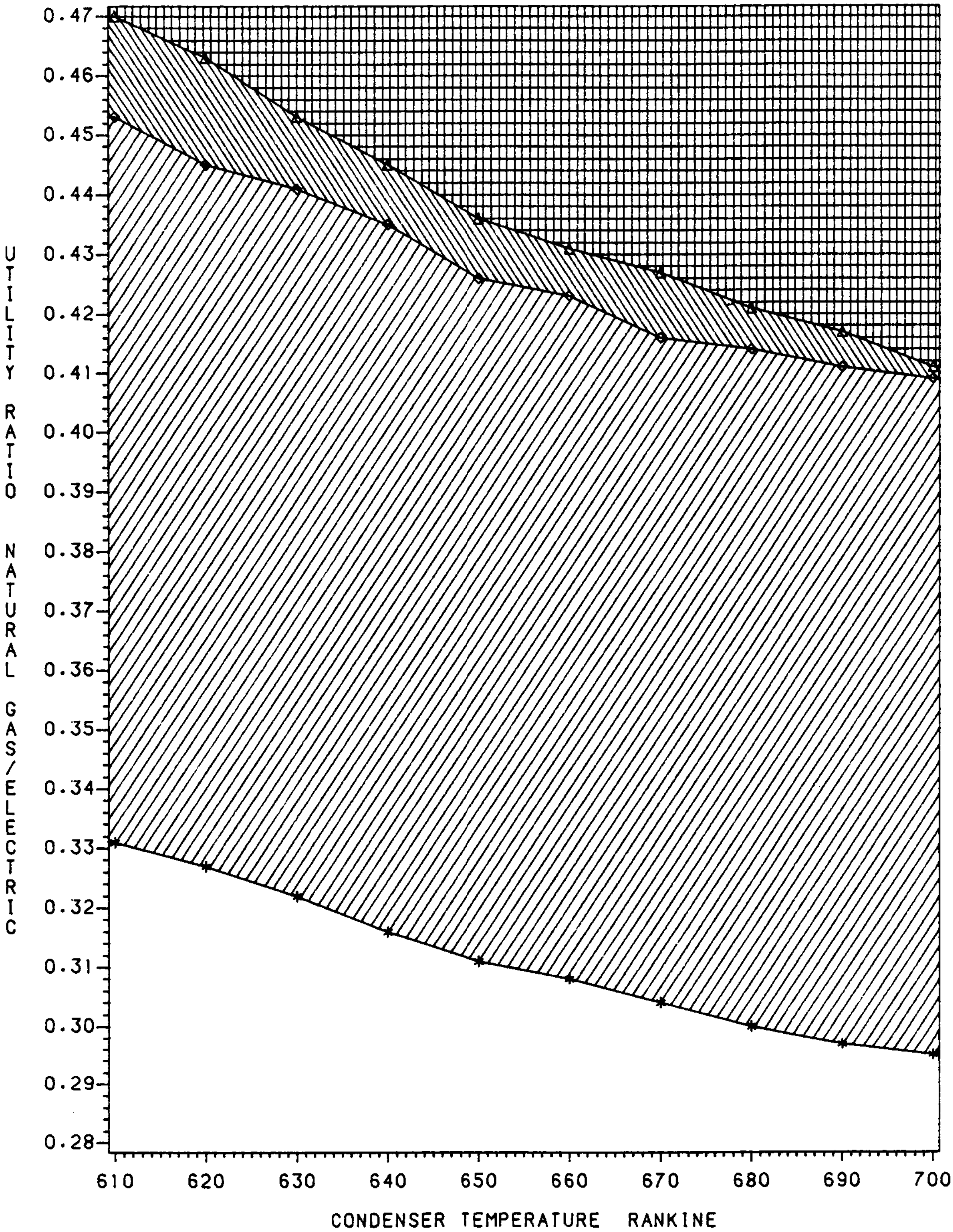

Figure 37. Condenser temperature vs. utility ratio for R-113 with INT = 10.0%, DT = 10 R, and QREB = 200,000,000 BTU/hr; ER = 1986 low TVA load forecast.

Diamonds: Q ratio = 0.5; Stars: Q ratio = 1.0; Triangles: Q ratio = 1.5.

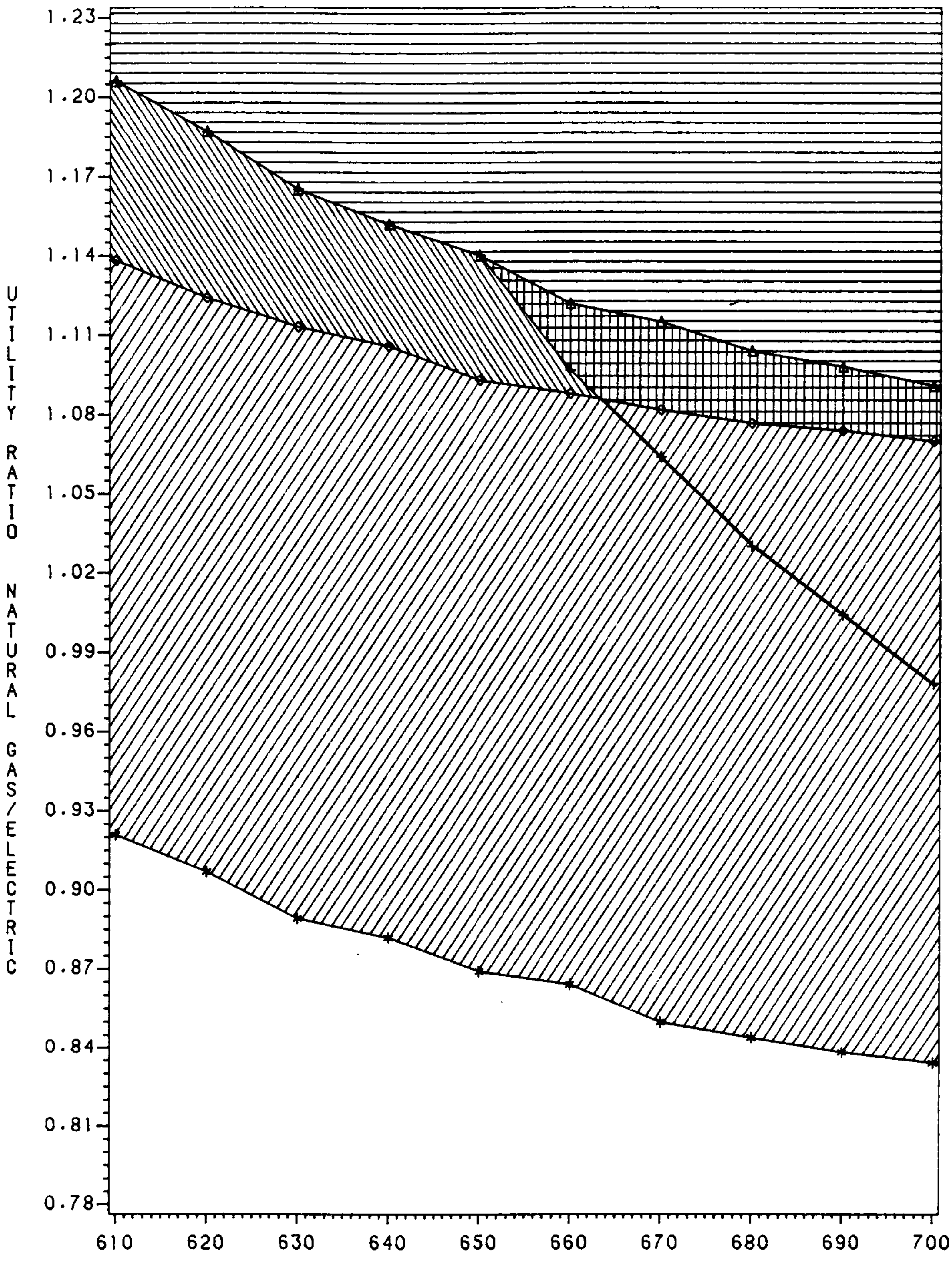

Figure 38. Condenser temperature vs. utility ratio for R-113 with INT = 10.0%, DT = 10 R, and QREB = 200,000 BTU/hr; ER = 1986 medium TVA load forecast.

Diamonds: Q ratio = 0.5; Stars: Q ratio = 1.0; Triangles: Q ratio = 1.5; Pluses: with energy recovery.

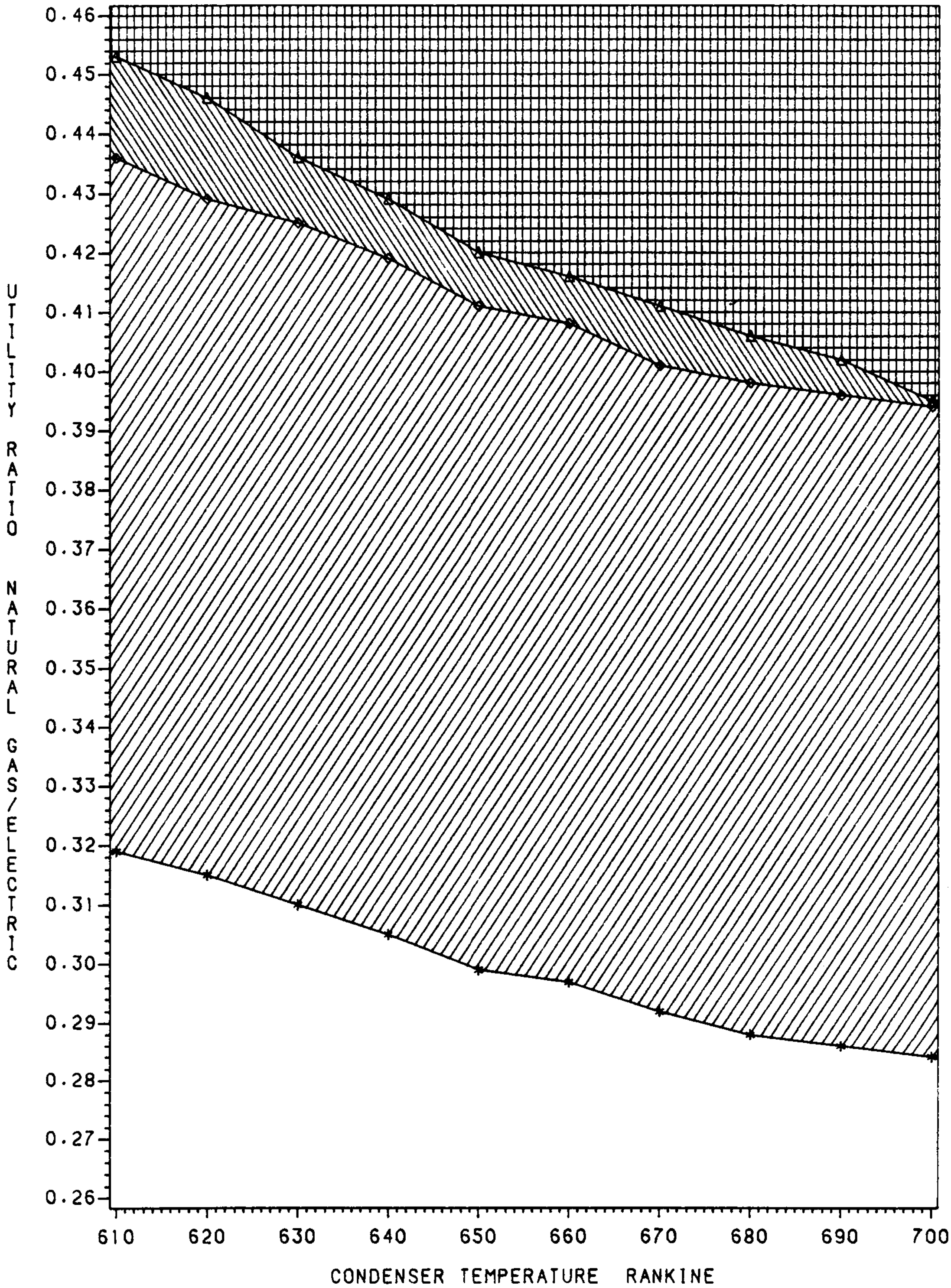

Figure 39. Condenser temperature vs. utility ratio for R-113 with INT = 10.0%, DT = 10 R, and QREB = 200,000,000 BTU/hr; ER = 1986 medium TVA load forecast.

Diamonds: Q ratio = 0.5; Stars: Q ratio = 1.0; Triangles: Q ratio = 1.5.

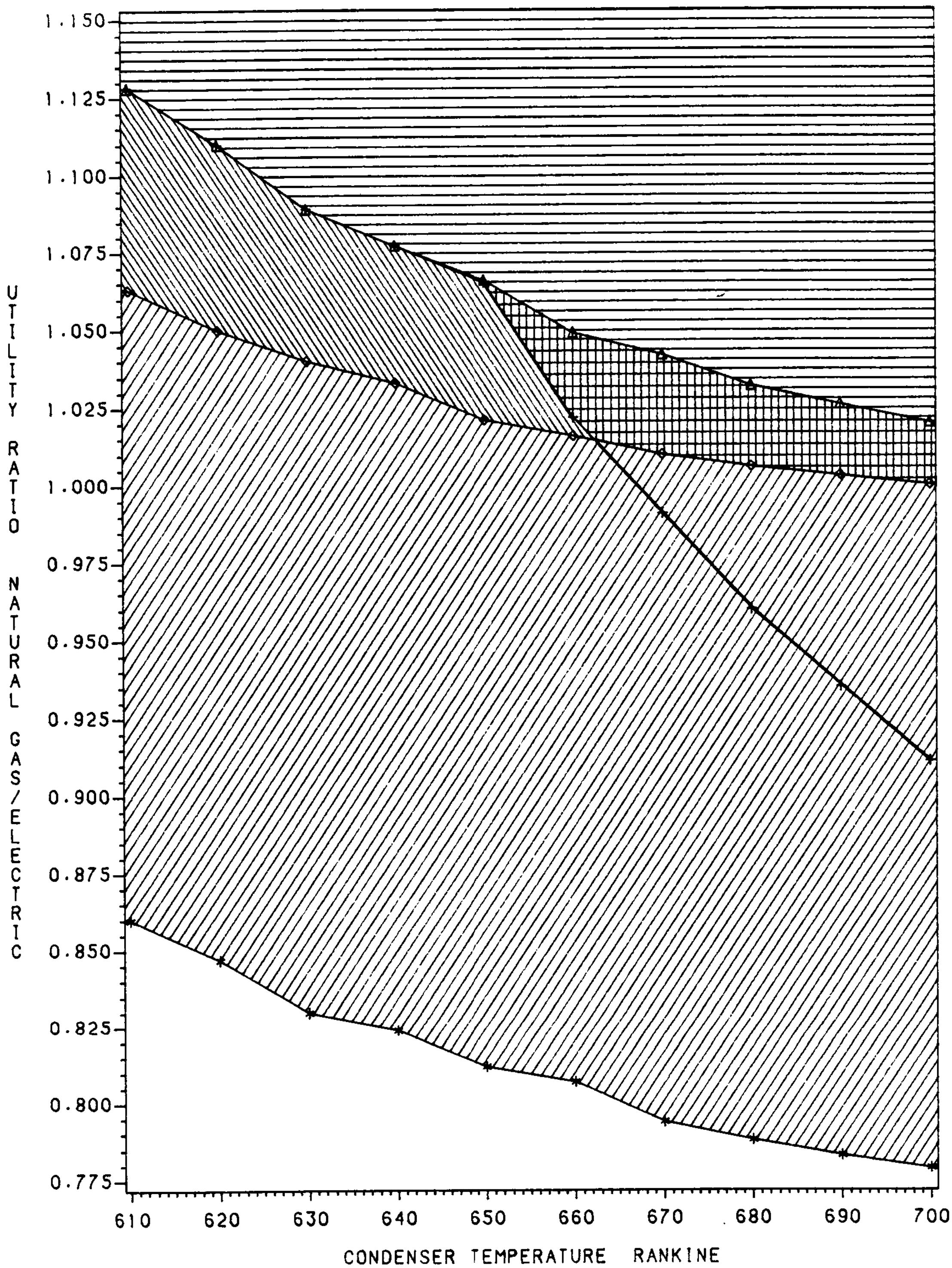

Figure 40. Condenser temperature vs. utility ratio for R-113 with INT = 10.0%, DT = 10 R, and QREB = 200,000 BTU/hr; ER = 1986 high TVA load forecast.

Diamonds: Q ratio = 0.5; Stars: Q ratio = 1.0; Triangles: Q ratio = 1.5; Pluses: with energy recovery.

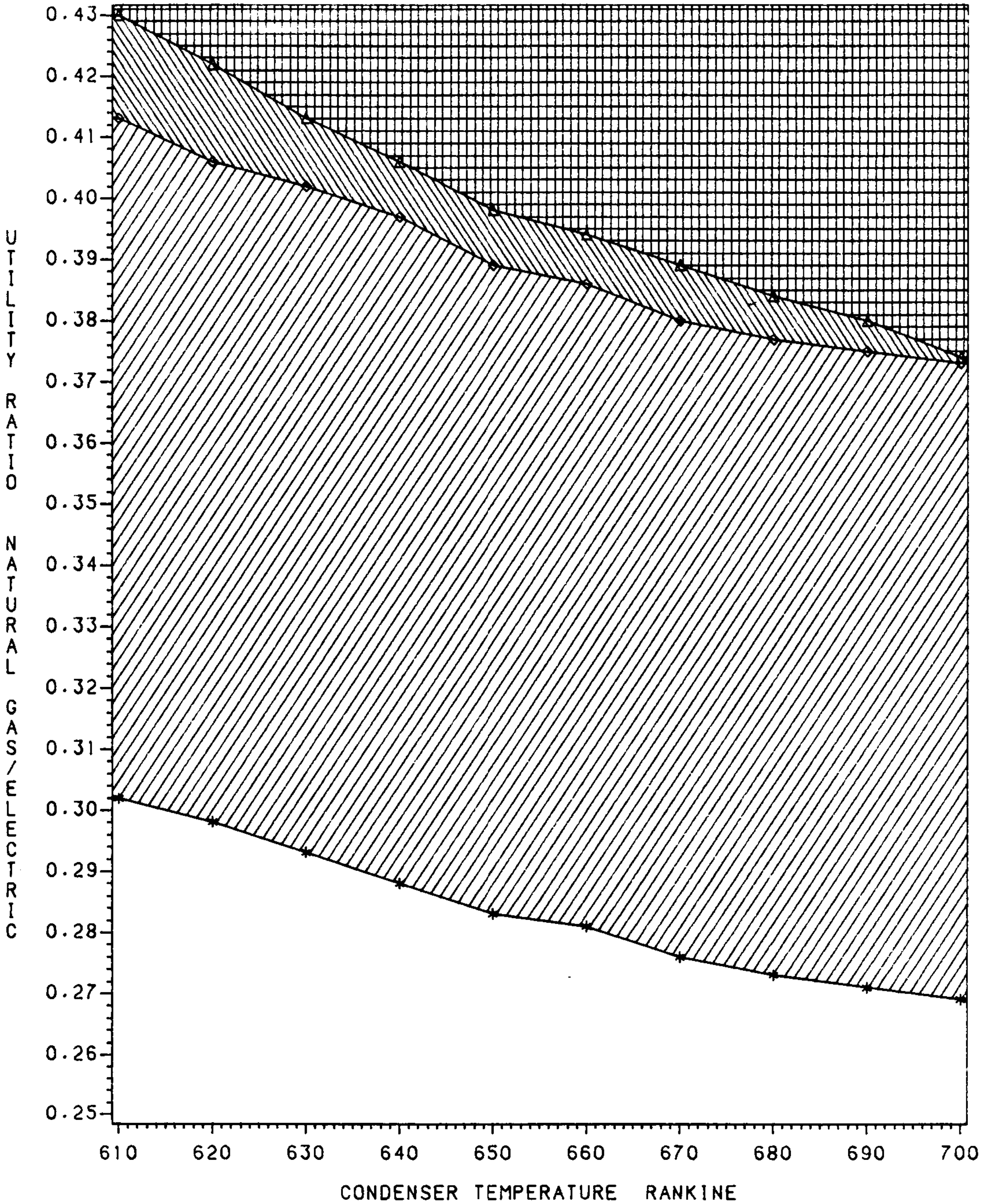

Figure 41. Condenser temperature vs. utility ratio for R-113 with INT = 10.0%, DT = 10 R, and QREB = 200,000,000 BTU/hr; ER = 1986 high TVA load forecast.

Diamonds: Q ratio = 0.5; Stars: Q ratio = 1.0; Triangles: Q ratio = 1.5.

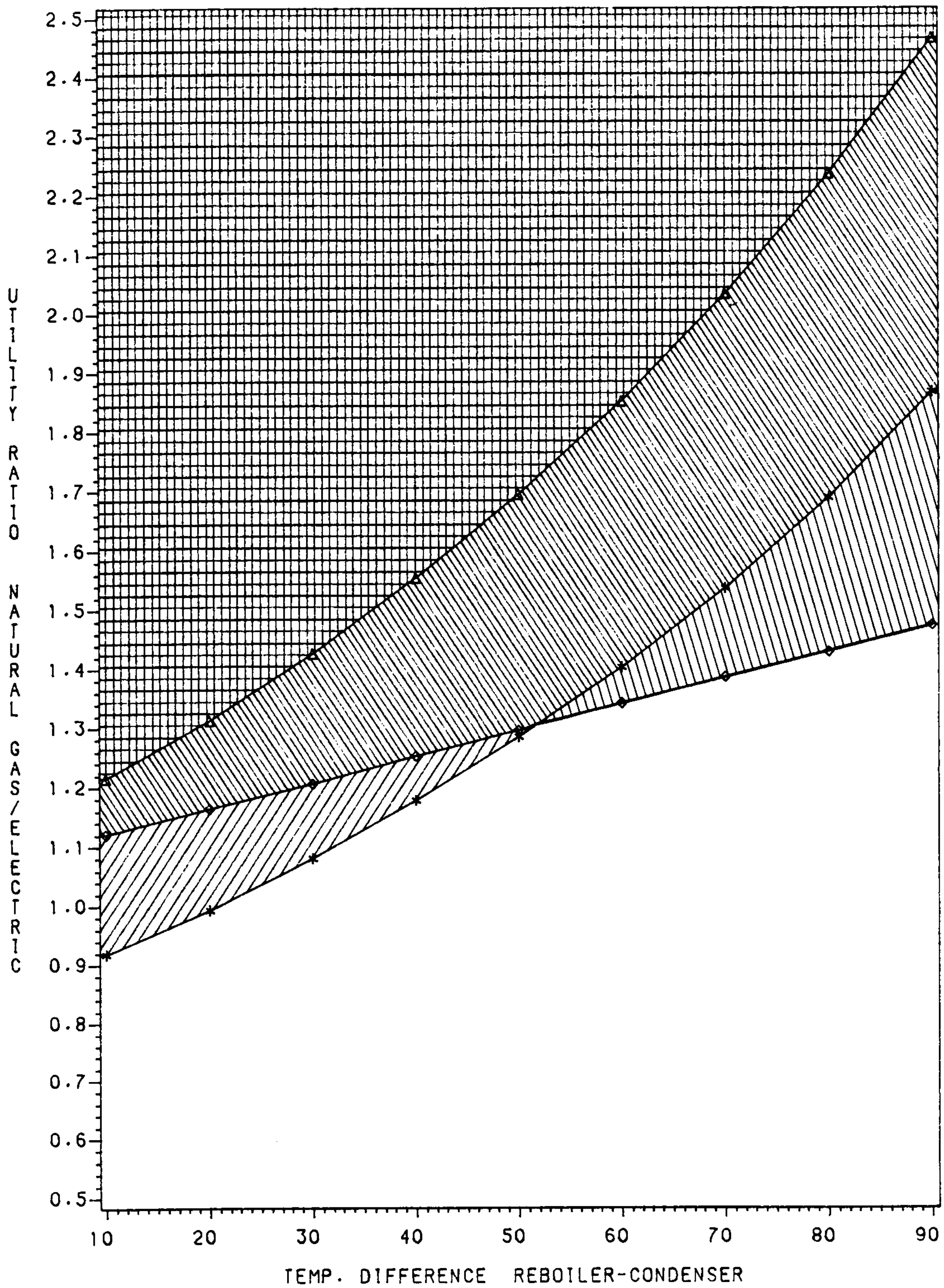

Figure 42. Temperature difference vs. utility ratio for R-114 with INT = 10.0%, T = 560 R, and QREB = 200,000 BTU/hr; ER = 1986 low TVA load forecast.

Diamonds: Q ratio = 0.5; Stars: Q ratio = 1.0; Triangles: Q ratio = 1.5.

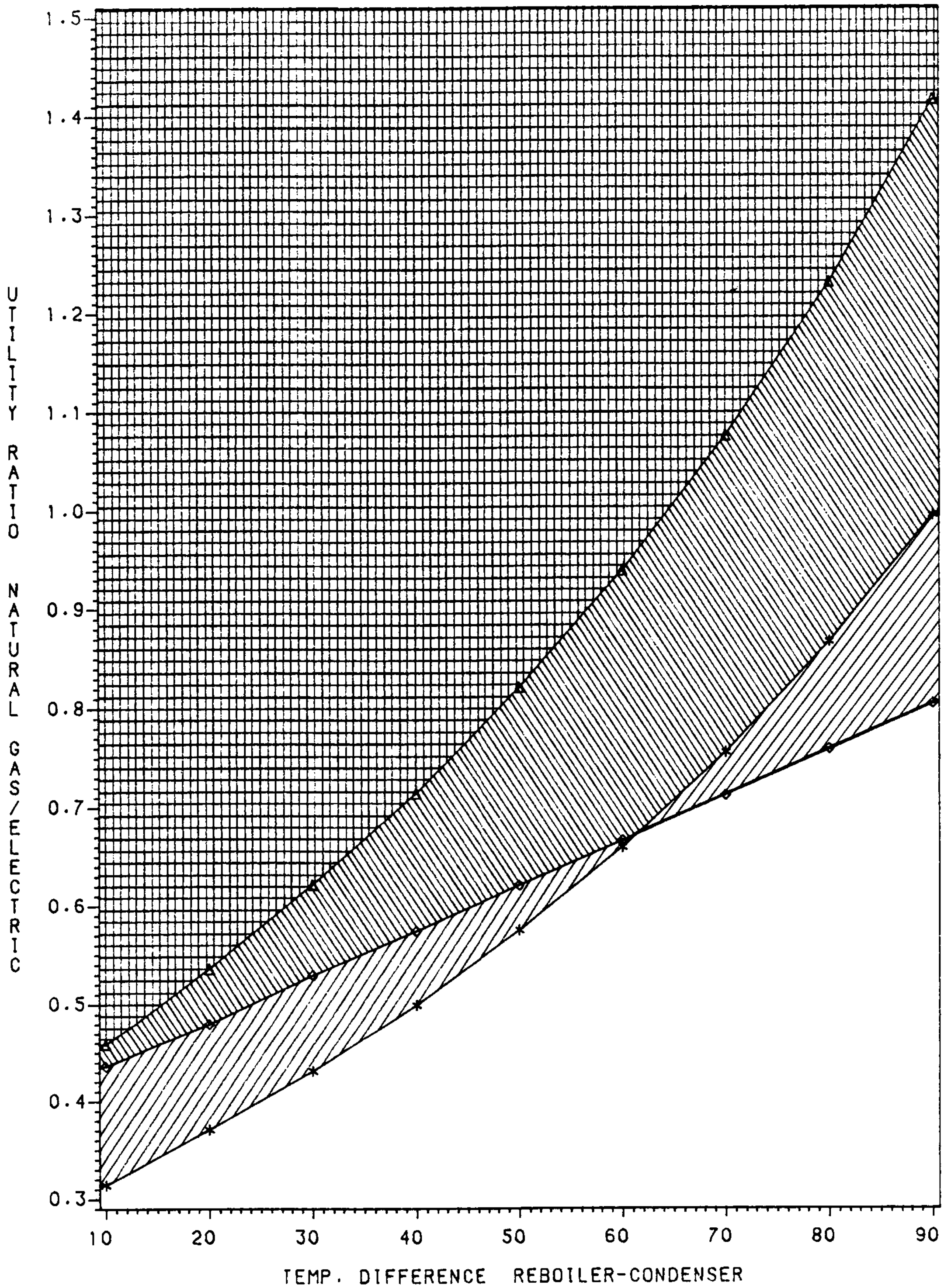

Figure 43. Temperature difference vs. utility ratio for R-114 with INT = 10.0%, T = 560 R, and QREB = 200,000,000 BTU/hr; ER = 1986 low TVA load forecast.

Diamonds: Q ratio = 0.5; Stars: Q ratio = 1.0; Triangles: Q ratio = 1.5.

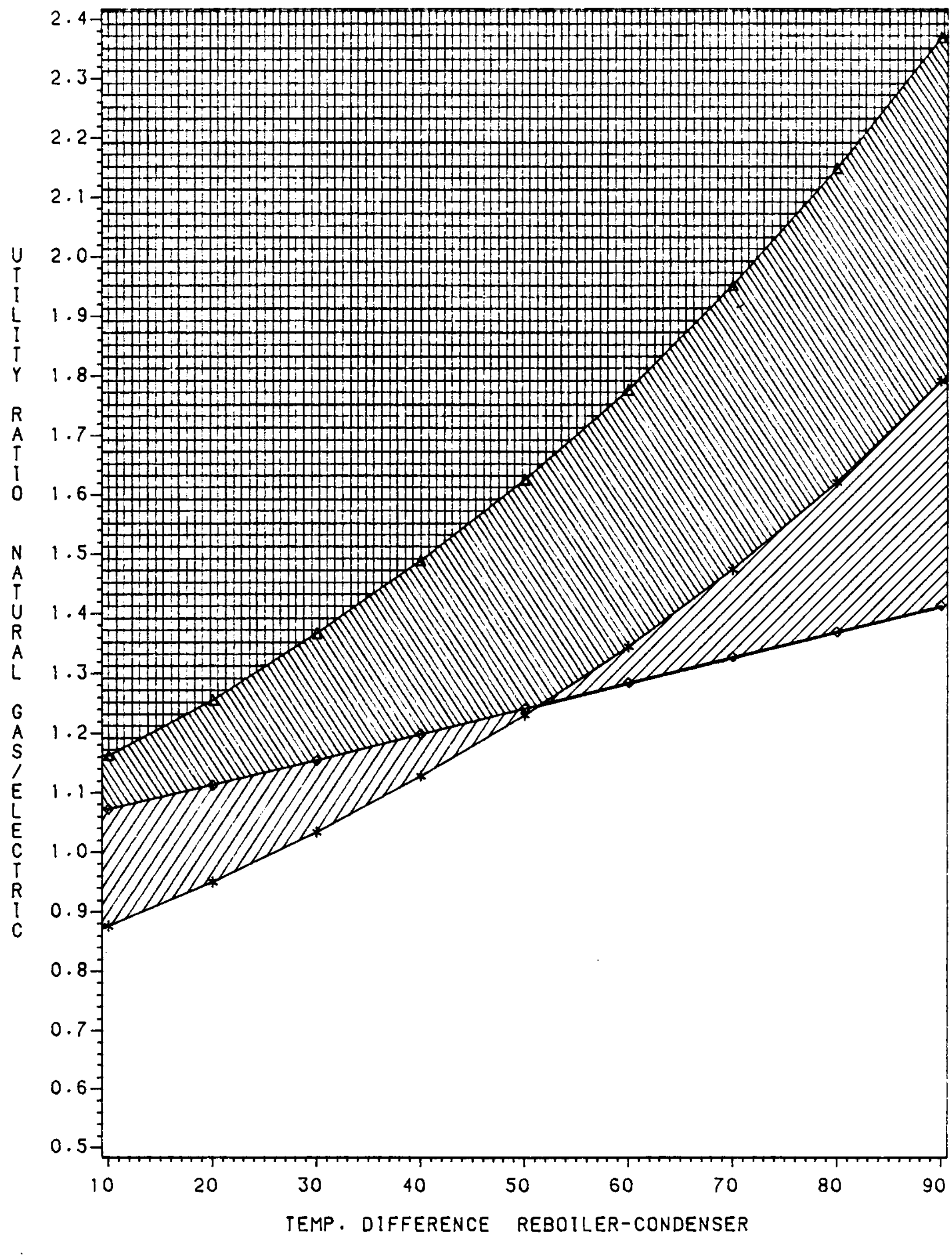

Figure 44. Temperature difference vs. utility ratio for R-114 with INT = 10.0%, T = 560 R, and QREB = 200,000 BTU/hr; ER = 1986 medium TVA load forecast.

Diamonds: Q ratio = 0.5; Stars: Q ratio = 1.0; Triangles: Q ratio = 1.5.

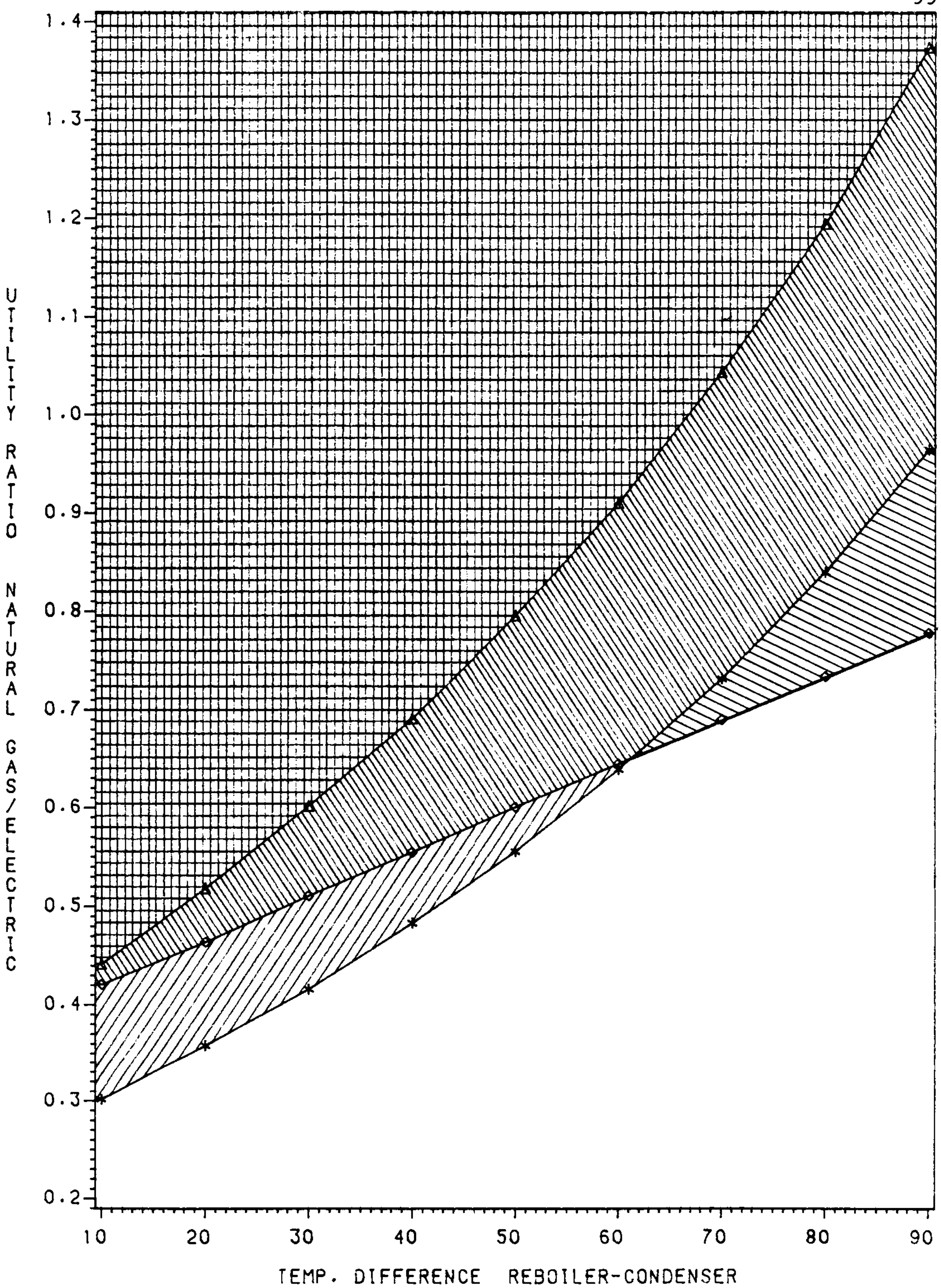

Figure 45. Temperature difference vs. utility ratio for R-114 with INT = 10.0%, T = 560 R, and QREB = 200,000,000 BTU/hr; ER = 1986 medium TVA load forecast.

Diamonds: Q ratio = 0.5; Stars: Q ratio = 1.0; Triangles: Q ratio = 1.5.

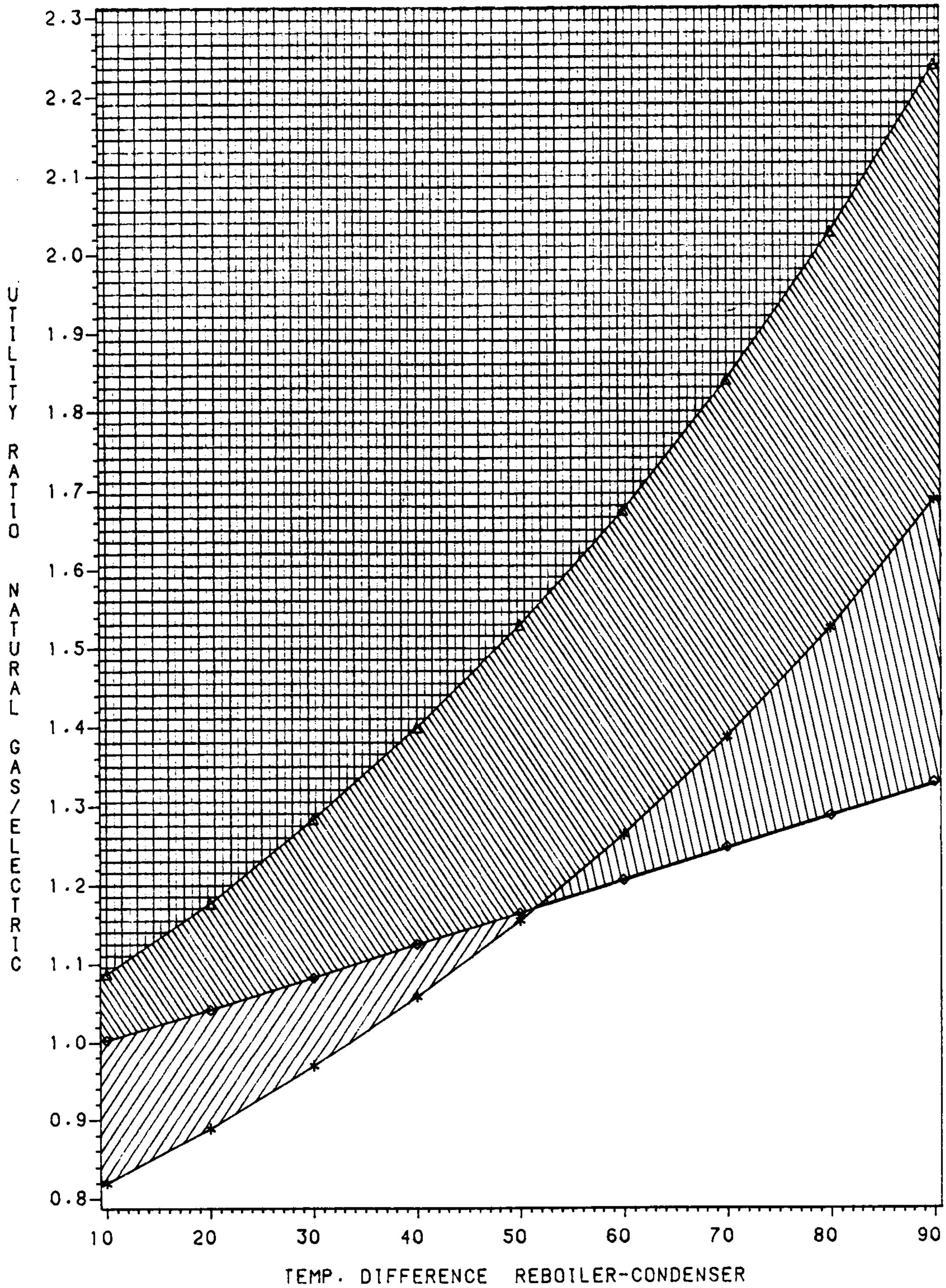

Figure 46. Temperature difference vs. utility ratio for R-114 with INT = 10.0%, T = 560 R, and QREB = 200,000 BTU/hr; ER = 1986 high TVA load forecast.

Diamonds: Q ratio = 0.5; Stars: Q ratio = 1.0; Triangles: Q ratio = 1.5.

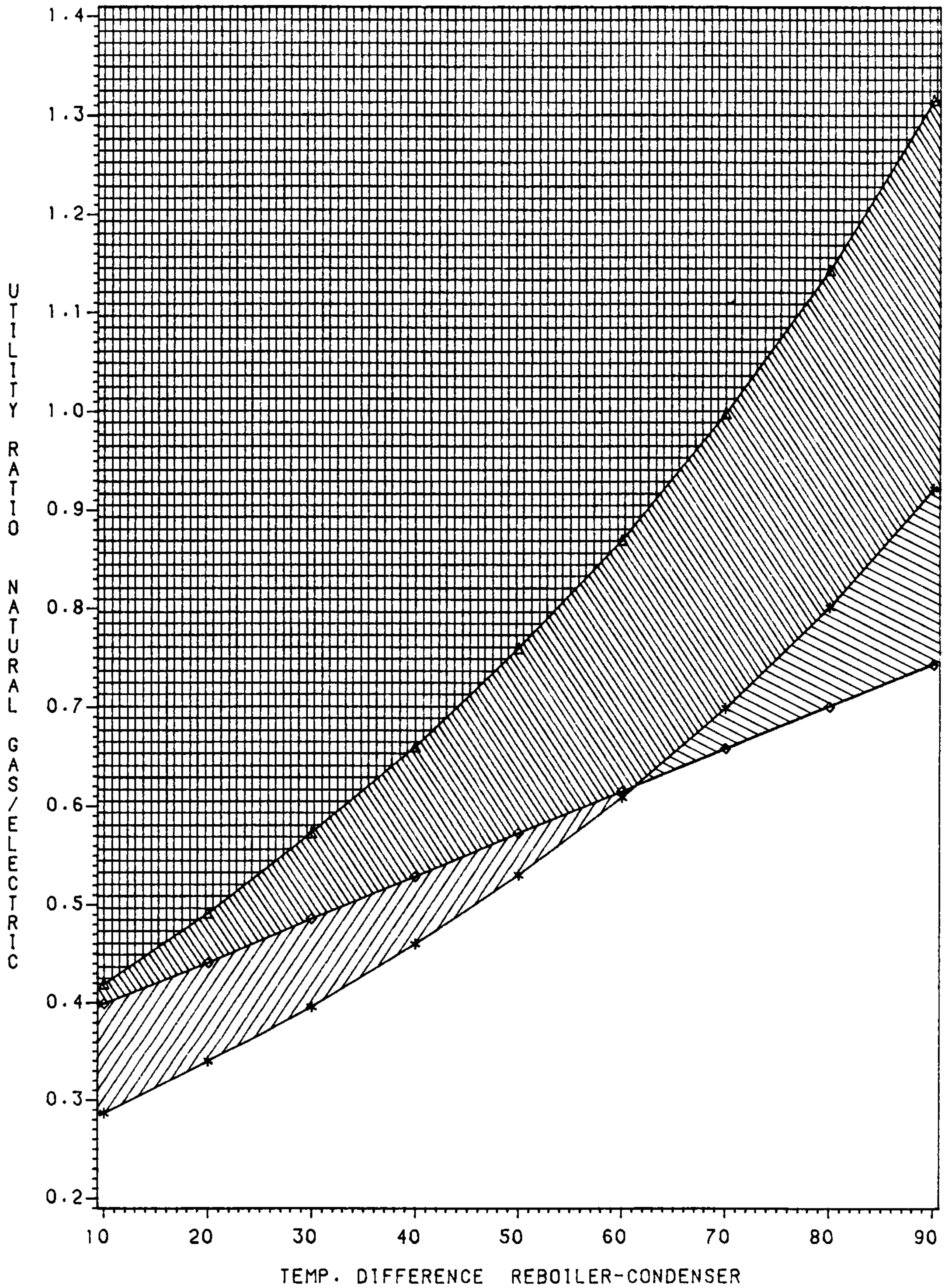

Figure 47. Temperature difference vs. utility ratio for R-114 with INT = 10.0%, T = 560 R, and QREB = 200,000,000 BTU/hr; ER = 1986 high TVA load forecast.

Diamonds: Q ratio = 0.5; Stars: Q ratio = 1.0; Triangles: Q ratio = 1.5.

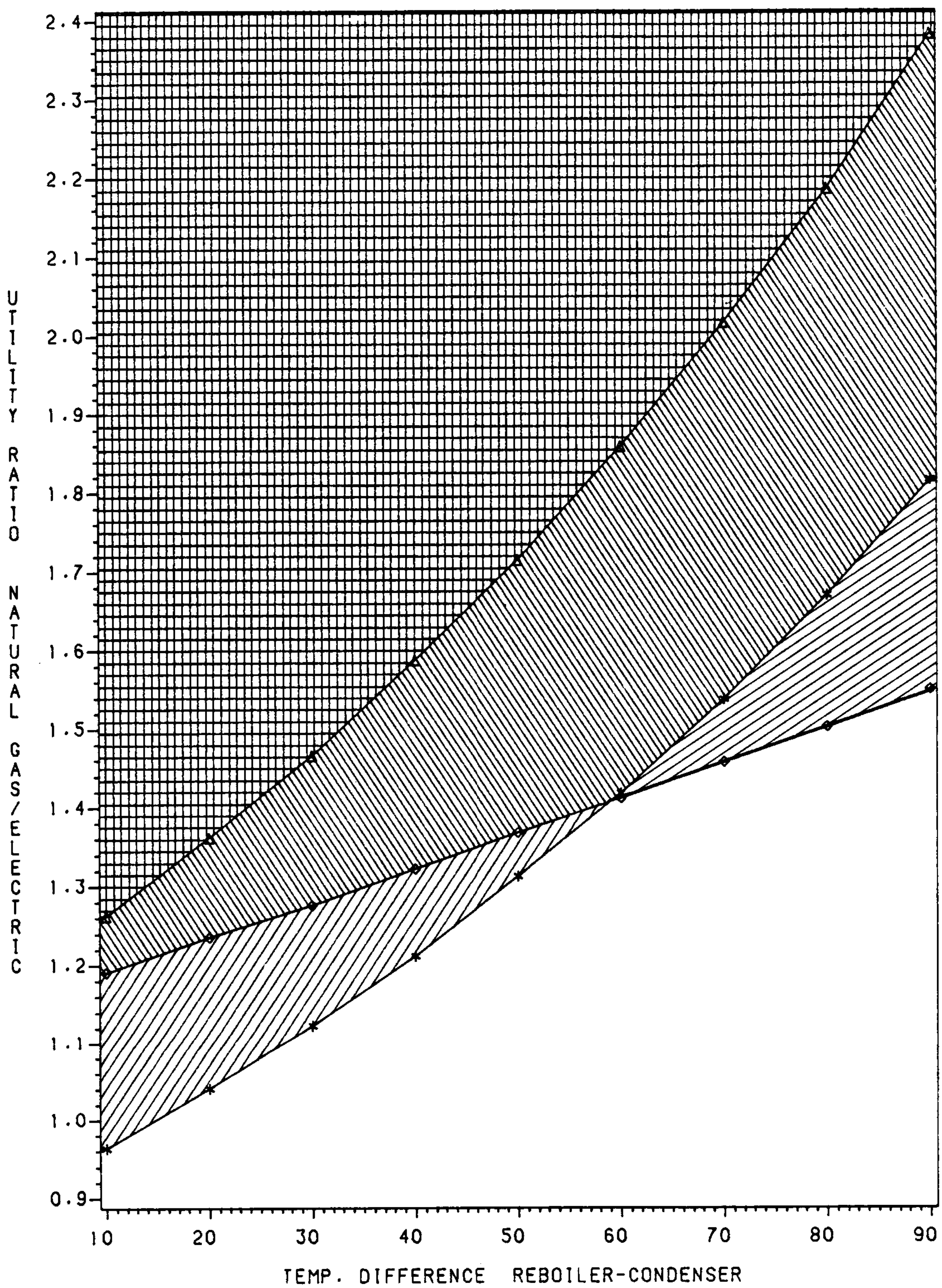

Figure 48. Temperature difference vs. utility ratio for R-113 with INT = 10.0%, T = 610 R, and QREB = 200,000 BTU/hr; ER = 1986 low TVA load forecast.

Diamonds: Q ratio = 0.5; Stars: Q ratio = 1.0; Triangles: Q ratio = 1.5.

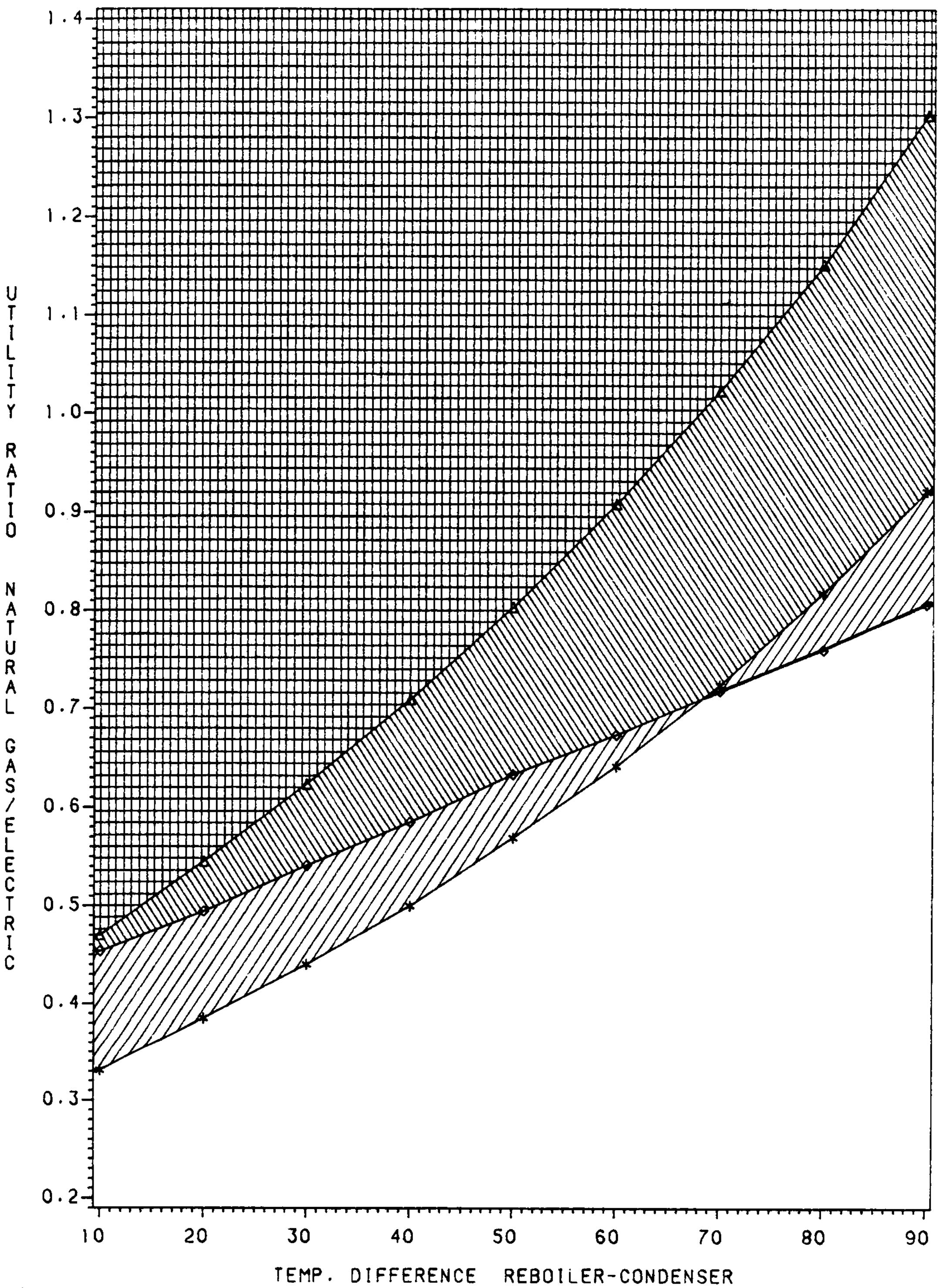

Figure 49. Temperature difference vs. utility ratio for R-113 with INT = 10.0%, T = 610 R, and QREB = 200,000,000 BTU/hr; ER = 1986 low TVA load forecast.

Diamonds: Q ratio = 0.5; Stars: Q ratio = 1.0; Triangles: Q ratio = 1.5.

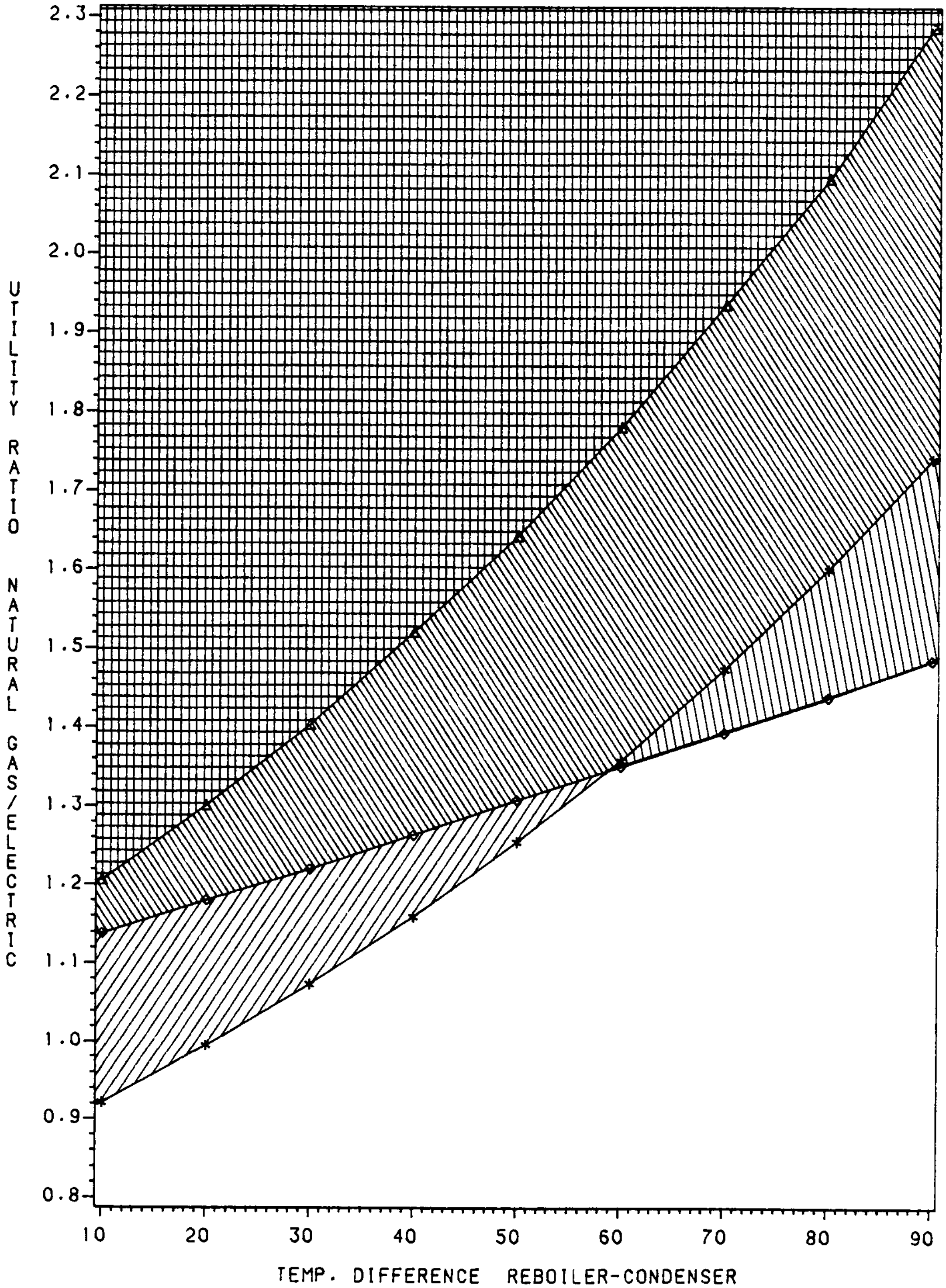

Figure 50. Temperature difference vs. utility ratio for R-113 with INT = 10.0%, T = 610 R, and QREB = 200,000 BTU/hr; ER = 1986 medium TVA load forecast.

Diamonds: Q ratio = 0.5; Stars: Q ratio = 1.0; Triangles: Q ratio = 1.5.

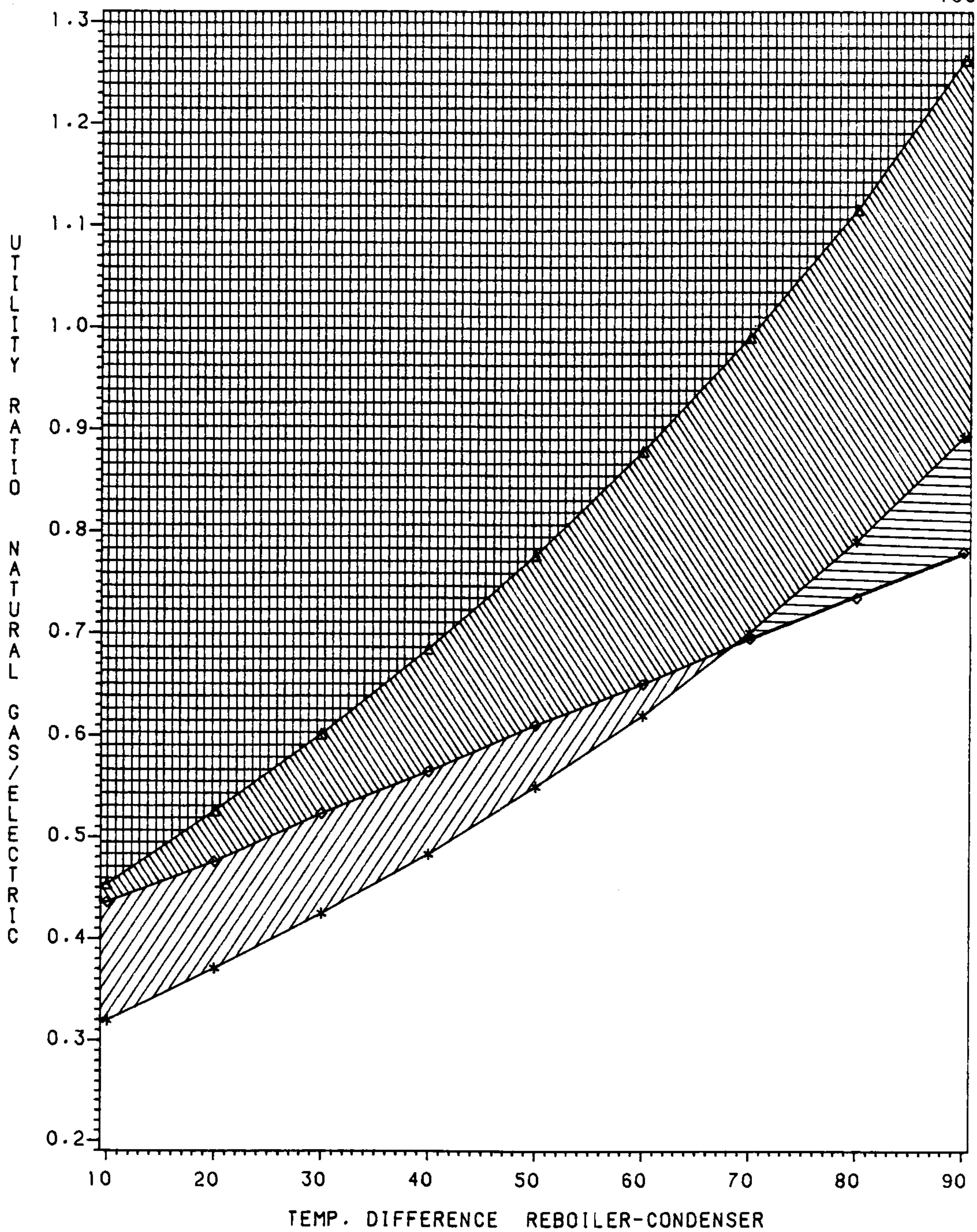

Figure 51. Temperature difference vs. utility ratio for R-113 with INT = 10.0%, T = 610 R, and QREB = 200,000,000 BTU/hr; ER = 1986 medium TVA load forecast.

Diamonds: Q ratio = 0.5; Stars: Q ratio = 1.0; Triangles: Q ratio = 1.5.

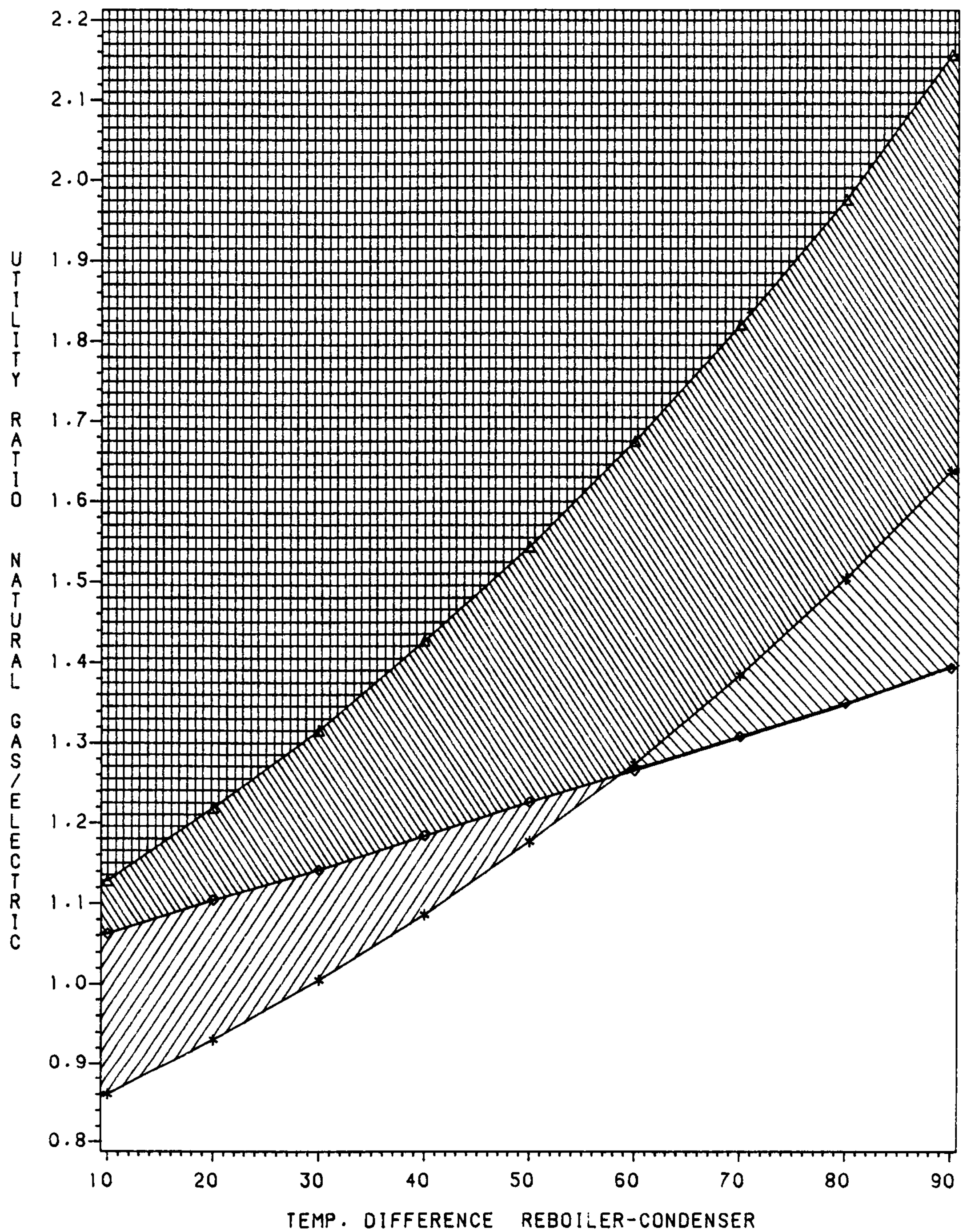

Figure 52. Temperature difference vs. utility ratio for R-113 with INT = 10.0%, T = 610 R, and QREB = 200,000 BTU/hr; ER = 1986 high TVA load forecast.

Diamonds: Q ratio = 0.5; Stars: Q ratio = 1.0; Triangles: Q ratio = 1.5.

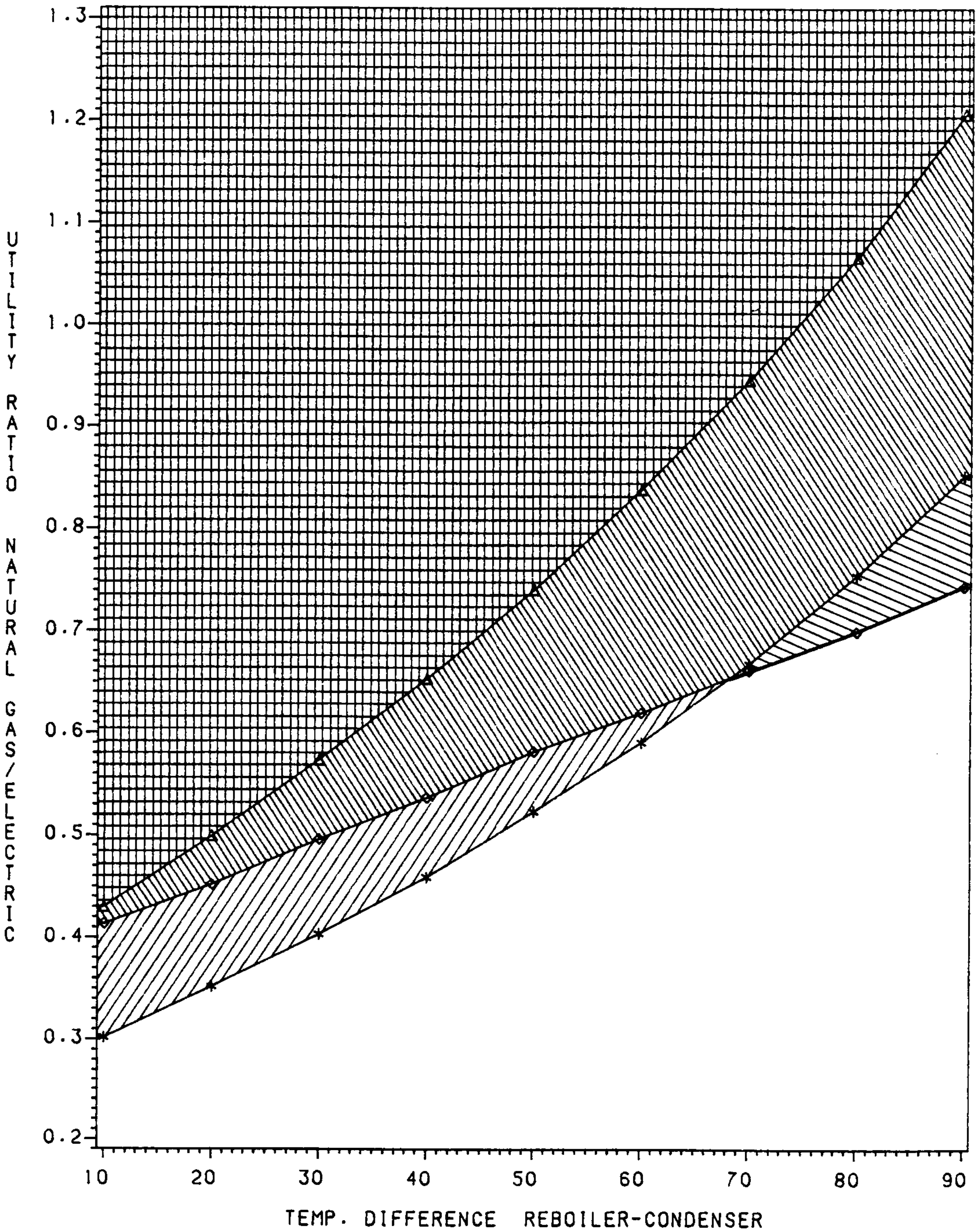

Figure 53. Temperature difference vs. utility ratio for R-113 with INT = 10.0%, T = 610 R, and QREB = 200,000,000 BTU/hr; ER = 1986 high TVA load forecast.

Diamonds: Q ratio = 0.5; Stars: Q ratio = 1.0; Triangles: Q ratio = 1.5.

increasing reboiler heat duty faster than do the costs. The application of a heat pump system is favored by large reboiler heat duties. This effect does level off as Q_{reb} reaches the 200 million/ BTU/hr level.

The point where no trim-heater or trim-cooler is required is the value of Q_{ratio} that is the most favorable as is shown in Figures 18 through 29. This value of Q_{ratio} is always less than one because of the heat added to the working fluid in the compressor. The value of Ur required for the heat pump system to be favorable grows rapidly as Q_{ratio} moves away from the minimum point.

As the condenser temperature increases the compression ratio required in the compressor decreases while the latent heat of vaporization of the refrigerant decreases. This causes the circulation rate of the refrigerant to increase while the energy consumed by the compressor decreases. The required value of Ur decreases slightly as the condenser temperature increases as is shown in Figures 30 through 41.

The effects of the temperature difference between the reboiler and the condenser is shown in Figures 42 through 53. An increase in the temperature difference causes both the compression ratio and the circulation rate to increase. The required value of the utility ratio therefore increases rapidly. The value of Q_{ratio} where no trim-heater or trim-cooler is required is decreased due to the increase in heat added in the compressor. The required utility ratio increases rapidly with an increase in temperature difference as is shown in the figures. The 0.5 Q_{ratio} line increases less rapidly due to the shift in the no trim-exchanger point.

The energy recovery option is utilized when the energy released in the trim-cooler is transferred to a boiling fluid (either the refrigerant used or water depending on the temperature). The receiving fluid is at the same temperature as the boiling process fluid in the reboiler. A credit is taken for this energy at a fraction of the steam cost based on the temperature of the boiling fluid.

Graphs in Figures 12 to 53 are used to determine whether a heat pump system applied to a specific column will be economically feasible. The effects of the four variables on the required utility ratio are additive in nature. For example, a column with a reboiler heat duty of 200 million BTU/hr, a heat duty ratio of 1.0, a condenser temperature of 610 degrees Rankine, and a temperature difference of 30 degrees Rankine would require a utility ratio of 0.42 for the medium load forecast and a R-114 heat pump. This was determined by first consulting Figure 13 where a value of 0.30 was found for a 200 million BTU/hr column with a Q_{ratio} of 1.0. Next, Figure 45 was consulted and the required utility ratio was found to increase from 0.30 where the temperature difference was 10 to a value of 0.42 where the temperature difference was 30. The combined effects of these two variables determines a value of 0.42 Finally, Figure 33 was consulted, and the required utility ratio was found to remain at a value of 0.30 while the condenser temperature changed from 560 degrees Rankine to 610. When this effect is combined with the previous value, the result is the expected 0.42. This value is compared with the actual utility ratio for the medium TVA forecast. If this actual value is greater, then the heat pump system is economically feasible. If the actual value is less, then the heat pump system may still be economically feasible provided that the difference between the two values is not great.

The heat pump system is favored by distillation columns that have large reboiler heat duties, heat duty ratios near the no trim-exchanger point and low temperature differences. The heat pump system requires a utility ratio of at least 0.30 to be an economically feasible alternative.

CHAPTER XI

SPECIFIC CASE ANALYSIS

This analysis is broken into two parts: a qualitative part and a quantitative part. The qualitative deals with whether a heat pump system is worth considering and what options regarding the heat pump system should be considered. The quantitative part is a comparison of the options under consideration on the basis of the required additional investment for the option and the return on that investment.

The analysis used is best explained with the use of an example. The example distillation column is one from a distillation train that is used in a wood to ethanol plant developed by Juan Ferrada.[26] The column chosen is an azcotropic distillation column where benzene is added to an ethanol-water solution to obtain a product stream richer in ethanol than that obtained without the benzene. Vapor is drawn off the top of the column at 614.0 degrees Rankine and is returned as a liquid at 601.4 degrees Rankine, for an average condenser temperature of 608.0 degrees Rankine. The reboiler heat duty is 56.6 million btu/hr with a heat duty ratio of 1.019. The temperature difference between the reboiler and condenser is 30.8 degrees Rankine and the column has an operating pressure of 1.0 atm.

Qualitative

The first step is to determine a levelized utility ratio present over the lifespan of the project. This is done by determining the net present worth of 1 BTU in each year of both electricity and natural gas. Then the net present worth of the natural gas is divided by that of the electricity to give the levelized utility ratio:

$$Ur = Gr/Er$$

where Ur is the utility ratio, Gr is the net present worth of the natural gas and Er is the net present worth of the electricity. With the project starting in 1987 and using a 10 percent discount rate, the utility ratios for the three TVA load forecasts are 0.249 for the low forecast, 0.271 for the medium forecast, and 0.287 for the high forecast.

With the utility ratio found, the graphs in the previous chapter are consulted. The minimum required utility ratio is determined by the method described there. This value is compared with the utility ratio present and if the present value is larger, then a heat pump system should be considered. If the present value is less, as it is for the example column, then the heat pump system is not favored. A heat pump system that utilizes one of the options discussed later may still be worth considering provided that the present value is at least in the same order of magnitude.

The example column requires a utility ratio of about 0.45 for both a system using R-114 and a system using R-113. This value does not remove the heat pump system from consideration provided an option can still be used.

Quantitative

Tables 1 through 12 outline the major expenses of the heat pump system with an external working fluid. The first six tables are for a system using R-114 and the rest are for a system using R-113. Tables 1 through 3 outline the capital investment required for the R-114 system. Tables 4 through 6 are discounted cash flow tables for each of the three forecasts using the actual utility ratio. The return on investment shown is not favorable.

The next set of tables (Tables 13 through 24) outline basically the same heat pump system but with the compressor being driven by a steam turbine rather than an electric motor. This increases the capital cost of the compressor by 15 percent but reduces the energy cost. The cost of steam was estimated as before as the cost of the natural gas required to produce it subjected to an 80 percent boiler efficiency. A turbine efficiency of 70 percent was used. These tables show that this is a favorable option.

Using the graphs from the previous chapter for this option requires that the utility ratio be redefined as the ratio of the cost to supply energy to the conventional reboiler to the cost to supply energy to the heat pump compressor on a $/million BTU basis. After

Table 1. Equipment Specifications for the Conventional and Heat Pump Systems Using R-114 and an Electric Compressor

Item	No. Req.	Size Each	Material Const.
Condenser	1	7223.216 Sq. Ft.	Carbon Steel
Reboiler	9	9207.785 Sq. Ft.	Carbon Steel
Trim-Cooler	1	2775.952 Sq. Ft.	Carbon Steel
Innercooler	1	1892.207 Sq. Ft.	Carbon Steel
Start-Up Boiler	1	1307.551 Sq. Ft.	Carbon Steel
Compressor	2	2423.278 HP	Carbon Steel
Conventional System			
Condenser	1	6942.006 Sq. Ft.	Carbon Steel
Reboiler	1	1492.556 Sq. Ft.	Carbon Steel

Table 2. Purchased Equipment Cost for the Conventional and Heat Pump Systems Using R-114 and an Electric Compressor (in 1984 dollars)

Item	No. Req.	Cost Each	Total Cost
Condenser	1	87258.206	87258.206
Reboiler	1	101726.614	915539.531
Trim-Cooler	1	39732.297	39732.297
Innercooler	1	31185.415	31185.415
Start-Up Boiler	1	32589.903	32589.903
Compressor	2	243187.574	486375.148
Total			1592680.516
Conventional System			
Condenser	1	85095.585	85095.585
Reboiler	1	35432.788	35432.788
Total			120528.373

Table 3. Capital Investment for the Conventional System and Heat Pump System Using R-114 and Electric Compressor (in millions of 1984 dollars)

Item	Heat Pump	Conventional
	Direct Costs	
Purchased Equipment	1.5927	0.1205
Installation	0.7486	0.0566
Control Systems	0.3345	0.0253
Piping	1.0512	0.0795
Electrical	0.2230	0.0169
Buildings	0.2867	0.0217
Yard Improvements	0.1593	0.0121
Service Facilities	1.1149	0.0844
Total Direct Cost	5.5107	0.4170
	Indirect Costs	
Engineering	0.5256	0.0398
Construction	0.6530	0.0494
Total	6.6893	0.5062
Contractor's Fee	0.3345	0.0253
Contingency	0.6689	0.0506
Fixed Capital	7.6926	0.5822
Working Capital	1.3697	0.1037
Total Capital	9.0624	0.6858
Total Additional Capital Investment for the Heat Pump System	8.3765	

Table 4. Heat Pump System Using R-114 and Electric Compressor Discounted Cash Flow with 1986 TVA Low Load Forecast

(millions of dollars)

Year	Investment at Start of Year	Annual Utility Savings	Annual Main. Expenses	Deprec.	Profit Before Tax	Federal Tax	Profit After Tax	After Tax Cash Flow	Investment at End of Year
1	8.3765	0.6561	0.2944	0.5404	-0.1787	0.0000	-0.1787	0.3617	7.4165
2	7.4165	0.5780	0.2944	0.9457	-0.6621	0.0000	-0.6621	0.2836	6.6031
3	6.6031	0.5086	0.2944	0.8106	-0.5964	0.0000	-0.5964	0.2142	5.9173
4	5.9173	0.4305	0.2944	0.6755	-0.5394	0.0000	-0.5394	0.1361	5.3585
5	5.3585	0.5078	0.2944	0.6755	-0.4621	0.0000	-0.4621	0.2134	4.7624
6	4.7624	0.5880	0.2944	0.6755	-0.3819	0.0000	-0.3819	0.2936	4.1286
7	4.1286	0.6597	0.2944	0.6079	-0.2427	0.0000	-0.2427	0.3653	3.4684
8	3.4684	0.7477	0.2944	0.6079	-0.1547	0.0000	-0.1547	0.4533	2.7674
9	2.7674	0.8280	0.2944	0.6079	-0.0744	0.0000	-0.0744	0.5336	2.0361
10	2.0361	0.9143	0.2944	0.6079	-0.0119	0.0000	0.0119	0.6199	1.2708

Return on Investment = -0.07143 Utility Ratio = 0.24915

Table 5. Heat Pump System Using R-114 and Electric Compressor Discounted Cash Flow with 1986 TVA Medium Load Forecast

(millions of dollars)

Year	Investment at Start of Year	Annual Utility Savings	Annual Main. Expenses	Deprec.	Profit Before Tax	Federal Tax	Profit After Tax	After Tax Cash Flow	Investment at End of Year
1	8.3765	0.7548	0.2944	0.5404	-0.0800	0.0000	-0.0800	0.4604	7.7305
2	7.7305	0.7177	0.2944	0.9457	-0.5224	0.0000	-0.5224	0.4233	7.1359
3	7.1359	0.6556	0.2944	0.8106	-0.4495	0.0000	-0.4495	0.3611	6.6166
4	6.6166	0.6016	0.2944	0.6755	-0.3683	0.0000	-0.3683	0.3072	6.1628
5	6.1628	0.7116	0.2944	0.6755	-0.2583	0.0000	-0.2583	0.4172	5.6091
6	5.6091	0.8238	0.2944	0.6755	-0.1461	0.0000	-0.1461	0.5294	4.9554
7	4.9554	0.9532	0.2944	0.6079	0.0508	0.0000	0.0508	0.6588	4.1868
8	4.1868	1.0856	0.2944	0.6079	0.1832	0.0089	0.1743	0.7823	3.3117
9	3.3117	1.2206	0.2944	0.6079	0.3182	0.0182	0.3000	0.9080	2.3304
10	2.3304	1.3418	0.2944	0.6079	0.4395	0.0266	0.4129	1.0208	1.2579

Return on Investment = -0.02216 Utility Ratio = 0.27075

Table 6. Heat Pump System Using R-114 and Electric Compressor Discounted Cash Flow with 1986 TVA High Load Forecast

(millions of dollars)

Year	Investment at Start of Year	Annual Utility Savings	Annual Main. Expenses	Deprec.	Profit Before Tax	Federal Tax	Profit After Tax	After Tax Cash Flow	Investment at End of Year
1	8.3765	0.8548	0.2944	0.5404	0.0200	0.0000	0.0200	0.5604	7.9253
2	7.9253	0.8306	0.2944	0.9457	-0.4095	0.0000	-0.4095	0.5361	7.4923
3	7.4923	0.8003	0.2944	0.8106	-0.3047	0.0000	-0.3047	0.5059	7.0840
4	7.0840	0.7610	0.2944	0.6755	-0.2089	0.0000	-0.2089	0.4666	6.7097
5	6.7097	0.9141	0.2944	0.6755	-0.0558	0.0000	-0.0558	0.6197	6.1774
6	6.1774	1.0724	0.2944	0.6755	0.1024	0.0033	0.0991	0.7746	5.4832
7	5.4832	1.2474	0.2944	0.6079	0.3451	0.0201	0.3250	0.9329	4.6217
8	4.6217	1.4139	0.2944	0.6079	0.5115	0.0315	0.4800	1.0879	3.5940
9	3.5940	1.6170	0.2944	0.6079	0.7146	0.0456	0.6691	1.2770	2.3638
10	2.3638	1.8171	0.2944	0.6079	0.9147	0.3883	0.5264	1.1344	1.2602

Return on Investment = 0.01302   Utility Ratio = 0.28671

Table 7. Equipment Specifications for the Conventional and Heat Pump Systems Using R-113 and an Electric Compressor

Item	No. Req.	Size Each	Material Const.
Condenser	1	7311.029 Sq. Ft.	Carbon Steel
Reboiler	10	9990.874 Sq. Ft.	Carbon Steel
Trim-Cooler	1	2428.114 Sq. Ft.	Carbon Steel
Innercooler	1	2542.016 Sq. Ft.	Carbon Steel
Start-Up Boiler	1	1356.505 Sq. Ft.	Carbon Steel
Compressor	2	2369.723 HP	Carbon Steel
Conventional System			
Condenser	1	7031.314 Sq. Ft.	Carbon Steel
Reboiler	1	1492.131 Sq. Ft.	Carbon Steel

Table 8. Purchased Equipment Cost for the Conventional and Heat Pump Systems Using R-113 and an Electric Compressor (in 1984 dollars)

Item	No. Req.	Cost Each	Total Cost
Condenser	1	87927.136	87927.136
Reboiler	10	107111.947	1071119.469
Trim-Cooler	1	36508.782	36508.782
Innercooler	1	37582.009	37582.009
Start-Up Boiler	1	33355.811	33355.811
Compressor	2	238771.715	477543.430
Total			1744036.656
Conventional System			
Condenser	1	85785.834	85785.834
Reboiler	1	35426.413	35426.413
Total			121212.247

Table 9. Capital Investment for the Conventional System and Heat Pump System Using R-113 and Electric Compressor (in millions of 1984 dollars)

Item	Heat Pump	Conventional
	Direct Costs	
Purchased Equipment	1.7440	0.1212
Installation	0.8197	0.0570
Control Systems	0.3662	0.0255
Piping	1.1511	0.0800
Electrical	0.2442	0.0170
Buildings	0.3135	0.0218
Yard Improvements	0.1744	0.0121
Service Facilities	1.2208	0.0848
Total Direct Cost	6.0344	0.4194
	Indirect Costs	
Engineering	0.5755	0.0400
Construction	0.7151	0.0497
Total	7.3250	0.5091
Contractor's Fee	0.3662	0.0255
Contingency	0.7325	0.0509
Fixed Capital	8.4237	0.5855
Working Capital	1.4999	0.1042
Total Capital	9.9236	0.6897
Total Additional Capital Investment for the Heat Pump System	9.2339	

Table 10. Heat Pump System Using R-113 and Electric Compressor Discounted Cash Flow with 1986 TVA Low Load Forecast

(millions of dollars)

Year	Investment at Start of Year	Annual Utility Savings	Annual Main. Expenses	Deprec.	Profit Before Tax	Federal Tax	Profit After Tax	After Tax Cash Flow	Investment at End of Year
1	9.2339	0.6781	0.3246	0.5957	-0.2422	0.0000	-0.2422	0.3535	8.1343
2	8.1343	0.5999	0.3246	1.0425	-0.7672	0.0000	-0.7672	0.2753	7.2018
3	7.2018	0.5304	0.3246	0.8936	-0.6877	0.0000	-0.6877	0.2058	6.4141
4	6.4141	0.4523	0.3246	0.7446	-0.6169	0.0000	-0.6169	0.1277	5.7681
5	5.7681	0.5292	0.3246	0.7446	-0.5400	0.0000	-0.5400	0.2046	5.0974
6	5.0974	0.6092	0.3246	0.7446	-0.4600	0.0000	-0.4600	0.2846	4.4010
7	4.4010	0.6805	0.3246	0.6702	-0.3142	0.0000	-0.3142	0.3559	3.6895
8	3.6895	0.7683	0.3246	0.6702	-0.2264	0.0000	-0.2264	0.4438	2.9476
9	2.9476	0.8483	0.3246	0.6702	-0.1464	0.0000	-0.1464	0.5237	2.1857
10	2.1857	0.9342	0.3246	0.6702	-0.0605	0.0000	-0.0605	0.6096	1.3995

Return on Investment = -0.08079 Utility Ratio = 0.24915

Table 11. Heat Pump System Using R-113 and Electric Compressor Discounted Cash Flow with 1986 TVA Medium Load Forecast

(millions of dollars)

Year	Investment at Start of Year	Annual Utility Savings	Annual Main. Expenses	Deprec.	Profit Before Tax	Federal Tax	Profit After Tax	After Tax Cash Flow	Investment at End of Year
1	9.2339	0.7772	0.3246	0.5957	-0.1431	0.0000	-0.1431	0.4526	8.4781
2	8.4781	0.7402	0.3246	1.0425	-0.6269	0.0000	-0.6269	0.4156	7.7840
3	7.7840	0.6782	0.3246	0.8936	-0.5399	0.0000	-0.5399	0.3536	7.1748
4	7.1748	0.6244	0.3246	0.7446	-0.4448	0.0000	-0.4448	0.2998	6.6394
5	6.6394	0.7342	0.3246	0.7446	-0.3350	0.0000	-0.3350	0.4096	6.0118
6	6.0118	0.8464	0.3246	0.7446	-0.2228	0.0000	-0.2228	0.5218	5.2926
7	5.2926	0.9757	0.3246	0.6702	-0.0190	0.0000	-0.0190	0.6511	4.4677
8	4.4677	1.1080	0.3246	0.6702	0.1133	0.0041	0.1092	0.7794	3.5416
9	3.5416	1.2429	0.3246	0.6702	0.2482	0.0134	0.2348	0.9050	2.5203
10	2.5203	1.3640	0.3246	0.6702	0.3693	0.0217	0.3475	1.0177	1.4199

Return on Investment = -0.03283 Utility Ratio = 0.27075

Table 12. Heat Pump System Using R-113 and Electric Compressor Discounted Cash Flow with 1986 TVA High Load Forecast

(millions of dollars)

Year	Investment at Start of Year	Annual Utility Savings	Annual Main. Expenses	Deprec.	Profit Before Tax	Federal Tax	Profit After Tax	After Tax Cash Flow	Investment at End of Year
1	9.2339	0.8778	0.3246	0.5957	-0.0425	0.0000	-0.0425	0.5532	8.7139
2	8.7139	0.8540	0.3246	1.0425	-0.5131	0.0000	-0.5131	0.5294	8.2159
3	8.2159	0.8241	0.3246	0.8936	-0.3940	0.0000	-0.3940	0.4995	7.7459
4	7.7459	0.7853	0.3246	0.7446	-0.2839	0.0000	-0.2839	0.4607	7.3131
5	7.3131	0.9386	0.3246	0.7446	-0.1306	0.0000	-0.1306	0.6140	6.7254
6	6.7254	1.0972	0.3246	0.7446	0.0280	0.0000	0.0280	0.7726	5.9770
7	5.9770	1.2725	0.3246	0.6702	0.2778	0.0154	0.2623	0.9325	5.0660
8	5.0660	1.4393	0.3246	0.6702	0.4446	0.0269	0.4176	1.0878	3.9965
9	3.9965	1.6427	0.3246	0.6702	0.6480	0.0410	0.6070	1.2772	2.7337
10	2.7337	1.8431	0.3246	0.6702	0.8484		0.6820	1.3522	1.3913

Return on Investment = 0.00360 Utility Ratio = 0.28671

Table 13. Equipment Specifications for the Conventional and Heat Pump Systems Using R-114 and a Steam Compressor

Item	No. Req.	Size Each	Material Const.
Condenser	1	5465.305 Sq. Ft.	Carbon Steel
Reboiler	8	9654.093 Sq. Ft.	Carbon Steel
Trim-Cooler	1	2988.887 Sq. Ft.	Carbon Steel
Innercooler	1	1647.495 Sq. Ft.	Carbon Steel
Start-Up Boiler	1	1297.314 Sq. Ft.	Carbon Steel
Compressor	2	2623.689 HP	Carbon Steel
Conventional System			
Condenser	1	6942.769 Sq. Ft.	Carbon Steel
Reboiler	1	1492.556 Sq. Ft.	Carbon Steel

Table 14. Purchased Equipment Cost for the Conventional and Heat Pump Systems Using R-114 and a Steam Compressor (in 1984 dollars)

Item	No. Req.	Cost Each	Total Cost
Condenser	1	73157.835	73157.835
Reboiler	8	104815.664	838525.313
Trim-Cooler	1	41632.189	41632.189
Innercooler	1	28571.967	28571.967
Start-Up Boiler	1	32428.412	32428.412
Compressor	2	298494.813	596989.625
Total			1611305.344
Conventional System			
Condenser	1	85101.495	85101.495
Reboiler	1	35432.788	35432.788
Total			120534.283

Table 15. Capital Investment for the Conventional System and Heat Pump System Using R-114 and a Steam Compressor (in millions of 1984 dollars)

Item	Heat Pump	Conventional
	Direct Costs	
Purchased Equipment	1.6113	0.1205
Installation	0.7573	0.0567
Control Systems	0.3384	0.0253
Piping	1.0635	0.0796
Electrical	0.2256	0.0169
Buildings	0.2900	0.0217
Yard Improvements	0.1611	0.0121
Service Facilities	1.1279	0.0844
Total Direct Cost	5.5751	0.4170
	Indirect Costs	
Engineering	0.5317	0.0398
Construction	0.6606	0.0494
Total	6.7675	0.5062
Contractor's Fee	0.3384	0.0253
Contingency	0.6767	0.0506
Fixed Capital	7.7826	0.5822
Working Capital	1.3857	0.1037
Total Capital	9.1683	0.6858
Total Additional Capital Investment for the Heat Pump System	8.4825	

Table 16. Heat Pump System Using R-114 and Steam Compressor Discounted Cash Flow with 1986 TVA Low Load Forecast

(millions of dollars)

Year	Investment at Start of Year	Annual Utility Savings	Annual Main. Expenses	Deprec.	Profit Before Tax	Federal Tax	Profit After Tax	After Tax Cash Flow	Investment at End of Year
1	8.4825	1.4869	0.2982	0.5472	0.6415	0.0405	0.6010	1.1482	7.7848
2	7.7848	1.4303	0.2982	0.9577	0.1745	0.0083	0.1662	1.1239	7.0745
3	7.0745	1.3783	0.2982	0.8208	0.2593	0.0141	0.2452	1.0660	6.3842
4	6.3842	1.3226	0.2982	0.6840	0.3404	0.0197	0.3207	1.0047	5.7186
5	5.7186	1.3598	0.2982	0.6840	0.3776	0.0223	0.3553	1.0393	4.9830
6	4.9830	1.3969	0.2982	0.6840	0.4147	0.1907	0.2240	0.9081	4.9830
7	4.3396	1.4303	0.2982	0.6156	0.5165	0.2376	0.2789	0.8946	3.6756
8	3.6756	1.4786	0.2982	0.6156	0.5648	0.2598	0.3050	0.9206	2.9502
9	2.9502	1.5157	0.2982	0.6156	0.6019	0.2769	0.3250	0.9407	2.1662
10	2.1662	1.5529	0.2982	0.6156	0.6391	0.2940	0.3451	0.9608	1.3205

Return on Investment = 0.05312 Utility Ratio = 0.24915

Table 17. Heat Pump System Using R-114 and Steam Compressor Discounted Cash Flow with 1986 TVA Medium Load Forecast

(millions of dollars)

Year	Investment at Start of Year	Annual Utility Savings	Annual Main. Expenses	Deprec.	Profit Before Tax	Federal Tax	Profit After Tax	After Tax Cash Flow	Investment at End of Year
1	8.4825	1.5715	0.2982	0.5472	0.7261	0.0464	0.6798	1.2270	7.8728
2	7.8728	1.5529	0.2982	0.9577	0.2971	0.0167	0.2803	1.2380	7.2078
3	7.2078	1.5157	0.2982	0.8208	0.3967	0.0236	0.3731	1.1939	6.5384
4	6.5384	1.4860	0.2982	0.6840	0.5038	0.0781	0.4257	1.1097	5.9045
5	5.9045	1.5529	0.2982	0.6840	0.5707	0.2625	0.3082	0.9922	5.3420
6	5.3420	1.6272	0.2982	0.6840	0.6450	0.2967	0.3483	1.0323	4.6985
7	4.6985	1.7089	0.2982	0.6156	0.7951	0.3658	0.4294	1.0450	3.9954
8	3.9954	1.7906	0.2982	0.6156	0.8768	0.4033	0.4735	1.0891	3.1971
9	3.1971	1.8760	0.2982	0.6156	0.9622	0.4426	0.5196	1.1352	2.2945
10	2.2945	1.9503	0.2982	0.6156	1.0365	0.4768	0.5597	1.1754	1.2861

Return on Investment = 0.07278 Utility Ratio = 0.27075

Table 18. Heat Pump System Using R-114 and Steam Compressor Discounted Cash Flow with 1986 TVA High Load Forecast

(millions of dollars)

Year	Investment at Start of Year	Annual Utility Savings	Annual Main. Expenses	Deprec.	Profit Before Tax	Federal Tax	Profit After Tax	After Tax Cash Flow	Investment at End of Year
1	8.4825	1.6717	0.2982	0.5472	0.8263	0.0533	0.7730	1.3203	7.9655
2	7.8655	1.6717	0.2982	0.9577	0.4159	0.0249	0.3909	1.3486	7.3712
3	7.3712	1.6717	0.2982	0.8208	0.5527	0.0344	0.5183	1.3392	6.7301
4	6.7301	1.6717	0.2982	0.6840	0.6895	0.3102	0.3793	1.0634	6.3041
5	6.3041	1.7832	0.2982	0.6840	0.8010	0.3685	0.4325	1.1166	5.7845
6	5.7845	1.9020	0.2982	0.6840	0.9198	0.4231	0.4967	1.1807	5.1515
7	5.1515	2.0320	0.2982	0.6156	1.1182	0.5144	0.6038	1.2195	4.4199
8	4.4199	2.1583	0.2982	0.6156	1.2445	0.5725	0.6720	1.2877	3.5508
9	3.5508	2.3069	0.2982	0.6156	1.3931	0.6408	0.7523	1.3679	2.5191
10	2.5191	2.4554	0.2982	0.6156	1.5416	0.7091	0.8325	1.4481	1.3096

Return on Investment = 0.09470 Utility Ratio = 0.28671

Table 19. Equipment Specifications for the Conventional and Heat Pump Systems Using R-113 and a Steam Compressor

Item	No. Req.	Size Each	Material Const.
Condenser	1	5394.071 Sq. Ft.	Carbon Steel
Reboiler	8	9308.844 Sq. Ft.	Carbon Steel
Trim-Cooler	1	2701.829 Sq. Ft.	Carbon Steel
Innercooler	1	2171.657 Sq. Ft.	Carbon Steel
Start-Up Boiler	1	1342.196 Sq. Ft.	Carbon Steel
Compressor	2	2688.796 HP	Carbon Steel
Conventional System			
Condenser		7035.549 Sq. Ft.	Carbon Steel
Reboiler		1492.131 Sq. Ft.	Carbon Steel

Table 20. Purchased Equipment Cost for the Conventional and Heat Pump Systems Using R-113 and a Steam Compressor (in 1984 dollars)

Item	No. Req.	Cost Each	Total Cost
Condenser	1	72553.747	72553.747
Reboiler	8	102430.820	819446.563
Trim-Cooler	1	39058.457	39058.457
Innercooler	1	34021.957	34021.957
Start-Up Boiler	1	33133.005	33133.005
Compressor	2	304555.230	609110.461
Total			1607324.188
Conventional System			
Condenser	1	85818.486	85818.486
Reboiler	1	35426.413	35426.413
Total			121244.899

Table 21. Capital Investment for the Conventional System and Heat Pump System Using R-113 and a Steam Compressor (in millions of 1984 dollars)

Item	Heat Pump	Conventional
	Direct Costs	
Purchased Equipment	1.6073	0.1212
Installation	0.7554	0.0570
Control Systems	0.3375	0.0255
Piping	1.0608	0.0800
Electrical	0.2250	0.0170
Buildings	0.2893	0.0218
Yard Improvements	0.1607	0.0121
Service Facilities	1.1251	0.0849
Total Direct Cost	5.5613	0.4195
	Indirect Costs	
Engineering	0.5304	0.0400
Construction	0.6590	0.0497
Total	6.7508	0.5092
Contractor's Fee	0.3375	0.0255
Contingency	0.6751	0.0509
Fixed Capital	7.7634	0.5856
Working Capital	1.3823	0.1043
Total Capital	9.1457	0.6899
Total Additional Capital Investment for the Heat Pump System	8.4558	

Table 22. Heat Pump System Using R-113 and Steam Compressor Discounted Cash Flow with 1986 TVA Low Load Forecast

(millions of dollars)

Year	Investment at Start of Year	Annual Utility Savings	Annual Main. Expenses	Deprec.	Profit Before Tax	Federal Tax	Profit After Tax	After Tax Cash Flow	Investment at End of Year
1	8.4558	1.4622	0.2972	0.5455	0.6195	0.0390	0.5805	1.1260	7.7583
2	7.7583	1.4074	0.2972	0.9546	0.1555	0.0070	0.1486	1.1032	7.0482
3	7.0482	1.3563	0.2972	0.8183	0.2408	0.0129	0.2280	1.0462	6.3591
4	6.3591	1.3016	0.2972	0.6819	0.3225	0.0185	0.3040	0.9859	5.6954
5	5.6954	1.3381	0.2972	0.6819	0.3590	0.0210	0.3380	1.0199	4.9641
6	4.9641	1.3746	0.2972	0.6819	0.3955	0.1466	0.2489	0.9308	4.2848
7	4.2848	1.4074	0.2972	0.6137	0.4965	0.2284	0.2681	0.8818	3.6201
8	3.6201	1.4549	0.2972	0.6137	0.5440	0.2502	0.2938	0.9075	2.8961
9	2.8961	1.4914	0.2972	0.6137	0.5805	0.2670	0.3135	0.9272	2.1157
10	2.1157	1.5279	0.2972	0.6137	0.6170	0.2838	0.3332	0.9469	1.2760

Return on Investment = 0.05067 Utility Ratio = 0.24915

Table 23. Heat Pump System Using R-113 and Steam Compressor Discounted Cash Flow with 1986 TVA Medium Load Forecast

(millions of dollars)

Year	Investment at Start of Year	Annual Utility Savings	Annual Main. Expenses	Deprec.	Profit Before Tax	Federal Tax	Profit After Tax	After Tax Cash Flow	Investment at End of Year
1	8.4558	1.5461	0.2972	0.5455	0.7034	0.0448	0.6586	1.2041	7.8485
2	7.8485	1.5279	0.2972	0.9546	0.2760	0.0153	0.2607	1.2154	7.1870
3	7.1870	1.4914	0.2972	0.8183	0.3759	0.0222	0.3537	1.1720	6.5222
4	6.5222	1.4622	0.2972	0.6819	0.4831	0.0456	0.4375	1.1193	5.8631
5	5.8631	1.5279	0.2972	0.6819	0.5488	0.2524	0.2964	0.9782	5.2987
6	5.2987	1.6009	0.2972	0.6819	0.6218	0.2860	0.3358	1.0177	4.6550
7	4.6550	1.6812	0.2972	0.6137	0.7703	0.3543	0.4160	1.0297	3.9538
8	3.9538	1.7615	0.2972	0.6137	0.8506	0.3913	0.4593	1.0730	3.1599
9	3.1599	1.8454	0.2972	0.6137	0.9345	0.4299	0.5046	1.1183	2.2646
10	2.2646	1.9184	0.2972	0.6137	1.0075	0.4634	0.5440	1.1577	1.2666

Return on Investment = 0.07057 Utility Ratio = 0.27075

Table 24. Heat Pump System Using R-113 and Steam Compressor Discounted Cash Flow with 1986 TVA High Load Forecast

(millions of dollars)

Year	Investment at Start of Year	Annual Utility Savings	Annual Main. Expenses	Deprec.	Profit Before Tax	Federal Tax	Profit After Tax	After Tax Cash Flow	Investment at End of Year
1	8.4558	1.6447	0.2972	0.5455	0.8020	0.0516	0.7504	1.2959	7.9429
2	7.9429	1.6447	0.2972	0.9546	0.3928	0.0234	0.3695	1.3241	7.3543
3	7.3543	1.6447	0.2972	0.8183	0.5292	0.0328	0.4965	1.3147	6.7206
4	6.7206	1.6447	0.2972	0.6819	0.6656	0.2737	0.3919	1.0737	6.2692
5	6.2692	1.7542	0.2972	0.6819	0.7751	0.3565	0.4186	1.1004	5.7493
6	5.7493	1.8710	0.2972	0.6819	0.8919	0.4103	0.4816	1.1635	5.1182
7	5.1182	1.9987	0.2972	0.6137	1.0878	0.5004	0.5874	1.2011	4.3910
8	4.3910	2.1228	0.2972	0.6137	1.2119	0.5575	0.6544	1.2681	3.5295
9	3.5295	2.2688	0.2972	0.6137	1.3579	0.6246	0.7333	1.3470	2.5094
10	2.5094	2.4148	0.2972	0.6137	1.5039	0.6918	0.8121	1.4258	1.3160

Return on Investment = 0.09260   Utility Ratio = 0.28671

this ratio is modified to remove the efficiency ratings used in preparation of the graphs (80 percent boiler efficiency and 95 percent motor efficiency) an equivalent present utility ratio of 0.59 was arrived at.

The options that could be considered regarding configurations are as follows: column overhead vapor as the working fluid, column bottom's liquid as the working fluid and temperature reservoirs. The column bottom's liquid was not considered in this study because the column operating pressure is just 1 atmosphere and this option would require a large vacuum in order to obtain a boiling liquid at a temperature below that of the condensing overhead fluid. The temperature reservoir option was not considered because of the need to expand this example to an entire plant for this option to be an improvement over the basic configuration, and the required information was not available. Therefore, only the overhead fluid as working fluid was considered.

The column overhead as working fluid option is illustrated in Figure 6 (p. 31). The major advantage of this option is the removal of the column condenser. The major drawback to this option is that the overhead fluid will be exposed to the compressor lubricant and the high temperature at the compressor discharge. The economics of this option are developed in Tables 25 through 30. This option has a favorable return on total investment. Tables 31 through 36 outline this option but with a steam turbine driven compressor instead of an

Table 25. Equipment Specifications for the Conventional System and Heat Pump System Using Overhead Fluid and Electric Compressor

Item	No. Req.	Size Each	Material Const.
Reboiler	1	7235.457 Sq. Ft.	Carbon Steel
Trim-Cooler	1	1150.477 Sq. Ft.	Carbon Steel
Innercooler	1	488.474 Sq. Ft.	Carbon Steel
Start-Up Boiler	1	1383.714 Sq. Ft.	Carbon Steel
Compressor	1	707.670 HP	Carbon Steel
Conventional System	1		
	1		
Condenser	1	7612.510 Sq. Ft.	Carbon Steel
Reboiler	1	1502.020 Sq. Ft.	Carbon Steel

Table 26. Purchased Equipment Cost for the Conventional System and Heat Pump System Using Overhead Fluid and Electric Compressor
(in 1984 dollars)

Item	No. Req.	Cost Each	Total Cost
Reboiler	1	87351.628	87351.628
Trim-Cooler	1	22770.998	22770.998
Innercooler	1	13251.188	13251.188
Start-Up Boiler	1	33777.116	33777.116
Compressor	1	88632.330	88632.330
Total			245783.264
Conventional System			
Condenser	1	90201.575	90201.575
Reboiler	1	35574.620	35574.620
Total			125776.195

Table 27. Capital Investment for the Conventional System and Heat Pump System Using Overhead Fluid and Electric Compressor (in millions of 1984 dollars)

Item	Heat Pump	Conventional
	Direct Costs	
Purchased Equipment	0.2458	0.1258
Installation	0.1155	0.0591
Control Systems	0.0516	0.0264
Piping	0.1622	0.0830
Electrical	0.0344	0.0176
Buildings	0.0442	0.0226
Yard Improvements	0.0246	0.0126
Service Facilities	0.1720	0.0880
Total Direct Cost	0.8504	0.4352
	Indirect Costs	
Engineering	0.0811	0.0415
Construction	0.1008	0.0516
Total	1.0323	0.5283
Contractor's Fee	0.0516	0.0264
Contingency	0.1032	0.0528
Fixed Capital	1.1871	0.6075
Working Capital	0.2114	0.1082
Total Capital	1.3985	0.7157
Total Additional Capital Investment for Heat Pump System	0.6828	

Table 28. Heat Pump System Using Overhead Fluid and Electric Compressor Discounted Cash Flow with 1986 TVA Low Load Forecast
(millions of dollars)

Year	Investment at Start of Year	Annual Utility Savings	Annual Main. Expenses	Deprec.	Profit Before Tax	Federal Tax	Profit After Tax	After Tax Cash Flow	Investment at End of Year
1	0.6828	0.4801	0.0240	0.0441	0.4120	0.1316	0.2805	0.3245	0.6275
2	0.6275	0.4575	0.0240	0.0771	0.3564	0.1639	0.1925	0.2696	0.6054
3	0.6054	0.4370	0.0240	0.0661	0.3469	0.1596	0.1873	0.2534	0.5906
4	0.5906	0.4144	0.0240	0.0551	0.3353	0.1543	0.1811	0.2361	0.5873
5	0.5873	0.4332	0.0240	0.0551	0.3541	0.1629	0.1912	0.2463	0.5726
6	0.5726	0.4524	0.0240	0.0551	0.3733	0.1717	0.2016	0.2567	0.5416
7	0.5416	0.4696	0.0240	0.0496	0.3960	0.1822	0.2139	0.2634	0.4917
8	0.4917	0.4922	0.0240	0.0496	0.4186	0.1926	0.2261	0.2756	0.4100
9	0.4100	0.5114	0.0240	0.0496	0.4378	0.2014	0.2364	0.2860	0.2856
10	0.2856	0.5315	0.0240	0.0496	0.4579	0.2107	0.2473	0.2968	0.1014

Return on Investment = 0.39424 Utility Ratio = 0.24915

Table 29. Heat Pump System Using Overhead Fluid and Electric Compressor Discounted Cash Flow with 1986 TVA Medium Load Forecast
(millions of dollars)

Year	Investment at Start of Year	Annual Utility Savings	Annual Main. Expenses	Deprec.	Profit Before Tax	Federal Tax	Profit After Tax	After Tax Cash Flow	Investment at End of Year
1	0.6828	0.5116	0.0240	0.0441	0.4435	0.1461	0.2975	0.3415	0.6418
2	0.6418	0.5024	0.0240	0.0771	0.4013	0.1846	0.2167	0.2938	0.6305
3	0.6305	0.4859	0.0240	0.0661	0.3958	0.1821	0.2137	0.2798	0.6281
4	0.6281	0.4721	0.0240	0.0551	0.3930	0.1808	0.2122	0.2673	0.6372
5	0.6372	0.5015	0.0240	0.0551	0.4224	0.1943	0.2281	0.2832	0.6344
6	0.6344	0.5328	0.0240	0.0551	0.4537	0.2087	0.2450	0.3001	0.6135
7	0.6135	0.5681	0.0240	0.0496	0.4945	0.2275	0.2671	0.3166	0.5669
8	0.5669	0.6039	0.0240	0.0496	0.5303	0.2440	0.2864	0.3359	0.4804
9	0.4804	0.6407	0.0240	0.0496	0.5671	0.2609	0.3063	0.3558	0.3360
10	0.3360	0.6733	0.0240	0.0496	0.5997	0.2759	0.3239	0.3734	0.1104

Return on Investment = 0.44007 Utility Ratio = 0.27075

Table 30. Heat Pump System Using Overhead Fluid and Electric Compressor Discounted Cash Flow with 1986 TVA High Load Forecast
(millions of dollars)

Year	Investment at Start of Year	Annual Utility Savings	Annual Main. Expenses	Deprec.	Profit Before Tax	Federal Tax	Profit After Tax	After Tax Cash Flow	Investment at End of Year
1	0.6828	0.5462	0.0240	0.0441	0.4781	0.1620	0.3162	0.3602	0.6539
2	0.6539	0.5426	0.0240	0.0771	0.4415	0.2031	0.2384	0.3155	0.6556
3	0.6556	0.5382	0.0240	0.0661	0.4481	0.2061	0.2420	0.3081	0.6656
4	0.6656	0.5324	0.0240	0.0551	0.4533	0.2085	0.2448	0.2999	0.6886
5	0.6886	0.5770	0.0240	0.0551	0.4979	0.2290	0.2689	0.3239	0.6986
6	0.6986	0.6239	0.0240	0.0551	0.5448	0.2506	0.2942	0.3493	0.6883
7	0.6883	0.6755	0.0240	0.0496	0.6019	0.2769	0.3250	0.3746	0.6476
8	0.6476	0.7251	0.0240	0.0496	0.6515	0.2997	0.3518	0.4014	0.5603
9	0.5603	0.7845	0.0240	0.0496	0.7109	0.3270	0.3839	0.4335	0.3986
10	0.3986	0.8435	0.0240	0.0496	0.7699	0.3542	0.4158	0.4653	0.1267

Return on Investment = 0.48511 Utility Ratio = 0.28671

Table 31. Equipment Specifications for the Conventional System and Heat Pump System Using Overhead Fluid and Steam Compressor

Item	No. Req.	Size Each	Material Const.
Reboiler	1	3609.149 Sq. Ft.	Carbon Steel
Trim-Cooler	1	1166.021 Sq. Ft.	Carbon Steel
Innercooler	1	384.547 Sq. Ft.	Carbon Steel
Start-Up Boiler	1	1383.714 Sq. Ft.	Carbon Steel
Compression	1	976.491 HP	Carbon Steel
Conventional System			
Condenser	1	7612.510 Sq. Ft.	Carbon Steel
Reboiler	1	1502.020 Sq. Ft.	Carbon Steel

Table 32. Purchased Equipment Cost for the Conventional System and Heat Pump System Using Overhead Fluid and Steam Compressor
(in 1984 dollars)

Item	No. Req.	Cost Each	Total Cost
Reboiler	1	56281.854	56281.854
Trim-Cooler	1	22964.966	22964.966
Innercooler	1	11391.871	11391.871
Start-Up Boiler	1	33777.116	33777.116
Compressor	1	132726.164	1322726.164
Total			257141.973
Conventional System			
Condenser	1	90201.575	90201.575
Reboiler	1	35574.620	35574.620
Total			125776.195

Table 33. Capital Investment for the Conventional System and Heat Pump System Using Overhead Fluid and Steam Compressor (in millions of 1984 dollars)

Item	Heat Pump	Conventional
	Direct Costs	
Purchased Equipment	0.2571	0.1258
Installation	0.1209	0.0591
Control Systems	0.0540	0.0264
Piping	0.1697	0.0830
Electrical	0.0360	0.0176
Buildings	0.0463	0.0226
Yard Improvements	0.0257	0.0126
Service Facilities	0.1800	0.0880
Total Direct Cost	0.8897	0.4352
	Indirect Costs	
Engineering	0.0849	0.0415
Construction	0.1054	0.0516
Total	1.0800	0.5283
Contractor's Fee	0.0540	0.0264
Contingency	0.1080	0.0528
Fixed Capital	1.2420	0.6075
Working Capital	0.2211	0.1082
Total Capital	1.4631	0.7157
Total Additional Capital Investment for Heat Pump System	0.7475	

Table 34. Heat Pump System Using Overhead Fluid and Steam Compressor Discounted Cash Flow with 1986 TVA Low Load Forecast
(millions of dollars)

Year	Investment at Start of Year	Annual Utility Savings	Annual Main. Expenses	Deprec.	Profit Before Tax	Federal Tax	Profit After Tax	After Tax Cash Flow	Investment at End of Year
1	0.7475	0.5530	0.0263	0.0482	0.4785	0.1567	0.3218	0.3701	0.6920
2	0.6920	0.5357	0.0263	0.0844	0.4250	0.1955	0.2295	0.3139	0.6693
3	0.6693	0.5196	0.0263	0.0723	0.4210	0.1937	0.2273	0.2997	0.6513
4	0.6513	0.5023	0.0263	0.0603	0.4157	0.1912	0.2245	0.2848	0.6406
5	0.6406	0.5138	0.0263	0.0603	0.4272	0.1965	0.2307	0.2910	0.6193
6	0.6193	0.5253	0.0263	0.0603	0.4387	0.2018	0.2369	0.2972	0.5827
7	0.5827	0.5357	0.0263	0.0542	0.4552	0.2094	0.2458	0.3000	0.5278
8	0.5278	0.5506	0.0263	0.0542	0.4701	0.2162	0.2538	0.3081	0.4419
9	0.4419	0.5622	0.0263	0.0542	0.4817	0.2216	0.2601	0.3144	0.3135
10	0.3135	0.5737	0.0263	0.0542	0.4932	0.2269	0.2663	0.3206	0.1249

Return on Investment = 0.42085 Utility Ratio = 0.24915

Table 35. Heat Pump System Using Overhead Fluid and Steam Compressor Discounted Cash Flow with 1986 TVA Medium Load Forecast
(millions of dollars)

Year	Investment at Start of Year	Annual Utility Savings	Annual Main. Expenses	Deprec.	Profit Before Tax	Federal Tax	Profit After Tax	After Tax Cash Flow	Investment at End of Year
1	0.7475	0.5794	0.0263	0.0482	0.5049	0.1688	0.3361	0.3843	0.7043
2	0.7043	0.5737	0.0263	0.0844	0.4630	0.2130	0.2500	0.3344	0.6913
3	0.6913	0.5622	0.0263	0.0723	0.4636	0.2133	0.2503	0.3227	0.6841
4	0.6841	0.5530	0.0263	0.0603	0.4664	0.2146	0.2519	0.3122	0.6841
5	0.6841	0.5737	0.0263	0.0603	0.4871	0.2241	0.2631	0.3233	0.6730
6	0.6730	0.5967	0.0263	0.0603	0.5101	0.2347	0.2755	0.3358	0.6443
7	0.6443	0.6220	0.0263	0.0542	0.5415	0.2491	0.2924	0.3466	0.5918
8	0.5918	0.6473	0.0263	0.0542	0.5668	0.2607	0.3061	0.3603	0.5015
9	0.5015	0.6738	0.0263	0.0542	0.5933	0.2729	0.3204	0.3746	0.3558
10	0.3558	0.6968	0.0263	0.0542	0.6163	0.2835	0.3328	0.3870	0.1311

Return on Investment = 0.45638 Utility Ratio = 0.27075

Table 36. Heat Pump System Using Overhead Fluid and Steam Compressor Discounted Cash Flow with 1986 TVA High Load Forecast
(millions of dollars)

Year	Investment at Start of Year	Annual Utility Savings	Annual Main. Expenses	Deprec.	Profit Before Tax	Federal Tax	Profit After Tax	After Tax Cash Flow	Investment at End of Year
1	0.7475	0.6105	0.0263	0.0482	0.5360	0.1831	0.3529	0.4011	0.7155
2	0.7155	0.6105	0.0263	0.0844	0.4998	0.2299	0.2699	0.3543	0.7147
3	0.7147	0.6105	0.0263	0.0723	0.5119	0.2355	0.2764	0.3488	0.7189
4	0.7189	0.6105	0.0263	0.0603	0.5239	0.2410	0.2829	0.3432	0.7307
5	0.7307	0.6450	0.0263	0.0603	0.5584	0.2569	0.3016	0.3618	0.7298
6	0.7298	0.6818	0.0263	0.0603	0.5952	0.2738	0.3214	0.3817	0.7085
7	0.7085	0.7221	0.0263	0.0542	0.6416	0.2951	0.3465	0.4007	0.6578
8	0.6578	0.7612	0.0263	0.0542	0.6807	0.3131	0.3676	0.4218	0.5609
9	0.5609	0.8073	0.0263	0.0542	0.7268	0.3343	0.3925	0.4467	0.3912
10	0.3912	0.8533	0.0263	0.0542	0.7728	0.3555	0.4173	0.4715	0.1128

Return on Investment = 0.49391 Utility Ratio = 0.28671

electrically driven compressor. In both of these systems, a trim-heater is required in spite of the heat duty ratio of 1.019. This is due to the reduction in the latent heat of vaporization of the overhead fluid as the temperature is increased (see Figure 2, p. 5).

The results of this study of heat pump system options are summarized in Table 37. It is apparent that the economic feasibility of these options have major effects upon a heat pump system. A heat pump system should be considered when the required utility ratio (from the graphs in the previous chapter) is less than the actual utility ratio, less than 0.59 for a steam driven compressor, or when the overhead fluid can be used as the heat pump working fluid.

The most promising option for the ethanol-water separation column appears to be the overhead fluid as the heat pump refrigerant. All three components have high enough critical temperatures that the exposure to the compressor discharge temperature is not a significant drawback. The ethanol product is to be fuel grade so contamination by exposure to the compressor lubricant is also not significant provided that lubricant loss is minimized. The return on the additional investment of 45 percent is favorable.

The use of a steam driven compressor showed promise but failed to show a favorable return on the additional investment. The return on investment was 0.07 while the electrically driven heat pump system has a return on additional investment of -0.02.

Table 37. Investment Summary (Millions of 1984 Dollars)

Option	Working Fluid	Compressor	Additional Investment	Return on Additional Investment TVA Load Forecast		
				Low	Medium	High
External	R-114	Electric	8.3765	-0.0714	-0.0222	0.0130
External	R-113	Electric	9.2339	-0.0808	-0.0328	0.0036
External	R-114	Steam	8.4825	0.0531	0.0728	0.0947
External	R-113	Steam	8.4558	0.0507	0.0706	0.0926
Overhead	Overhead	Electric	0.6826	0.3942	0.4401	0.4851
Overhead	Overhead	Steam	0.7475	0.4209	0.4564	0.4939

CHAPTER XII

CONCLUSIONS AND RECOMMENDATIONS

This project was divided into two parts. In the first part the effects of five variables on the economic feasibility of a heat pump system were studied. The first of these variables, the utility ratio, was defined as the ratio of the natural gas rate to the electric rate on a dollar per million BTU basis. The utility ratio was used as a measure of economic performance by determining the utility ratio required for the heat pump system to be economically feasible and comparing it to the actual value of the utility ratio predicted.

The second variable, the reboiler heat duty, was found to have a major impact when the heat pump system was economically feasible. The heat pump system was favored only where the reboiler heat duty was large. This is due to the benefits increasing faster than the cost of the heat pump system with increases in the reboiler heat duty.

The third variable, the heat duty ratio defined as the ratio of the condenser heat duty to that of the reboiler, was found to also have a major impact. The heat pump system was most economically feasible if the heat duty ratio was such that neither a trim-cooler nor a trim-heater were required. The cost of supplying the heat pump system with additional cooling or heating becomes prohibitive and the required utility ratio increases rapidly as the heat duty ratio moves away from the no trim-heater point.

The fourth variable, the condenser temperature, was found to not have a major impact on the economic feasibility of the heat pump system. It caused slight reductions in the required utility ratio with increases in the condenser temperature. This was caused by increases in the refrigerant circulation rate offset by reductions in the compression ratio as the condenser temperature was increased.

The fifth variable, the temperature difference between the reboiler and condenser on the process side, caused large increases in the required utility ratio as it increased. This was caused by increases in both the refrigerant circulation rate and compression ratio as the temperature difference increased.

In the second part, a specific column is used to compare heat pump system options on the basis of additional investment and the return on that additional investment. The specific column chosen was an azeotropic distillation column used in wood-to-ethanol plant. The required utility ratio was greater than the utility ratio projected by TVA and the heat pump system utilizing an external refrigerant cycle without an option was not favorable as expected.

The option of using a steam turbine to power the compressor was a promising one. This option increases the cost of the compressor but reduces the energy cost to drive the compressor. This was an improvement over the electrically driven heat pump system but was not economically feasible at this time.

The final option considered was that of using the column overhead fluid as the heat pump refrigerant. This option reduces the capital investment of the heat pump system significantly by removing the condenser. It was found to be a favorable option with a return on the required additional investment to be 44 percent.

There are many more options that were not included in this study that hold considerable potential. Among these are enhanced surface heat exchangers. In this set of options, the tubes of a heat exchanger are altered to increase the film heat transfer coefficients or to increase the heat transfer area per length of tube. The methods of altering tube surfaces include teflon coating to decrease surface wetting when a fluid is condensing outside tubes, porous surface coatings to increase the number of nucleation sites for a boiling fluid, and fins which increase the heat transfer area per length of tube. These methods can reduce the number of tubes required by as much as 60 percent but increase the cost of each tube substantially.

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APPENDIX

APPENDIX

Three programs were written to aid in the analysis required in this thesis. The first, Feglin.For, was used to determine the required utility ratio of a column or the required utility ratios of a uniform progression of column sizes. It requires the input of the column condenser temperature, the temperature difference between the reboiler and condenser on the process side, the reboiler heat duty, and the ratio of the condenser heat duty to that of the reboiler. If a uniform progression is desired, a code is used to choose which variable is to increase, by how much and the upper limit of the variable chosen. All except the reboiler heat duty are assumed to be arithmetic progressions while the reboiler heat duty is assumed to be geometric. Feglin-For also requires the input of the forecasted electric rates for each year in the life span of the project and the constants necessary to estimate the physical properties of both water and the chosen refrigerant. Feglin-For is a Fortran program that occupies 141 blocks of memory on the DEC-10. It is 2,428 lines long and has two subroutines.

The second program is Extern.For, and it was used to determine the return on the additional investment required by a heat pump system utilizing an external refrigerant cycle. It requires the same input as Feglin.For but the column condenser temperatures vary along the condenser and both the inlet and exit temperatures of the column overhead fluid are required. In addition, an input of the exit compositions of both the overhead fluid and bottoms liquid as well as their mass flow rates and pressures are required. The constants necessary to estimate the physical properties, the UNIFAC liquid activity coefficients and forecasted natural gas rates for each year in the life span of the project are also required. A series of six tables is generated by this program. These tables are an equipment specification table of major equipment, purchased equipment cost table, capital investment table, and three discounted cash flow tables, one for each forecast. Extern.For is a Fortran program that occupies 163 blocks of memory on the DEC-10. It is 3,011 lines long and has four subroutines.

The third program, Overhd.For, was used to determine the return on additional investment for a heat pump system that utilizes the column overhead fluid as the heat pump refrigerant. It requires the same input as Extern.For and produces a similar series of tables. It occupies 158 blocks of memory on the DEC-10 and is 3,125 lines long. It is a Fortran program with four subroutines.

Listings of these three programs are available through the Department of Chemical Engineering, The University of Tennessee (Attention Dr. J. M. Holmes), Dougherty Engineering Building, Knoxville, TN 37996.

VITA

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