DESIGN OF A TUBE BANK HEAT RECLAIMER FOR RESIDENTIAL HEATING SYSTEMS

by

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Submitted in Partial Fulfillment of the Requirement

for the Degree of

Master of Science in Engineering

in the

Mechanical Engineering

Program

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YOUNGSTOWN STATE UNIVERSITY

March, 1987

ABSTRACT

DESIGN OF A TUBE BANK HEAT RECLAIMER FOR RESIDENTIAL HEATING SYSTEMS

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Forced convection tube bank heat reclaimers are analyzed in detail for residential natural gas and oil-fired furnaces, that are vented by natural draft. Optimum reclaimer designs are obtained based on improved system efficiency, and considerations regarding manufacturing costs, which include reclaimer size, materials used, and overall weight. Each reclaimer meets safety restrictions regarding allowable system pressure losses and minimum chimney gas temperatures. Computer-generated solutions aid in determining heat recovery as a function of furnace fuel, furnace efficiency, ambient temperature, flue pipe size, and chimney height. The analysis considers a range of furnace efficiencies from 50 to 80%, and ambient temperatures from 0 to 60°F, which are values considered typical for most domestic combustion heating equipment. Flue pipe sizes range from 4 to 6 inches in diameter and are 2 to 4 feet long. Chimney sizes range from 5 to 7 inches in equivalent diameter and include draft heights from 15 to 35 feet. The piping sizes correspond to furnace input capacities ranging from 50,000 to 170,000 Btu/h. For many domestic heating systems, the potential exists to recover the lost heat by as much as 30%, and to reduce fuel costs by as much as 15% by installing a flue pipe heat reclaimer.

ACKNOWLEDGEMENTS

The author acknowledges Dr. Thomas I. Elias for guidance and support, and Youngstown State University for partial funding and computer time.

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LIST OF SYMBOLS

SYMBOL	DEFINITION	UNITS
a	Reclaimer casing width	in, ft
A	Area	in^2, ft^2
A_d	Surface area of reclaimer as a rectangular duct	in^2, ft^2
b	Reclaimer casing depth	in, ft
B	Barometric pressure	in. mercury
c	Reclaimer casing length	in, ft
C_m, C_n	Correction variables for small numbers of tube rows	-
C_o	Correction variable for duct geometry change	-
c_p	Specific heat at constant pressure	Btu/lbm-^oF
C_p	Specific heat for one mole of flue gas constituent at constant pressure	Btu/lbmol-^oF
C_R	Correction variable for chimney flue gas rise	-
C_u	Chimney temperature multiplier	-
D	Outside tube diameter	in, ft
D_a	Available draft	in. water
d_c	Chimney inside equivalent diameter	in, ft
D_H	Hydraulic diameter	ft
d_i	Inside tube diameter	in, ft
D_t	Theoretical draft	in. water
Eu	Euler number	-
f	Friction factor for pressure drop	-
g_1	Gravity constant	ft/s^2
h	Average heat-transfer coefficient	Btu/h-ft^2-^oF
H	Chimney height	ft
Δh	Change in enthalpy	Btu/lbmol

LIST OF SYMBOLS continued

SYMBOL	DEFINITION	UNITS
$\bar{h}_f$	Enthalpy of formation	Btu/lbmol
H_P	Enthalpy of the products	Btu/lbmol
H_R	Enthalpy of the reactants	Btu/lbmol
I	Furnace input	Btu/h
k	Thermal conductivity	Btu/h-ft-^oF
k_1	Euler number geometry constant	-
K, K_1	Power law variables	-
k_t	Combined system resistance coefficients	-
L	Flue pipe horizontal breeching	ft
L_e	Equivalent flue pipe breeching	ft
$\dot{m}$	Mass flow rate	lbm/h
M	Mass flow-furnace input ratio	lbm/1000-Btu
m_i	Mass fraction of a constituent	-
M_i	Molecular weight of a constituent	-
m_{fi}	Mass fraction-molecular weight ratio	-
n	Number of tube rows	-
n_i	Moles of a constituent	-
n_P	Moles of product	-
n_R	Moles of reactant	-
n_t	Number of tubes	-
Nu	Nusselt number	-
P	Pressure	in. water
ΔP	Pressure drop	in. water
Pr_1	Prandtl number at upstream conditions	-
Pr_f	Prandtl number at film conditions	-

LIST OF SYMBOLS continued

SYMBOL	DEFINITION	UNITS
Pr_t	Prandtl number at the tube surface conditions	-
Q	Heat transfer	Btu/h
Q_g	Total heat loss from flue gas	Btu/h
Q_o	Furnace output	Btu/h
Re	Reynolds number	-
Re_d	Reynolds number in unobstructed reclaimer duct	-
S_D	Diagonal tube spacing	in, ft
S_L	Longitudinal tube spacing	in, ft
S_T	Transverse tube spacing	in, ft
T	Temperature	^oF, ^oR
ΔT	Temperature change	^oF
$\hat{T}_{f2}$	Furnace temperature at design conditions	^oF, ^oR
T_R	Chimney flue gas temperature rise	^oF
$\hat{T}_R$	Temperature rise at design conditions	^oF
T_s	Design temperature for draft analysis	^oF
u_1	Reclaimer upstream flow velocity	ft/h
u_m	Reclaimer maximum flow velocity	ft/h
u_ϕ	Average tube bank flow velocity as a function of the flow separation angle	ft/h
U	Overall heat-transfer coefficient	Btu/h-ft^2-^oF
V	Flue pipe and chimney flue gas velocity	fpm, ft/s
V_d	Volume of reclaimer without a tube bank	ft^3
V_t	Volume of tube bank	ft^3
W_b	Weight of blower and accessories	lbs
W_T	Total reclaimer weight	lbs

LIST OF SYMBOLS continued

SYMBOL	DEFINITION	UNITS
y_i	Mole fraction	-
Θ	Reference temperature	$^oR/180$
γ_t	Weight of tubing per linear foot	lb/ft
γ_w	Weight of steel sheet per square foot	lb/ft^2
ε	Emissivity	-
η	Furnace efficiency	-
μ	Dynamic viscosity	lbm/ft-h
μ_m	Dynamic viscosity at bulk conditions	lbm/ft-h
ρ	Density	lbm/ft^3
ρ_m	Density at bulk conditions	lbm/ft^3
σ	Stefan-Boltzmann constant	$Btu/h\text{-}ft^2\text{-}{}^oR^4$
ϕ	Angle of separation	rad.

LIST OF SUBSCRIPTS

SUBSCRIPT	DEFINITION
a	Air
a1	Air inlet and surrounding air conditions
a2	Air outlet conditions
am	Air bulk conditions
b	Basement
bi	Basement below grade conditions
bo	Basement above grade conditions
c	Chimney
c1	Chimney inlet conditions
c2	Chimney exit conditions
cm	Chimney mean conditions
co	Average exterior chimney conditions
f1	Furnace flame conditions
f2	Furnace exit conditions
g	Flue gas
g1	Reclaimer flue gas inlet conditions
g2	Reclaimer flue gas outlet conditions
gm	Flue gas bulk conditions
p	Flue pipe
P	Combustion products
p1	Flue pipe conditions at reclaimer inlet
p2	Flue pipe conditions at reclaimer exit
R	Combustion reactants
t	Tube surface
ti	Tube interior surface

LIST OF SUBSCRIPTS continued

SUBSCRIPT	DEFINITION
to	Tube exterior surface
w	Reclaimer casing surface
wi	Reclaimer interior surface
wo	Reclaimer exterior surface

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CHAPTER I

INTRODUCTION

Fossil fuels such as coal, petroleum, and natural gas constitute approximately 90% of America's energy sources [5]. One is reminded that these are dwindling natural resources when fuel and electric bills begin to rise. Therefore, it is important to demonstrate concern for the future needs by conserving energy today.

1.1 Waste Heat Recovery and Utilization

The industrial sector is responsible for approximately 40% of the nation's total energy use [10]. Due to the low costs of fossil fuel 20 to 40 years ago, American industry was not persuaded into efficient energy use. Even in the past decade, it has been determined that 50% of industrial energy was lost as waste heat to the environment [4].

Typical examples of waste heat in industry include those from furnaces, ovens, process plants, and incinerators. Most of this waste heat is flue gas resulting from the combustion of fossil fuels, and is dispersed into the atmosphere. As fuel costs rise, the concepts regarding waste heat utilization become more apparent.

Europe is perhaps several years ahead of the United States in waste heat utilization technology, because the impact of rising fuel costs is more severe in European nations [7]. Currently, waste heat utilization exists in most major North American industries, and new sources and uses are being discovered each day.

Besides saving on fuel costs, industrial utilization of flue gas waste heat also reduces air pollution. Some American industries have been forced into utilizing waste heat from flue gases, because of air pollution regulations [28]. Precipitating flue gas is a common method of separating toxic molecules, and is accomplished through condensing flue gases by lowering the temperature.

Sources and uses of waste heat are not restricted to industrial applications. Residential energy use amounts to 20% of the total national energy consumption, and more than half of this is for space heating purposes [10]. Although electric heat has been increasing steadily since 1950, fossil fuels still dominate as the major residential heating source. Of the three predominant fossil fuels (coal, petroleum, and natural gas), coal has decreased in use from the most widely used fuel in 1950, to a fuel rarely used today. As of 1980, natural gas is America's top residential heating source, followed by oil and electricity [10].

During the energy crisis of the 1970's, home owners were stunned by rapidly increasing heating costs: 10% for electricity, 22% for natural gas, and 57% for oil, between 1972 and 1976 [24]. This rise contributed greatly to improved home heating efficiency, and soon, many energy saving materials and devices were available on the market. Also, there was an increase in alternative home energy sources, such as solar and wind power, portable heaters, and wood burning stoves, to name a few. Although many of these energy conservation measures have merit, conservation regarding warm-air heating furnaces will be emphasized here.

Heating a house with electricity produces a seasonal efficiency of 100%, where seasonal efficiency refers to the overall heating efficiency, averaged on a yearly basis. Electric heat, however, is still more expensive than comparable fossil fuel heat (assuming efficient combustion heating equipment). Electric heat pumps are also very efficient, but their life expectancy and the cold North American climates cannot often justify their use. In general, electric heat holds great promise in the future; however, due to the cost of replacing an existing combustion heating system and the leveling of fossil fuel prices, natural gas and oil will continue as the major residential heating sources.

There have been substantial improvements in the technology of combustion heating equipment. Natural gas pulsating furnaces that have steady-state efficiencies well above 90% are currently available, where steady-state efficiency refers to the system efficiency when the furnace has reached a near steady-state condition. These furnaces condense flue gases, but the condensate is properly drained from the system, and the remaining gases are expelled from the house in plastic chimneys. However, the cost of replacement and the uncertainty of future natural gas prices have discouraged many existing home owners from converting to these high efficiency furnaces.

Currently, many North American homes that were built prior to 1950 have outdated heating equipment. Many of the coal furnaces that were popular during this period were converted to natural gas furnaces, and by today's standards, these furnaces are very inefficient. Prior to 1980, most of the new natural gas furnaces operated with steady-state efficiencies below 80% [3]. For oil furnaces the efficiency was 1 to 2% greater, because there was less water vapor in the combustion flue gases, which reduced the amount of latent heat loss. The maximum steady-state efficiency that a combustion heating system can attain, before condensation occurs, is approximately 84%.

The concern here is with older combustion heating equipment, where the average steady-state efficiencies are below 80%. It is presumed in this thesis that these systems constitute the majority of urban North American homes, and will continue to dominate for at least the next 20 years.

Investigation into reasons for low efficiency heating systems may provide key insights for possible furnace upgrading. Oversizing heating systems was common practice years ago, and contributed to the heating system inefficiency. In many cases, oversizing (use of larger flue pipe diameters and inside chimney liners) amounted to as much as 200%, which reduced the system steady-state efficiency by 20% [3]. Oversizing assured a lower flue gas velocity and system pressure drop, which translated to a better factor of safety against flue gas back-flow and leakage. Other reasons for oversizing included uncertainty in calculations, compensation for severe wind conditions, and the low cost of fuel.

There is the continuous depletion of furnace air through the draft regulator, especially after the furnace cycles off. In natural draft heating systems, combustion air, usually from the furnace surroundings, is constantly being drawn through the furnace or draft regulator, due to the draft created between the inside and outside air density differences. On very cold days, a substantial amount of heat can be lost, and this heat loss reduces the system seasonal efficiency. Also, in many older gravity-air heating systems, much of the heat remaining inside the furnace after the furnace is cycled off is lost out the chimney. The exact amount of heat loss depends largely on weather conditions, oversizing, and furnace cycle behavior.

The furnace combustion system also contributes to the inefficiency in the older heating equipment. Several factors that affect the combustion efficiency include the air-fuel ratio, the temperature of the combustion air, and the flame behavior. Recent developments in the field of improved burner efficiency have increased combustion efficiencies over 80%, by simply reducing the burner nozzle size [25]. This forces the furnace to run for longer periods of time and burn the fuel more completely, which results in a higher combustion efficiency. The combustion efficiency can also be improved by periodically cleaning the burners and checking the combustion chamber for cracks or leaks.

Many of the existing combustion furnaces in North America are simply outdated or are in need of cleaning. Developments in residential furnace heat exchangers have not only reduced the size of the heating appliance, but are more effective and resistant to fouling. In many cases, keeping an existing heat exchanger clean is the only method of improving furnace efficiency.

It is clear that a potential exists with older combustion heating equipment to capture and utilize flue gas waste heat, which is normally lost through the chimney. A device inserted into the flue pipe, for the purpose of capturing waste heat, is defined as a heat reclaimer. A heat reclaimer is an energy-saving device that reduces fuel costs for inefficient home heating systems.

1.2 Statement of the Problem

The purpose of this thesis is to optimize tube bank heat exchangers that will be used as heat reclaimers for natural draft, residential combustion heating systems which burn either natural gas or No.2 heating oil. The reclaimed heat will be used as additional space heat.

In general, the thesis investigates heat transfer and pressure drop due to flow of flue gases through various in-line and staggered tube-bank arrangements, and analyzes specific reclaimer designs for various residential heating systems. Optimized designs are obtained using the computer.

The optimum reclaimer designs are determined in terms of safe operation, heat recovered, reclaimer size and weight. Safe operation is ensured by maintaining a minimum chimney updraft of 0.05 inches (of water column), to prevent flue gas leakage into living areas, and requiring that the flue gas temperature throughout the system remain above its dew-point, to prevent possible corrosion due to flue-gas condensation.

The optimum reclaimer designs are further investigated for typical applications in North American homes, in which furnace efficiencies and ambient temperatures are varied. Reclaimer heat output, percentage heat recovered, and fuel saved, are studied as a function of the furnace steady-state efficiency and ambient temperature.

1.3 Previous Work on Heat Reclaimers and Tube Bank Heat Exchangers

During the energy crisis of the 1970's, several residential heat reclaimers appeared on the market [19]. Tube bank heat reclaimers were found to be the most effective type, showing a fuel savings of 3 to 6% using an oil-fired test furnace. The specific reclaimer performance varied according to the flue gas temperature, which is a function of the fuel type, the furnace size and efficiency, and system oversizing.

Feldman and Tsai [18] incorporated finned heat pipes into tube bank heat reclaimers, and showed that the reclaimer could be sold at a lower cost than comparable existing tube bank heat reclaimers. They also showed that potential for heat recovery also exists in residential hot water heaters and clothes dryers, with reasonable payback periods.

Several other concepts for residential flue gas heat recovery have been studied by a number of investigators. Gorman [21] proposed an efficient method of flue gas heat recovery, using a pebble or rock storage bed to capture both the sensible and latent flue gas heat. This concept could improve a natural gas furnace efficiency from 55 to 96 percent. Deniega [17] suggested the preheating of boiler feedwater by the use of a helical water tube wrapped around the flue pipe.

Domestic heat reclaiming devices, in order to conform to current building codes, usually must be approved by a national testing organization, such as UL (Underwriters Laboratories) or AGA (American Gas Association). Such organizations have perhaps the best consumer testing facilities in the nation, and provide important information regarding residential heat reclaimers. UL and AGA directories provide the names and addresses of current manufacturers of residential heat reclaimers. Also, UL and AGA testing standards outline test procedures for residential heat reclaimers, which can be used as guidelines when developing reclaimers.

Flow through tube banks, which comprises a large area of convective heat transfer, has been studied by a number of investigators. Currently, most relationships used to estimate heat transfer and pressure drop through tube banks are based on correlated results from experimental data, of which some of the earliest was presented by Shack [12]. Grimson [22] gave results which are widely used for calculating heat transfer and pressure drop through in-line and staggered tube bank arrangements. Experimental results for compact heat exchangers were published by Kays and London [7], and included a variety of heat exchanger geometries, such as finned and noncircular tubes.

Based on his recent experimental work, Zukauskas [16] presented results for flow through tube banks for a wide range of Reynolds numbers, including those Reynolds numbers less than 1000. These results are ideal for natural draft flue gas flows, whereas most of the previous results were restricted to highly turbulent flow, such as in typical industrial applications.

1.4 Defining the Heat Reclaimer Domain

Since there are many different varieties of residential combustion heating systems, only typical systems are considered. In general, the heating system is assumed to incorporate natural gas and oil-fired heating appliances that vary in input and steady-state efficiency. The heating appliance is connected to a masonry chimney by a flue pipe, and flue gases are drawn by natural draft and influenced by a draft regulator, which is fixed to the system flue pipe or heating appliance. It is further assumed that the horizontal flue pipe is long enough to attach the reclaimer, and that the heat reclaimer will be positioned on the chimney side of the draft regulator. Other heating system variables include flue-pipe diameter and length, chimney hydraulic diameter and height, and the ambient temperature.

The heat reclaimer designs considered a variety of materials for the best heat transfer qualities and the least materials and manufacturing costs. Variables such as tube diameter, tube spacing, tube arrangement (in-line and staggered), number of tube rows, number of tubes per row, and the reclaimer casing dimensions, were also considered. It is assumed that the reclaimer is equipped with an air blower which delivers 160 cfm for the tube bank, and is activated by a thermostat located in the device.

1.5 Computer Analysis and Optimization

The bulk of the analysis and designs of the heat reclaimer is done using the computer, and chapter V describes this in detail. The initial steps in the analysis of the tube bank heat reclaimer involve establishing the combustion process of natural gas and heating oil. By forming the respective balanced combustion equations, important flue gas information regarding the furnace temperature, dew-point temperature, and various heat transfer properties are evaluated. The computer program utilizes correlated results in conjunction with general energy equations for heat transfer and pressure drop. The computer results for safe reclaimer designs are evaluated for heat recovery, and the optimum designs are chosen. Reclaimers are then grouped together for versatility for use in heating systems.

The performance of the optimized reclaimer designs are examined under variable furnace efficiencies and ambient temperatures, and in terms of the reduction in home heating load.

CHAPTER II

WASTE HEAT RECOVERY

Waste heat recovery is important in all industrial, commercial, and residential heating processes. As fossil-fuel resources diminish, technology utilizing waste heat improves.

2.1 Industrial Waste Heat Recovery

Industrial sources of waste heat are usually divided into the three common categories: high, medium, and low temperatures. They are divided in this way because of the many diversified heat sources found in industry.

In the high temperature range, sources of heat measuring between $1200^{\circ}F$ and $3000^{\circ}F$ are common [7]. Most of the sources in this category are from direct fuel-fired processes in furnaces and incinerators. In the metals industry, where waste heat temperatures are well above $1000^{\circ}F$, special considerations must include the choice of materials for handling such high temperatures. Combustor flame temperatures of $3000^{\circ}F$ are considered the upper limit for flue gas temperatures.

Medium temperature heat recovery occurs from $450^{\circ}F$ to $1200^{\circ}F$, and is commonly found in exhaust from directly-fired process-equipment, such as steam boilers, gas turbines, and reciprocating engines [7]. Other sources of waste heat occur in processes such as in heat-treating furnaces, and drying ovens.

Low temperature heat recovery (below $450^{\circ}F$) is usually found in liquids or coolants that have been used in conjunction with other heat transfer processes. Some typical sources include process steam condensate, and coolants used in furnace doors, bearings, pumps, and condensers [7].

There is little dispute regarding available sources of waste heat found currently in industry. Process efficiency can be sufficiently improved by utilizing waste heat (see Fig. 2.1). Many methods and devices have been developed for industrial waste heat recovery.

In general, there are two methods of collecting and transfering industrial waste heat, direct and indirect. Most direct applications involve the mixing of high and low temperature fluids, such as condensing flue gas water vapor by direct contact between the flue gas and the cooling medium [26]. Although this method collects both sensible and latent heat, and helps reduce pollution, the corrosive products may be difficult to handle.

Indirect contact is the most popular method of utilizing waste heat. The difference is that the indirect method utilizes an intermediate fluid to relay heat between heat exchangers, whereas the direct method does not. Corrosion may also be a problem in the indirect method, if flue gases are condensed over a heat exchanging medium. It is important, regardless of how waste heat is collected, that the proper materials be chosen for the heat exchangers.

Preheating is one of the most popular applications of medium-to-high-temperature waste heat. One example of preheating is that of combustion air. A primary air temperature increase of $80^{\circ}F$ translates into a 1% increase in combustion efficiency [13]. Another example of preheating involves that of boiler feedwater, which reduces the amount of energy needed to raise steam. The final example involves the necessity to preheat liquids or feedstocks for processing, or further development.

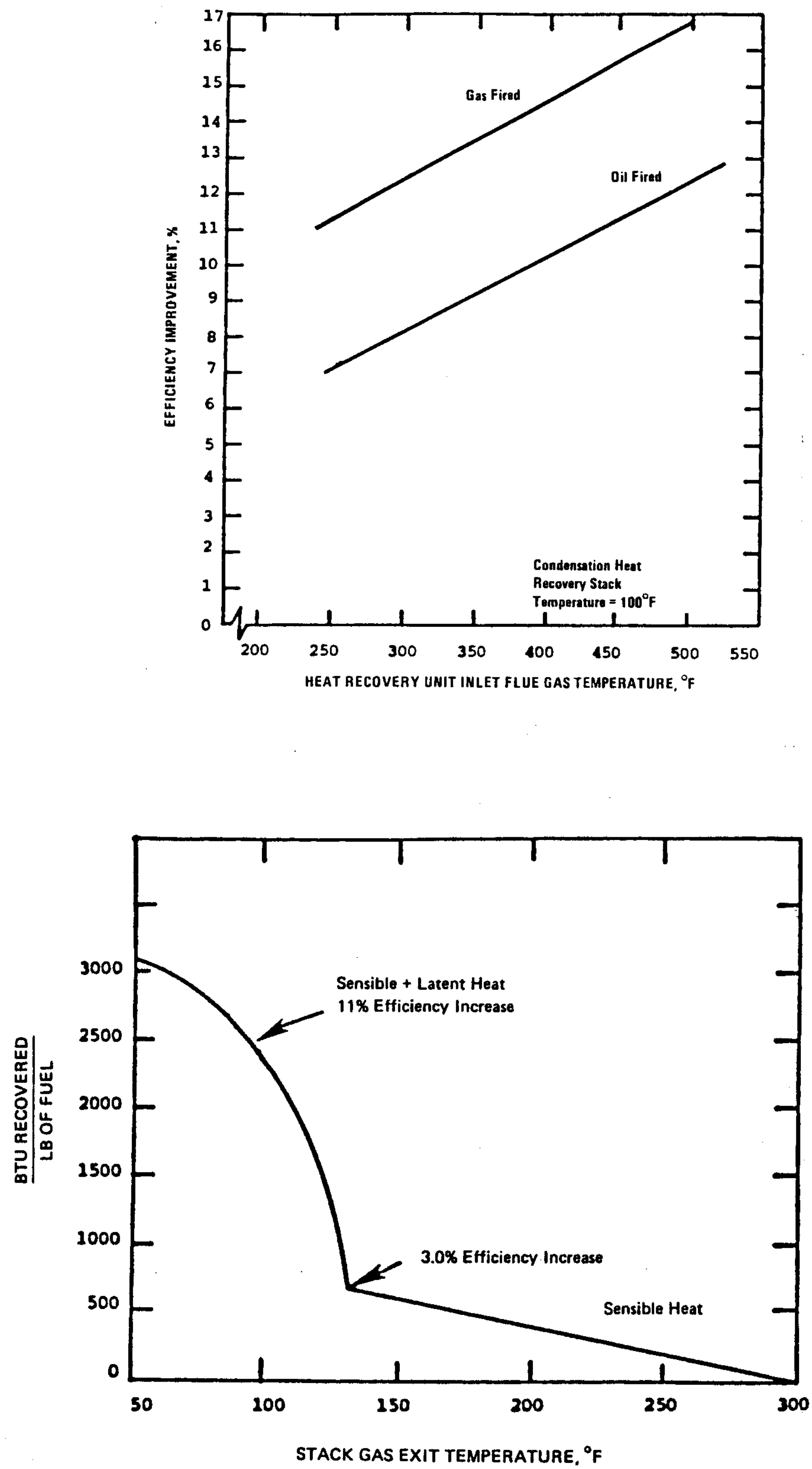


Fig. 2.1 Industrial Waste Heat Recovery (7)

Metals are often preheated for forming and rolling, and some liquids need preheating for chemical processes.

Another application of waste heat, particularly for medium temperature sources, is for a waste heat boiler. A waste heat boiler utilizes flue gas, or some other process gas at a high temperature, to raise steam for heating, processing, or power generation. Some waste heat boilers have been built to handle from 1,000 to 1,000,000 cfm of exhaust gas [7]. Typical sources for waste heat boilers are gas turbines and incinerators.

In a related topic, special gas and vapor expanders have been developed that utilize low pressure waste gases. Recovery of as much as 20 to 30% can be achieved by a waste-gas turbine [7].

Most applications of low temperature waste heat are in the area of absorption cooling for refrigeration and air conditioning equipment. Perhaps the best example of low temperature recovery is through an industrial heat pump. Since the available heat is at such a low temperature, industries often discourage waste heat sources below 300 $^{\circ}$F, so that flue gas condensation can be avoided [7].

The core of industrial waste heat recovery is in the heat exchanger technology. The first class of industrial heat recovery heat exchangers (HRHX) is the recuperator. A recuperator is defined as a system of walled ducts used to exchange heat between gases. The two basic types of recuperators are briefly discussed.

The first type is a metallic radiation recuperator, which is composed of concentric pipes or ducts (see Fig. 2.2a). Typical applications of this type of recuperator are for primary air preheating. Metallic radiation recuperators used with extremely hot flue gases often have the inner duct constructed of stainless steel, and operate in

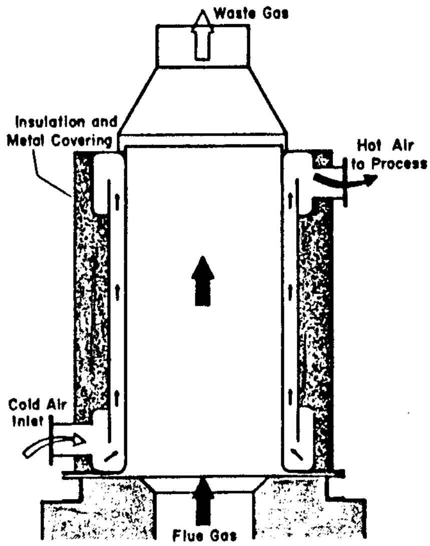


Fig. 2.2a Metallic Radiation Recuperator

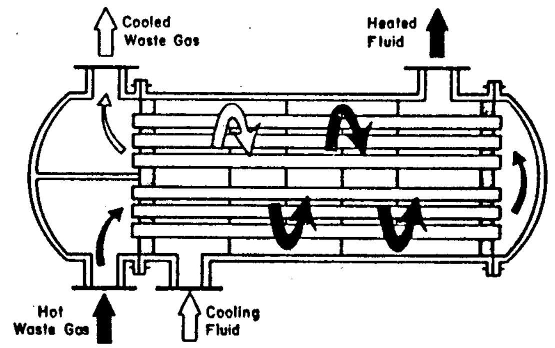


Fig. 2.2b Shell-and-Tube Recuperator

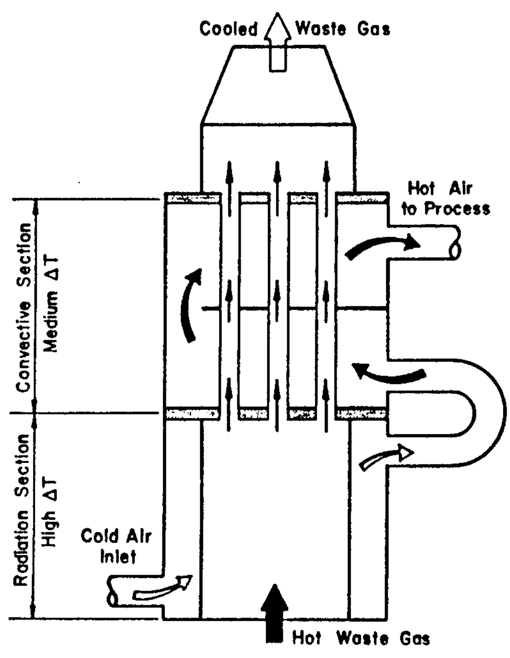


Fig. 2.2c Combination Recuperator

Fig. 2.2 Industrial Heat Recovery Heat Exchangers (7)

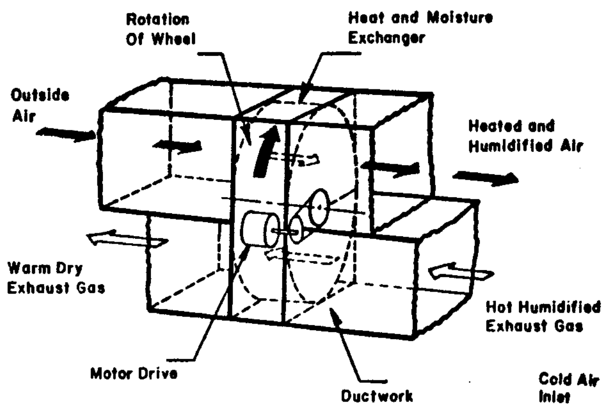


Fig. 2.2d Heat Wheel

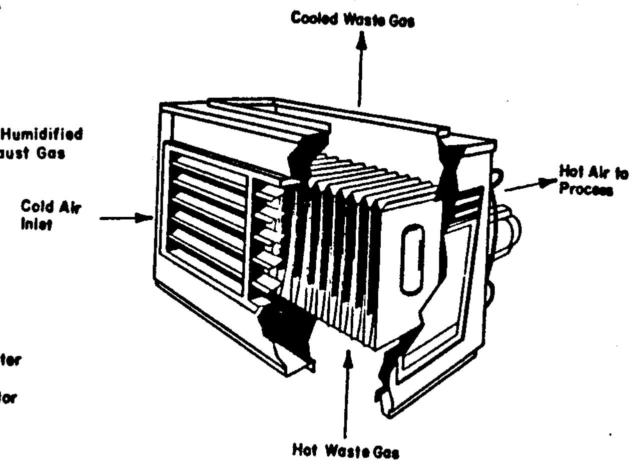


Fig. 2.2e Gas-to-Gas Regenerator

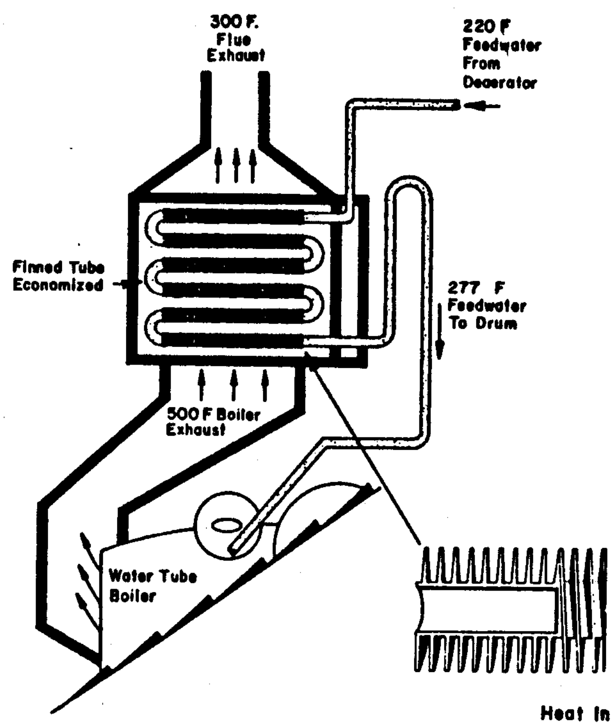


Fig. 2.2f Economizer

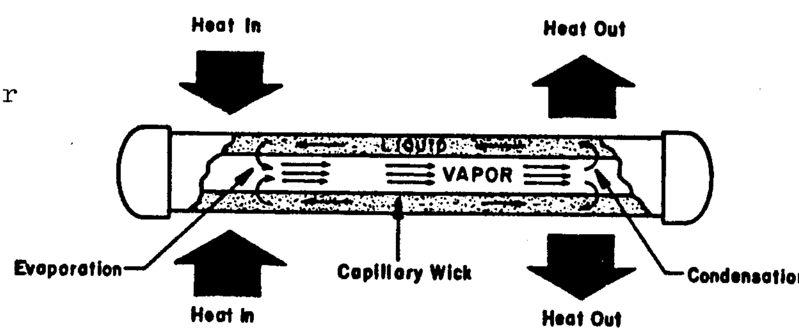


Fig. 2.2g Heat Pipe

Fig.2.2 continued Industrial Heat Recovery Heat Exchangers (7) parallel flow (as opposed to more effective counter flow) to reduce undesirable thermal stresses. Another common recuperator is the convective-type, which usually operates by passing process-air over tube banks (or ducts) containing flue gases. The most common variety is the shell-and-tube recuperator shown in Fig. 2.2b, using single or multiple passes. Some advantages over radiation recuperators are an increased effectiveness and relative size.

Other, less common, types of recuperators used in industry include ceramic, tube-within-tube, and combination recuperators. Ceramic recuperators have the advantage of handling extremely high temperatures ($2800^{o}F$), but tend to crack and leak at the joints [7]. Current technology, using short, carbide tubes and flexible seals has reduced leakage to only a few percent. Tube-within-tube recuperators poke into flue gas areas, sending cold air through the inner tube and out the annulus, and have the advantage of easy replacement [4]. Combination (radiation and convective-type) recuperators, shown in Fig. 2.2c, produce a higher effectiveness than comparable radiation types, but are more expensive.

The cost of introducing recuperators in existing systems, replacing or overhauling existing recuperators, or incorporating recuperators into new construction, is often weighed carefully against heat recovery.

Another class of HRHX is a regenerator, which is defined as a heat exchanger that transfers mid-to-low temperature heat into some medium that can store it for a later use. The first type is the switched regenerator, which consists of two or more refractory chambers where both hot gases and cold fluids are alternately distributed throughout [4]. As one chamber is being charged by flue gases, the other chamber (previously charged) is preheating some other fluid, such as primary air or boiler feedwater. It is through the process of storing heat energy in refractory, to be later used, that the system operates. Unfortunately, switched regenerators have a tendency to leak, take up large areas, are expensive, and require sophisticated controlling.

Another type of regenerator is the heat wheel, which is fabricated from a porous material and transfers heat by rotating between two adjacent ducts, as shown in Fig. 2.2d. Heat wheels have the advantage of transfering both sensible and latent heat, so that maximum efficiencies can reach above 85%.

There are four main types of heat wheels, which are constructed either in a disk or drum configuration [7]. The first is packed with an aluminum or steel-knitted mesh, which must be corrosion-resistant for latent transfer. The last three varieties, called laminar wheels, are constructed of channels that store heat and circulate fluid. The first laminar-type consists of corrugated metal channels, whereas the second consists of a ceramic matrix honeycomb, which can handle higher temperatures. The final laminar type is coated with a dessicant to resist corrosive condensation properties inherent in flue gases.

In many cases heat wheels must be equipped with a purge section to prevent cross-contamination between fluids. A small section of the heat wheel flushes out flue gases with fresh air, and common practice consists of 6 air-changes for purging [7]. Purging also keeps the heat exchanger surfaces clean, but contributes to additional costs associated with extra equipment, and adds to the complexity of the system.

The final type of regenerator is the passive gas-to-gas heat exchanger, which is used strictly for air preheating (see Fig. 2.2e). Advantages of this type of regenerator are that cross contamination does not exist, and there are no moving parts to break down. Disadvantages include difficult temperature control, expense of construction, and bulkiness [7].

Waste heat recovery equipment is often labeled by various industries, causing confusion about correct names associated with heat recovery processes. For example, some heat exchangers that are referred to as regenerators, do not follow the definition outlined previously. Some of these heat exchangers are outlined here.

The first such heat exchanger is the bare and finned-tube economizer, shown in Fig. 2.2f. Economizers are usually associated with heating boiler feedwater, using low-to-medium temperature waste heat. Since economizers often operate in highly corrosive environments, tubes are bundled together in modular sections that can be easily replaced. Deep economizers, using finned tubes, have been developed which can handle flue gas below 270^{o}F [7]. Materials considered for such economizers include carbon tubes (used as inexpensive, modular throw-away sections), stainless steel tubes, combination stainless and carbon tubes, glass tubes, and teflon tubes. Typical deep economizer gas outlet temperatures can reach below 160^{o}F, whereas boiler feedwater can reach above 130^{o}F.

Shell-and-tube heat exchangers are mentioned here again, because of their versatile use. Besides being a recuperator for preheating primary air, shell-and-tube heat exchangers have become a standard in many types of fluid heat transfer. Some examples include heating boiler feedwater, and other process liquids from the condensates contained in air conditioning systems, coolants, lubricants, and distillation processes [7].

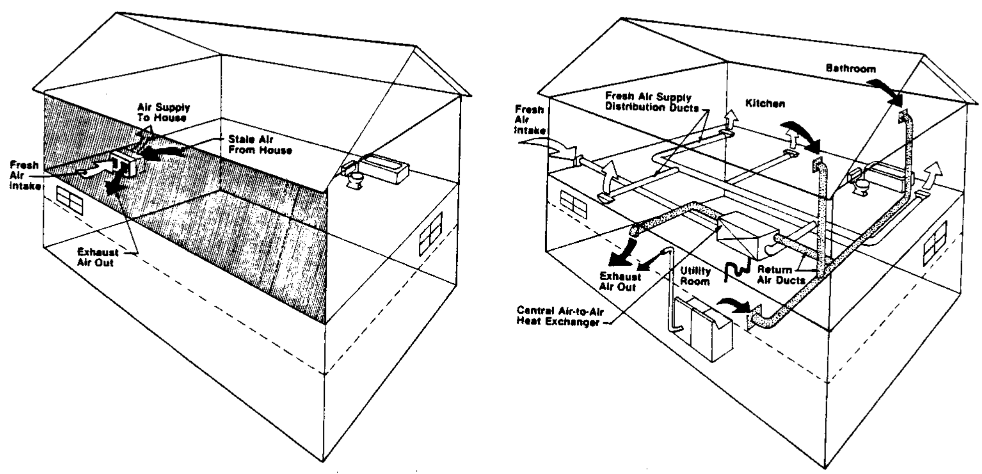
The heat pipe heat exchanger, shown in Fig. 2.2g, is relatively new and primarily used in research and development situations. The heat pipe is a self-contained heat exchanger that operates on the principles of condensation and evaporation, coupled with capillary circulation, using an enclosed refrigerant. As a finned-tube-bank heat exchanger, the heat pipe makes a very compact and effective heat recovery device. Unfortunately, it is too expensive for most practical uses.

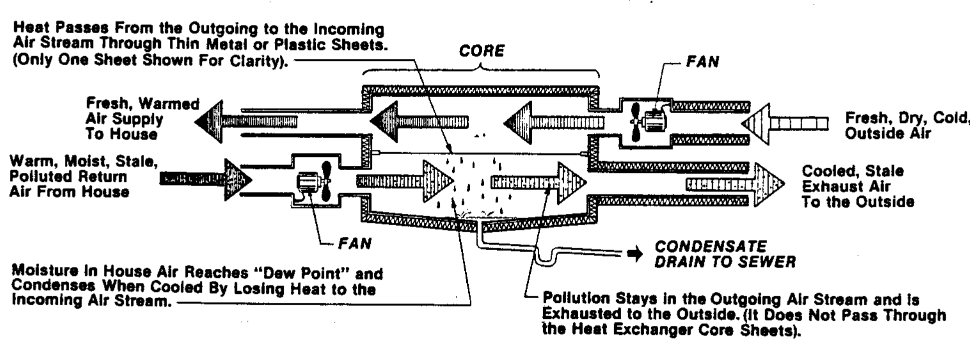
Utilizing waste heat is the correct path for a more efficient industry. The primary source of waste heat is found in combustion equipment, and the primary use of waste heat is for primary air, boiler feedwater, heating process fluids, and power generation. Many of the ideas for industrial waste heat recovery can be easily transfered to commercial and residential heat recovery.

2.2 Residential Heat Recovery

Residential heat recovery evolved rapidly during the energy crisis of the mid-to-late 1970's. Many methods and devices were developed that recovered waste heat from the house and the environment. Perhaps the most popular method of home conservation, besides turning down the thermostat, was to replace old insulation and weather stripping. Older homes typically have deteriorated insulation and window caulking, which allows heat to escape readily. Improved insulation materials and federal energy tax incentives helped support such energy conservation measures. Sealing a house against heat loss, however, has adverse effects concerning fresh air circulation and house air pressure.

An air-to-air heat exchanger is a heat recovery device which heats incoming fresh air with previously heated exhaust (stale) air. Application and a section of a typical residential air-to-air heat exchanger is shown in Fig. 2.3.





Air-to-Air Heat Exchanger

Fig. 2.3 Residential Air-to-Air Heat Recovery (23)

There are several different air-to-air heat exchanger core types available [23]. Among the most popular are the flat plate-types, which operate on counter, parallel, and cross flows. Similarly, there are concentric tube-types for counter and parallel flows. Also, there are heat wheel-types that recover both sensible and latent heat, and finally, heat pipe-types.

In general, air-to air heat exchangers must deal with moisture. For such low-level heat recovery, where warm house air at $65^{\circ}F$ passes by cold winter air, condensation of the warm air is likely to occur, causing corrosion of the heat exchanger core. Some models are equipped with automatic defrosting controls, which essentially preheats the cold air before air-to-air heat exchange occurs. Fan noise may also be a problem if the heat exchanger is improperly mounted. Finally, depending on how tight the house is sealed and at what rate the air is flowing into the house, there could exist a pressure loss or gain that may cause discomfort or heat loss.

The outside environment also provides waste heat sources for residential energy use. One example, is the outside air as a heat source for heat pumps. The heat pump operates on a refrigeration cycle, which can be reversed for air conditioning. In the heating mode, the refrigerant gives off heat in the condenser and absorbs heat from outside air. Many new homes in North America currently use electric heat pumps as a main or back-up source of heat. However, the heat pump life expectancy and output have not been substantially developed for its practical use in colder climates.

Solar energy can also be a good source of additional home heat. Solar heat gain depends on what type of system is used, but more importantly, how clear the sky is. The two fundamental solar energy systems are passive and active. An example of a passive solar energy system, is directing sun rays onto heat absorbing objects located inside the house (thermal mass), which will give off heat to the indoor environment. Glass and curtains must be carefully selected to prevent major heat dissipation at night.

An active solar system is usually more complicated and expensive to operate. Operation consists of circulating a working fluid through an exposed piping system, which encompasses a large heat transfer area. The fluid is then returned indoors for residential applications, such as hot water heating. Solar collector components, however, have a tendency to corrode since they are often exposed to outside climate conditions.

Another environmental source of waste heat energy is thermal ground storage. Below the ground frost level, the earth is roughly at a constant temperature of between 50 and 60 ^{o}F. Rock-bed storage areas can be developed which will aid in heating the house long after the furnace cycles off. Ground waste heat can also be coupled with heat pumps and active solar systems, providing an excellent environment for supplying and storing heat.

A large source of residential waste heat occurs in combustion appliances. Several different devices have been developed that reclaim or prevent waste heat from escaping out the chimney. The first type of heat recovery device is the flue damper, which originated in Europe in the early 1930's [27]. Operation of the flue damper (see Fig. 2.4a) is rather simple in that when the furnace is on the damper is open, and when the furnace shuts off the damper is closed. Much of the success of the flue damper hinges on the heating system that it is attached to, and the weather conditions. For example, many combustion heating systems continue to blow air over the furnace heat exchanger long after the

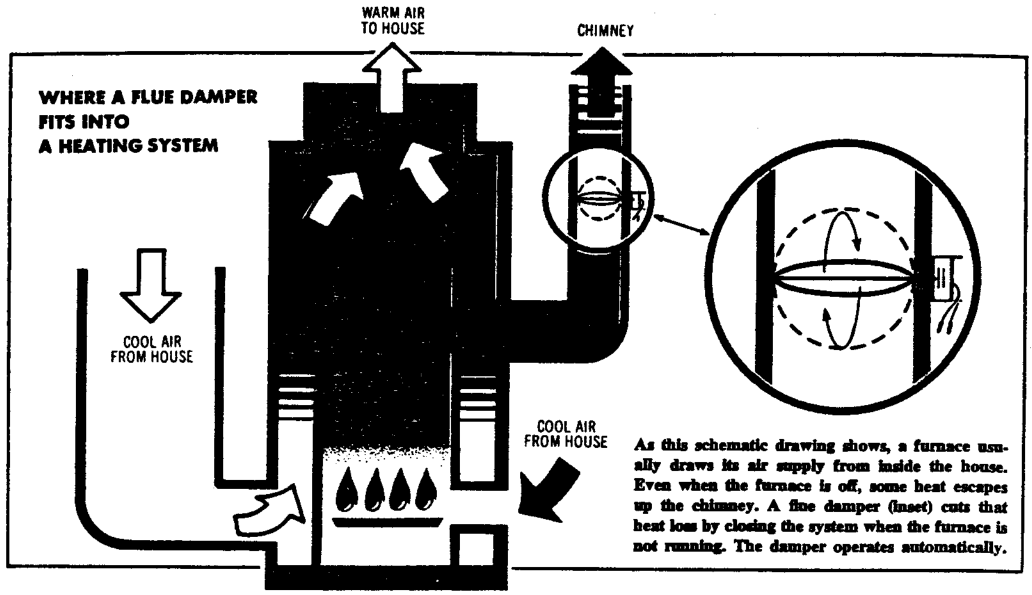


Fig. 2.4a Residential Flue Dampers (27)

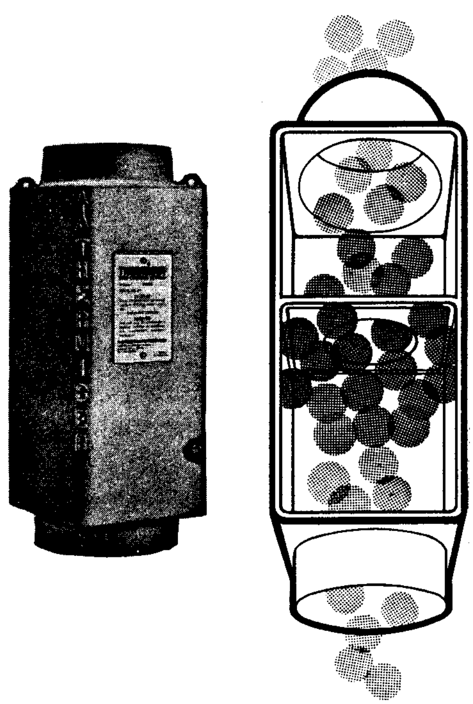


Fig. 2.4b Flue Gas Choker
(Thermiser)

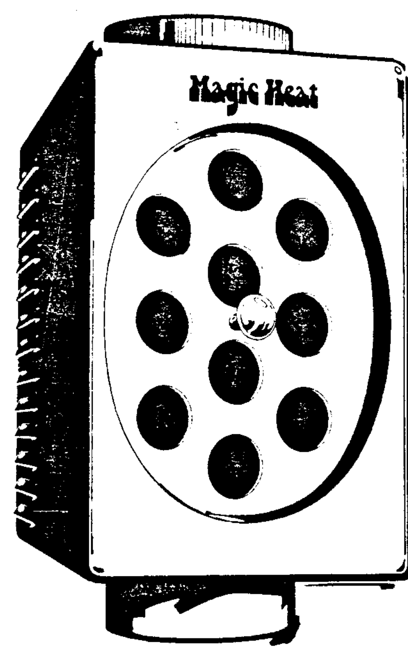


Fig. 2.4c Residential Tube
Bank Heat Reclaimer

Fig. 2.4 Residential Flue Gas Heat Recovery Devices combustion process is over, so that the flue damper does not prevent significant heat loss. In older heating equipment, where heated air is assisted by gravity only, flue dampers can save 10 to 30% in fuel cost, depending on the combustion system.

Flue dampers must incorporate special design features which will prevent flue gas leakage in the case of device failure. One such feature is that the furnace is prevented from being turned on, unless the damper is fully open. Another feature is that the damper is mechanically controlled to open in case of a power outage or an electrical device failure.

Another type of residential heat recovery device is the flue gas choker, or "Thermiser" (brand name, see Fig. 2.4b). The device operates on the principle that most combustion processes are incomplete, and significant amounts of unburned fuel escape out the chimney. The device chokes the flow of flue gases so that fuel is burned more completely in the combustion chamber. In addition, the device helps prevent flue gas heat from escaping when the furnace cycles off, and the device has no moving parts.

A less expensive approach to heat recovery is flue pipe heat fins. Heat fins are corrugated aluminum strips that wrap around the flue pipe, and improve the flue pipe convective and radiative heat transfer to the surrounding air. Although there is no flue gas blockage to be concerned with, the heat fins do not reclaim much heat, and are justified only as an inexpensive method of heat recovery [19].

Tube bank heat reclaimers are the most versatile residential heat recovery devices available [19]. The two basic types are water and air-heated. Water-heated tube bank heat reclaimers can be used for a variety of applications, including preheating for boiler feedwater and domestic hot water. Air-heated tube bank heat reclaimers (see Fig. 2.4c) can be used as a combustion air preheater, a ventilation air preheater, or a space heater. The device, however, is limited to improving a system efficiency to only 84%, due to unwanted condensation of the latent heat contained in the flue gases.

In general, residential energy conservation depends significantly on safety, cost of improved efficiency, and the amount of waste heat recovered. Conservation technology changes with fuel costs, and as the concern for waste heat recovery increases, more effective heat exchangers will develop.

CHAPTER III

RESIDENTIAL COMBUSTION HEATING SYSTEMS

Residential combustion heating systems expel a significant amount of flue gas heat into the atmosphere. In most cases, at least 20% of the heat produced by the combustion furnace is lost to the environment.

In this section, basic relationships for the combustion of natural gas and No.2 heating oil will be emphasized, so that various flue gas properties and system temperatures can be predicted. Also, residential combustion heating systems will be analyzed for available chimney draft and heat transfer.

Combustion of residential fuels, such as natural gas and No. 2 heating oil, have certain fundamental similarities. When burned with air, both fuels produce similar flue gas constituents, such as water vapor, carbon dioxide, nitrogen, and oxygen, which simplifies the analysis of the flue gases. Other products of combustion, such as carbon monoxide and carbon, if any, depend on how complete the combustion process is, and in general, are not included in the following combustion analysis.

3.1 The Combustion Process

Natural gas, composed primarily of methane, is a clean burning fuel that is distributed virtually everywhere in the United States (see Table 3.1). Natural gas contains 73 to 95% methane (by volume), depending on regional sources. Other trace substances typically found in natural gas include ethane and other hydrocarbon fuels, carbon dioxide,

Table 3.1 Natural Gas in the United States (2)

No.	City	Components of Gas, Percent by Volume: Methane	Ethane	Propane	Butanes	Pentanes	Hexanes plus	CO_2	N_2	Misc.	Heat Value,† Btu/Cu Ft	Sp Gr
1	Abilene, Tex.	73.52	13.23	4.35	0.56	0.06	0.11	0.16	8.01	—	1121	0.710
2	Akron, Ohio	93.30	3.49	0.69	.18	.04	.00	0.50	1.80	—	1037	.600
3	Albuquerque, N.M.	86.10	9.49	2.34	.44	.08	.03	1.02	0.50	—	1120	.646
4	Atlanta, Ga.	93.42	2.80	0.65	.33	.12	.10	1.38	1.20	—	1031	.604
5	Baltimore, Md.	94.40	3.40	0.60	.50	.00	.00	0.60	0.50	—	1051	.590
6	Birmingham, Ala.	93.14	2.50	0.67	.32	.12	.05	1.06	2.14	—	1024	.599
7	Boston, Mass.	93.51	3.82	0.93	0.28	0.07	0.06	0.94	0.39	—	1057	0.604
8	Brooklyn, N.Y.	94.52	3.29	0.73	.26	.10	.09	0.70	0.31	—	1049	.595
9	Butte, Mont.	87.38	3.02	1.09	.11	.06	.00	1.98	6.36	—	1000	.610
10	Canton, Ohio	93.30	3.49	0.69	.18	.04	.00	0.50	1.80	—	1037	.600
11	Cheyenne, Wyo.	91.00	4.73	1.20	.30	.06	.04	1.86	0.81	—	1060	.610
12	Cincinnati, Ohio	94.25	3.98	0.57	.16	.03	.03	0.68	0.30	—	1031	.591
13	Cleveland, Ohio	93.30	3.49	0.69	0.18	0.04	0.00	0.50	1.80	—	1037	0.600
14	Columbus, Ohio	93.54	3.58	0.66	.22	.06	.03	.85	1.11	—	1028	.597
15	Dallas, Tex.	86.30	7.25	2.78	.48	.07	.02	.63	2.47	—	1093	.641
16	Denver, Colo.	81.11	6.01	2.10	.57	.17	.03	.42	9.19	—	1011	.659
17	Des Moines, Iowa	80.38	6.39	2.46	.61	.08	.03	.20	9.53	0.32 He	1012	.669
18	Detroit, Mich.	89.92	4.21	1.34	.34	.09	.01	.59	3.30	0.20 He	1016	.616
19	El Paso, Tex.	86.92	7.95	2.16	0.16	0.00	0.00	0.04	2.72	0.05 He	1082	0.630
20	Ft. Worth, Tex.	85.27	8.43	2.98	.62	0.09	0.04	0.27	2.30	—	1115	.649
21	Houston, Tex.‡	92.50	4.80	2.00	.30	—	—	0.27	0.13	—	1031	.623
22	Kansas City, Mo.	72.79	6.42	2.91	.50	0.06	Trace	0.22	17.10	—	945	.695
23	Little Rock, Ark.	94.00	3.00	0.50	.20	0.20	—	1.00	1.10	—	1035	.590
24	Los Angeles, Calif.	86.50	8.00	1.90	.30	0.10	0.10	0.50	2.60	—	1084	.638
25	Louisville, Ky.	94.05	3.41	0.40	0.13	0.05	0.09	1.20	0.67	—	1034	0.596
26	Memphis, Tenn.	92.50	4.37	0.62	.18	.07	.10	1.60	0.56	—	1044	.608
27	Milwaukee, Wis.	89.01	5.19	1.89	.66	.44	.02	0.00	2.73	0.06 He	1051	.627
28	New Orleans, La.	93.75	3.16	1.36	.65	.66	.00	.42	0.00	—	1072	.612
29	New York City	94.52	3.29	0.73	.26	.10	.09	.70	0.31	—	1049	.595
30	Oklahoma City, Okla.	89.57	6.31	1.36	.36	.00	.00	.13	2.06	0.21 O_2	1080	.615
31	Omaha, Neb.	80.46	6.30	2.59	0.68	0.09	0.05	0.17	9.32	0.34 He	1020	0.669
32	Parkersburg, W. Va.	94.50	3.39	0.68	.12	.07	.03	0.67	0.41	0.01 O_2	1049	.592
33	Phoenix, Ariz.	87.37	8.11	2.26	.13	.00	.00	0.61	1.37	—	1071	.633
34	Pittsburgh, Pa.	94.03	3.58	0.79	.28	.07	.04	0.80	0.40	0.01 O_2	1051	.595
35	Providence, R.I.	93.05	4.01	1.02	.34	.08	.08	1.00	0.42	—	1057	.601
36	Provo, Utah	91.40	3.95	0.84	.39	.03	.01	0.52	2.86	—	1032	.605
37	Pueblo, Colo.	73.86	5.71	3.20	1.34	0.14	0.06	0.13	15.26	—	980	0.706
38	Rapid City, S.D.	90.60	7.20	0.82	0.19	.03	.03	.18	0.93	0.02 He	1077	.607
39	St. Louis, Mo.	93.32	4.17	0.69	.19	.05	—	.98	0.61	—	—	—
40	Salt Lake City, Utah	91.17	5.29	1.69	.55	.16	.03	.29	0.82	—	1082	.614
41	San Diego, Calif.	86.85	8.37	1.86	.15	.00	.00	.41	2.32	0.04 He	1079	.643
42	San Francisco, Calif.	88.69	7.01	1.93	.28	.03	.00	.62	1.43	0.01 He	1086	.624
43	Toledo, Ohio	93.54	3.58	0.66	0.22	0.06	0.03	0.85	1.11	—	1028	0.597
44	Tulsa, Okla.	86.29	8.36	1.45	0.18	.14	.01	0.23	2.95	0.39 O_2	1086	.630
45	Waco, Tex.	93.48	2.57	0.89	0.43	.17	.11	1.69	0.66	—	1042	.607
46	Washington, D.C.	95.15	2.84	0.63	0.24	.05	.05	0.62	0.42	—	1042	.586
47	Wichita, Kan.	79.62	6.40	1.42	1.12	.48	.14	0.10	10.62	0.10 O_2	1051	.660
48	Youngstown, Ohio	93.30	3.49	0.69	0.18	.04	.00	0.50	1.80	—	1037	0.600

* Average analyses obtained from the operating utility company(s) supplying the city; the gas supply may vary considerably from these data—especially where more than one pipeline supplies the city. Also, as new supplies may be received from other sources, the analyses may change. Peak shaving (if used) is not accounted for in these data.

† Gross or higher heating value at 30 in. Hg, 60 F, dry. To convert to a saturated basis deduct 1.73 percent; i.e., 17.3 from 1000, 19 from 1100.

‡ 1954 data.

nitrogen, and other miscellaneous substances.

Typical reactions for common fuels are as follows:

$$\text{Methane} \quad CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O \tag{3.1}$$

$$\text{Ethane} \quad C_2H_6 + 3.5O_2 \rightarrow 2CO_2 + 3H_2O \tag{3.2}$$

$$\text{Propane} \quad C_3H_8 + 5O_2 \rightarrow 3CO_2 + 4H_2O \tag{3.3}$$

$$\text{Butane} \quad C_4H_{10} + 6.5O_2 \rightarrow 4CO_2 + 5H_2O \tag{3.4}$$

The stoichiometric balanced reaction for the combustion of natural gas fuel (No. 48 fuel shown on Table 3.1) is

$$\underbrace{0.933CH_4 + 0.035C_2H_6 + 0.007C_3H_8 + 0.002C_4H_{10} + 0.018N_2 + 0.005CO_2}_{\text{Natural Gas Fuel}}$$

$$+ \underbrace{2.035(O_2 + 3.76N_2)}_{\text{Theoretical Air}} \rightarrow \underbrace{2.01H_2O + 1.036CO_2 + 7.67N_2}_{\text{Combustion Products}} \tag{3.5}$$

Excess air is usually required to assure complete combustion of fuel constituents, because unburned products lower the fuel heating value and the combustion efficiency [14]. Excess air can be expressed in terms of the carbon dioxide per cent in the flue gas products. For natural gas used in residential heating systems, the suggested value is 5.3% CO_2 (volume basis) [3]. The balanced combustion now becomes

$$0.933CH_4 + 0.035C_2H_6 + 0.007C_3H_8 + 0.002C_4H_{10} + 0.018N_2 + 0.005CO_2$$
$$+ 3.89(O_2 + 3.76N_2) \rightarrow 2.01H_2O + 1.04CO_2 + 14.65N_2 + 1.86O_2 \tag{3.6}$$

On a volumetric basis, the per cent excess air can readily be estimated by the following equation:

$$\%\ \text{Excess Air} = (\text{Actual Air} - \text{Theoretical Air})/\text{Theoretical Air} \tag{3.7}$$

For natural gas, the amount of excess air required for complete combustion equals 91.4%. This can also be regarded as 191.4% of the theoretical air (air used for combustion is assumed to contain 79% nitrogen and 21% oxygen).

The products of combustion (flue gas constituents) give important information regarding the combustion adiabatic flame temperature. Assuming that the combustion process involves no external work, or heat transfer, the adiabatic flame temperature can be estimated from the first law of thermodynamics [14]. For a steady-state, steady-flow process,

$$H_R = H_P, \tag{3.8}$$

where H_R is the total enthalpy of the reactants, and H_P is the total enthalpy of the products. Treating the reactants and products as mixtures, equation (3.8) becomes

$$n_R[\bar{h}_f + \Delta h]_R = n_P[\bar{h}_f + \Delta h]_P, \tag{3.9}$$

where n_R is the number of moles of a reactant constituent, n_P the number of moles of a product constituent, $\bar{h}_f$ the enthalpy of formation, and Δh the change of enthalpy, from standard conditions. Based on equation (3.6), the total enthalpy for the reactants becomes

$$\begin{aligned} H_R = {} & 0.933[\bar{h}_f + \Delta h](CH_4) + 0.035[\bar{h}_f + \Delta h](C_2H_6) + 0.007[\bar{h}_f + \Delta h](C_3H_8) \\ & + 0.002[\bar{h}_f + \Delta h](C_4H_{10}) + 0.018[\bar{h}_f + \Delta h](N_2) + 0.005[\bar{h}_f + \Delta h](CO_2) \\ & + 3.89[\bar{h}_f + \Delta h](O_2) + 3.89(3.76)[\bar{h}_f + \Delta h](N_2) \end{aligned} \tag{3.10}$$

Assuming that the reactants are at standard conditions ($77^{o}F$ and 1 atm), the terms $\bar{h}_f$ and Δh for oxygen and nitrogen in equation (3.10) vanish.

A similar equation for the products is

$$H_P = 2.01[\bar{h}_f + \Delta h](H_2O) + 1.04[\bar{h}_f + \Delta h](CO_2) + 14.65[\bar{h}_f + \Delta h](N_2) + 1.86[\bar{h}_f + \Delta h](O_2) \quad (3.11)$$

The enthalpy of formation and the change of enthalpy for various flue gas constituents are shown in Appendix A. The enthalpy of formation of some common fuels, at standard conditions, can be readily obtained [14]. The procedure for solving for the adiabatic flame temperature involves a trial and error solution. For natural gas, the adiabatic flame temperature (T_{fl}) is estimated to be 2287 ^oF. T_{fl} is the maximum temperature, and dissociation and other combustion inefficiencies will produce a lower actual flame temperature [14]. However, it remains the initial temperature in the furnace, and any irreversibilities are recognized by a furnace steady-state efficiency introduced later in the analysis.

Another important flue gas characteristic is its dew-point. Since the water vapor contained in the flue gas is only a small portion of the total flue gas make-up, the dew-point temperature must be related to the water vapor partial-pressure in the flue gas. Assuming that the flue gas is at atmospheric pressure, the partial-pressure of the water vapor is

$$P_{H_2O} = (\text{Moles of Water Vapor / Moles of Flue Gas})\ B, \quad (3.12)$$

where B refers to the atmospheric pressure (assumed at 1000 feet elevation). In the case of natural gas, the partial pressure was estimated to be 1.465 psia, which corresponds to a flue gas dew-point temperature of approximately 115 ^oF.

Reclaiming heat from flue gases that are below the dew-point temperature adds considerably to the heat recovery, but condensation of flue gases may create undesirable system corrosion.

In addition to the adiabatic flame and dew-point temperatures, the heat transfer properties of the flue gases can be evaluated. Flue gas properties used in this thesis include density, constant pressure specific heat, conductivity, and absolute viscosity. There is a negligible difference between the densities of residential flue gas and air, at the same temperatures [3]. For the other heat transfer properties, a composite analysis of the flue gas constituents is performed. The break-down of the flue gas is as follows:

TABLE 3.2

FLUE GAS CONSTITUENTS FOR COMBUSTION OF NATURAL GAS

Mixture component	% by volume	Mole fraction y_i	Molecular weight M_i	Mass per mole of mixture	mass fraction m_i
N_2	74.9	0.749	28	20.97	0.744
O_2	9.5	0.095	32	3.04	0.1078
H_2O	10.3	0.103	18	1.85	0.0656
CO_2	5.3	0.053	44	2.33	0.0826

For flue gas mixtures consisting of several constituents, an average constant pressure specific heat is estimated by

$$C_p = \Sigma C_{pi} m_i, \tag{3.13}$$

where C_{pi} is the constant pressure specific heat of one mole of a flue gas constituent [14], and m_{fi} is defined by

$$m_{fi} = m_i / M_i \quad \text{(for a constituent)} \tag{3.14}$$

For natural gas combustion products, the average specific heat is

$$C_p = 1.0857\text{x}10^{-5}\,\theta^2 + 1.6185\text{x}10^{-5}\,\theta^{1.5} - 0.005\theta + 0.0858\theta^{0.5} - 0.1599\theta^{0.25}$$
$$+ 0.4011 - 3.4044\theta^{-1.5} + 7.0031\theta^{-2} - 5.2101\theta^{-3}\ (\text{Btu/lbm-}^{\circ}\text{F}), \quad (3.15)$$

where

$$\theta = T(^{\circ}R)/180 \quad (3.16)$$

Other properties, such as conductivity k and absolute viscosity μ can be estimated in a similar manner [6]. The equations are

$$k = [\Sigma\, y_i k_i (M_i)^{1/3}]/[\Sigma y_i (M_i)^{1/3}], \quad (3.17)$$

and

$$\mu = [\Sigma\, y_i \mu_i (M_i)^{1/2}]/[\Sigma y_i (M_i)^{1/2}], \quad (3.18)$$

where y_i is the mole fraction, μ_i the absolute viscosity, k_i the conductivity, and M_i the molecular weight of a constituent. Constituent properties can be found in various heat transfer texts, and some are given in Appendix B. The individual properties were approximated as linear from 100 to 800°F, and equations for the properties are summarized below:

TABLE 3.3

CONSTITUENT RELATIONSHIPS FOR FLUE GAS T = (°F)

Constituent	μ_i (lbm/h-ft)	k_i (Btu/h-ft-°F)
N_2	$4.365\text{x}10^{-5}T + 0.0423$	$1.975\text{x}10^{-5}T + 0.0133$
O_2	$5.175\text{x}10^{-5}T + 0.0497$	$2.400\text{x}10^{-5}T + 0.0132$
H_2O	$5.125\text{x}10^{-5}T + 0.0202$	$3.025\text{x}10^{-5}T + 0.0079$
CO_2	$4.860\text{x}10^{-5}T + 0.0342$	$2.412\text{x}10^{-5}T + 0.0077$

Combining the relationships in Table 3.3 and Table 3.2, gives an average value for the flue gas conductivity and absolute viscosity, and are given in the following equations:

$$k = 2.1375\text{x}10^{-5}\ T + 0.012465 \quad (\text{Btu/h-ft-}^{\circ}\text{F}), \tag{3.19}$$

and

$$\mu = 4.5319\text{x}10^{-5}\ T + 0.040693 \quad (\text{lbm/h-ft}) \tag{3.20}$$

There are many similarities between natural gas and No.2 heating oil. Fuel oil, for residential heating purposes, is a heavy hydrocarbon fuel, which belongs to the distillate fuel oil family [1]. The carbon content of No.2 fuel oil is approximately 84 to 86%, which corresponds to a fuel structure of $C_{12}H_{26}$. Other trace substances exist in the fuel, but will be considered negligible and excluded from the combustion analysis.

The stoichiometric combustion of No.2 fuel oil in air is given by the following equation:

$$\underbrace{C_{12}H_{26}}_{\text{Fuel}} + \underbrace{18.5(O_2 + 3.76N_2)}_{\text{Theoretical Air}} \longrightarrow \underbrace{13H_2O + 12CO_2 + 69.56N_2}_{\text{Combustion Products}} \tag{3.21}$$

The amount of excess air required for complete combustion is expressed in terms of an accepted CO_2 content, which is 9% for heating oil [3]. The corresponding equation becomes

$$C_{12}H_{26} + 26.64(O_2 + 3.76N_2) \longrightarrow 13H_2O + 12CO_2 + 100.2N_2 + 8.14O_2 \tag{3.22}$$

The excess air (volume basis), based on 9% CO_2 in the combustion products, was calculated, using equation (3.7), to be 44%.

The adiabatic flame temperature for heating oil was calculated using equation (3.9), and was found to be 2918°F. The dew-point temperature, using equation (3.12), was 112.5°F.

Table 3.4 gives the properties of the flue gas constituents. The overall flue gas properties were calculated as for natural gas.

TABLE 3.4

FLUE GAS CONSTITUENTS FOR THE COMBUSTION OF HEATING OIL

Mixture component	% by volume	Mole fraction y_i	Molecular weight M_i	Mass per mole of mixture	mass fraction m_i
N_2	75.1	0.751	28	21.03	0.733
O_2	6.1	0.061	32	1.95	0.0679
H_2O	9.8	0.098	18	1.76	0.0613
CO_2	9.0	0.090	44	3.96	0.1380

The average constant pressure specific heat of the flue gas was estimated by the following equation:

$$\begin{aligned} c_p = {} & 1.7929\text{x}10^{-5}\theta^2 + 1.0089\text{x}10^{-5}\theta^{1.5} - 6.0461\text{x}10^{-3}\theta + 8.9865\text{x}10^{-2}\theta^{0.5} \\ & - 0.14915\theta^{0.25} + 0.37592 - 3.2884\theta^{-1.5} + 6.1804\theta^{-3} \\ & - 5.1177\theta^{-3} \quad (\text{Btu/lbm-}^{\circ}\text{F}), \end{aligned} \tag{3.23}$$

where θ is defined by equation (3.16).

The conductivity and absolute viscosity as a function of temperature ($^{\circ}$F) are readily obtained from Table 3.3 and Table 3.4, and are given by

$$k = 2.1357\text{x}10^{-5}T + 0.012257 \quad (\text{Btu/h-ft-}^{\circ}\text{F}), \tag{3.24}$$

and

$$\mu = 4.5319\text{x}10^{-5}T + 0.04158 \quad (\text{lbm/h-ft}) \tag{3.25}$$

3.2 The Furnace-Chimney System

The above discussion of the combustion process provides the basis for a furnace-chimney analysis. Typical residential chimney systems serving a single appliance (natural gas or oil-fired furnace), as shown in Fig. 3.1, are considered. The flue gas mass flow rate, assuming a minimal amount of air infiltration through the draft regulator, is given by [3]

$$\dot{m} = I\ M, \qquad (\text{lbm/h}) \tag{3.26}$$

where I is the heating appliance input (Btu/h), and M is the mass flow input ratio [lbm (flue gas)/1000 Btu (fuel burned)]. For typical residential burning equipment, M = 1.6 for a natural gas appliance, and M = 1.24 for a heating oil appliance [3].

The mean chimney flue gas temperature, which is often used as an average temperature throughout the furnace-chimney system, was calculated next. Various factors affect the mean chimney temperature, such as the fuel used and the flue pipe configuration. The mean chimney temperature is a function of the flue gas temperature rise (above ambient) in the heating system, which is estimated to be 300^{o}F for natural gas appliances, and 500^{o}F for heating oil appliances [3]. Also, the mean chimney temperature is governed by the amount of horizontal flue pipe breeching from the furnace to the chimney. The general relationship for the mean chimney temperature is given by

$$T_{cm} = T_R(C_u) + T_a \qquad (^{o}\text{F}), \tag{3.27}$$

where T_R is the system temperature rise (^{o}F), C_u a temperature multiplier, and T_a the ambient temperature (^{o}F). The temperature

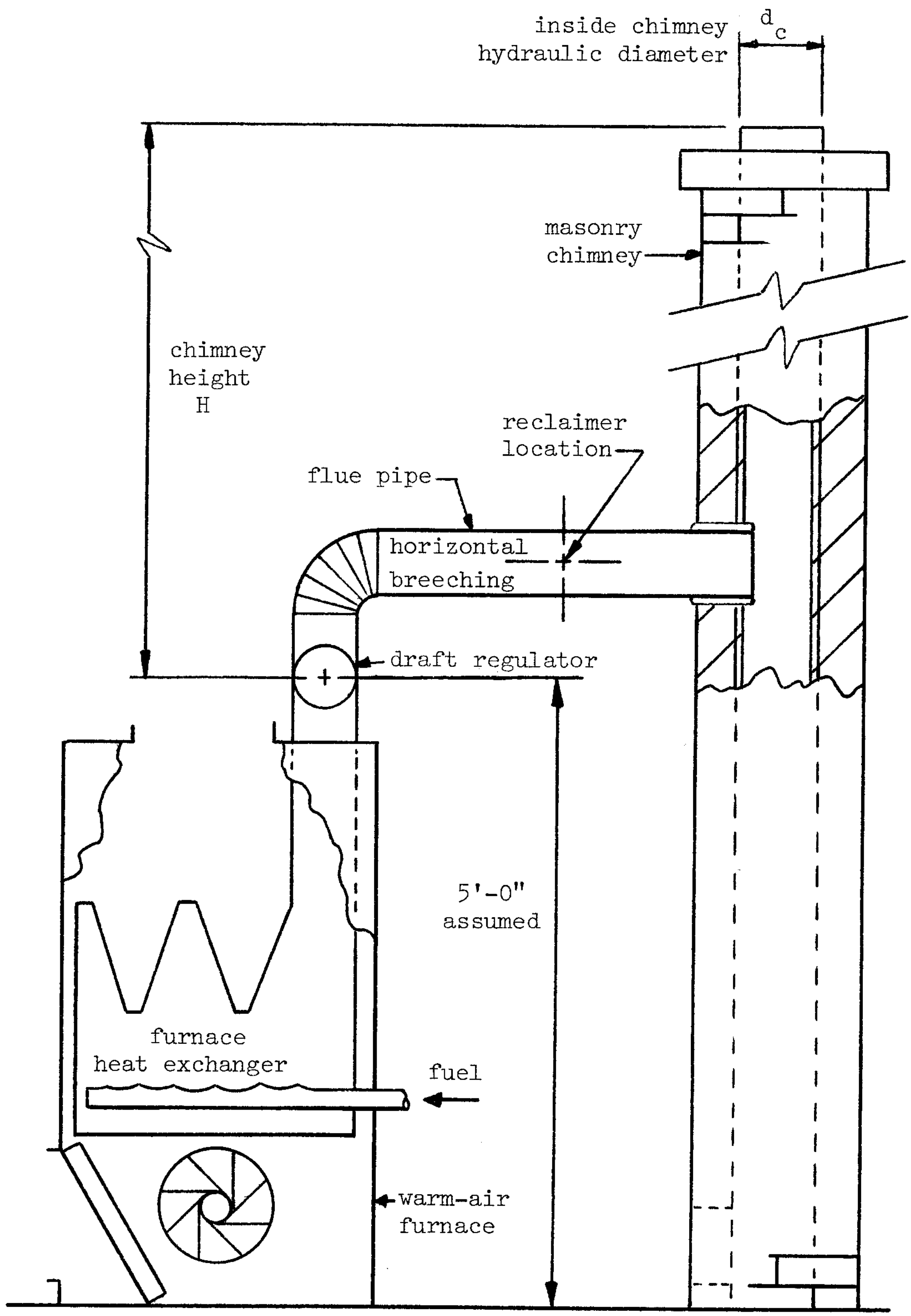


Fig. 3.1 Natural Draft Combustion Heating System

multiplier is given in Fig. 3.2 as a function of breeching length and was estimated as

$$C_{u4} = 0.800 - 0.0333L, \text{ for a 4 inch flue pipe diameter,} \qquad (3.28a)$$

$$C_{u5} = 0.815 - 0.0259L, \text{ for a 5 inch flue pipe diameter,} \qquad (3.28b)$$

and $$C_{u6} = 0.845 - 0.0202L, \text{ for a 6 inch flue pipe diameter,} \qquad (3.28c)$$

where L is the flue pipe horizontal length (ft).

Also, the flue gas density (based on the mean chimney temperature) can be found as [3]

$$\rho_m = 1.325(B/T_{cm}) \qquad (lbm/ft^3), \qquad (3.29)$$

where B is the local barometric pressure (28.86 inches of mercury, at 1000 ft elevation). Equation (3.29) assumes that air and flue gas have comparable densities at the same temperature.

The next step is estimating the system theoretical draft, which is the weight difference between equal columns of flue gas and ambient air per unit area [3]. The theoretical draft is written as

$$D_t = 0.2554\ B\ H(1/T_a - 1/T_{cm}) \quad \text{(inches of water column)}, \qquad (3.30)$$

where H refers to the chimney height (feet). The theoretical draft represents the maximum draft prior to system friction losses. Available draft is defined as the difference between the theoretical draft and the flow losses, for natural draft systems. It is the resulting available draft (updraft) that becomes a key safety design factor for the residential combustion heating system.

System pressure losses, due primarily to friction forces, are a function of the duct geometry, flow obstruction, piping roughness, and

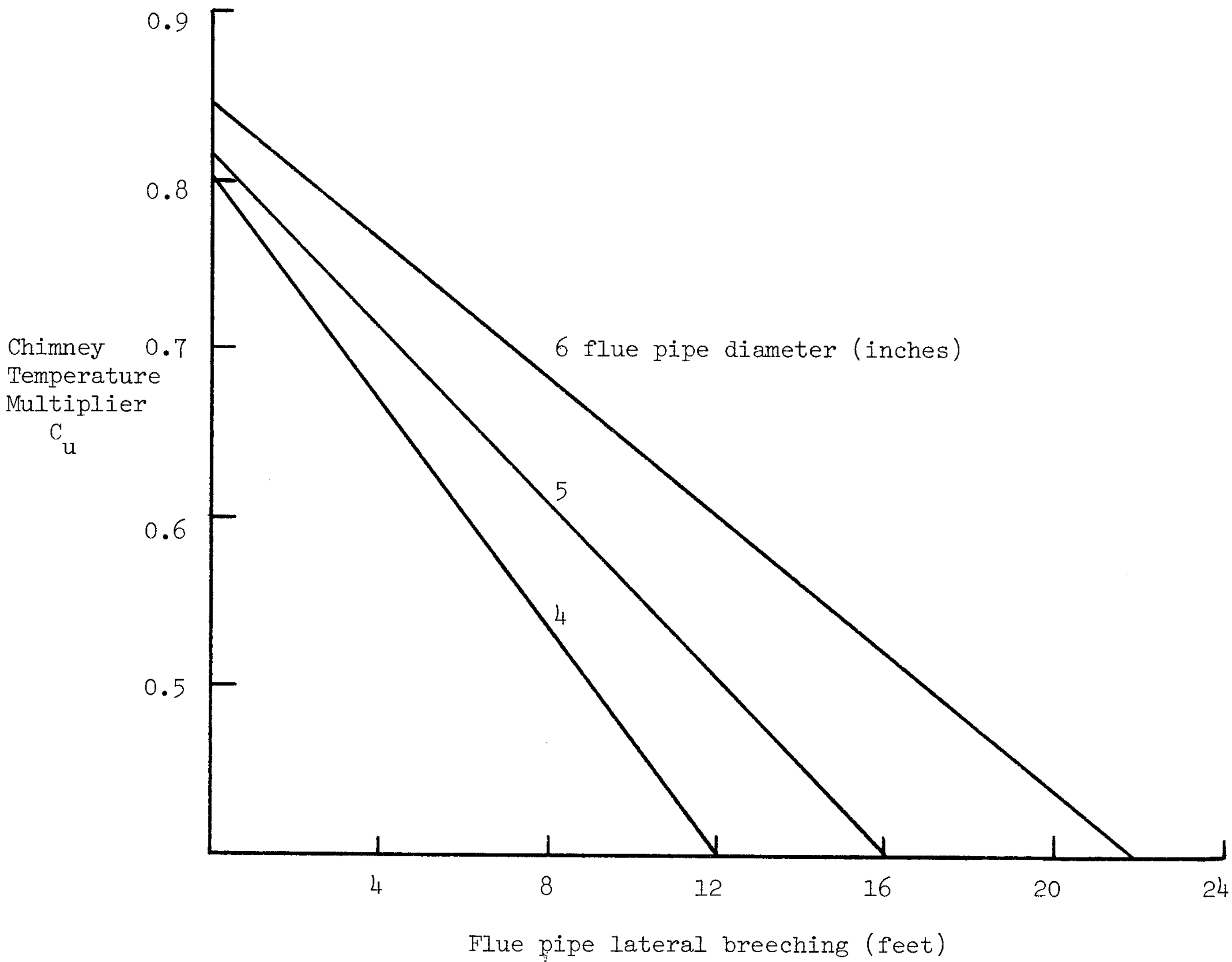


Fig. 3.2 Flue Pipe Horizontal Breeching vs. Chimney Temp. Multiplier

flow velocity. The calculation of system pressure losses begins with estimating the mean chimney flow velocity, which is given as [3]

$$V_{cm} = \dot{m}/(\rho_m A_c) \qquad (ft/h), \qquad (3.31a)$$

where A_c is the inside chimney cross-sectional area, based on the chimney hydraulic diameter. In terms of the hydraulic diameter,

$$V_{cm} = \dot{m}/(19.63 \rho_m d_c) \qquad (ft/s), \qquad (3.31b)$$

where d_c is the inside chimney hydraulic diameter (inches). Various hydraulic chimney diameters for standard nominal liner sizes are shown in Table 3.5.

Residential heating system flow losses, caused by friction, are generalized in the following equation [3]:

$$\Delta P = k_t \rho_m V_{cm}^2 /[(5.2)(2)g_1] \qquad \text{(inches of water column)}, \qquad (3.32)$$

where k_t is the total sum of the resistance coefficients, due to piping and fittings, and $g_1 = 32.2$ (ft/s^2). System resistance coefficients for various residential heating systems are shown in Table 3.6.

The next step is to calculate the system available draft (updraft) after the pressure losses. The general equation is [3]

$$D_a = D_t - \Delta P \qquad \text{(inches of water column)} \qquad (3.33)$$

It is stressed here that D_a will decrease if any additional flue pipe alterations, such as a heat reclaimer, are introduced into the system. A minimum available draft of 0.05 (inches of water column) was set as the design criterion for this thesis. This provided a safety factor against down wind outside the chimney.

Knowing the pressure drop ΔP, the initially assumed input is

Table 3.5 Masonry Chimney Sizes (3)

Masonry Chimney Liner Dimensions with Circular Equivalents*

Nominal Liner Size, In.	Inside Dimensions of Liner, In.	Inside Diameter or Equivalent Diameter, In.	Equivalent Area, Sq In.	Typical Outside Dimensions of Casing, In.
4 × 8	2 1/2 × 61/2	4	12.2	
		5	19.6	
		6	28.3	
		7	38.5	
8 × 8	6 3/4 × 6 3/4	7.4	42.7	16 × 16
		8	50.3	
8 × 12	6 1/2 × 10 1/2	9	63.6	16 × 21
		10	78.5	
12 × 12	9 3/4 × 9 3/4	10.4	83.3	21 × 21
		11	95.0	
12 × 16	9 1/2 × 13 1/2	11.8	107.5	21 × 25
		12	113.0	
		14	153.9	
16 × 16	13 1/4 × 13 1/4	14.5	162.9	25 × 25
		15	176.7	
16 × 20	13 × 7	16.2	206.1	25 × 29
		18	254.4	
20 × 20	16 3/4 × 16 3/4	18.2	260.2	29 × 29
		20	314.1	
20 × 24	16 1/2 × 20 1/2	20.1	314.2	29 × 34
		22	380.1	
24 × 24	20 1/4 × 20 1/4	22.1	380.1	34 × 34
		24	452.3	
24 × 28	20 1/4 × 24 1/4	24.1	456.2	34 × 38
28 × 28	24 1/4 × 24 1/4	26.4	543.3	38 × 38
		27	572.5	
30 × 30	25 1/2 × 25 1/2	27.9	607.0	48 × 48
		30	706.8	
30 × 36	25 1/2 × 31 1/2	30.9	749.9	48 × 54
		33	855.3	
36 × 36	31 1/2 × 31 1/2	34.4	929.4	54 × 54
		36	1017.9	

*SI conversions: mm = in. × 25.4; mm^2 = $in.^2$ × 645.

Talbe 3.6 Heating System Resistance Loss Coefficients (3)

Component	Resistance Loss Coefficients (Velocity Heads, Dimensionless) Suggested Design Value	Estimated Span and Notes
Inlet—acceleration		
Gas vent with draft hood	1.5	1.0 to 3.0
Barometric regulator	0.5	0.0 to 0.5
Direct connection	0.0	Also dependent on blocking damper position
Round elbow, 90 deg	0.75	0.5 to 1.5
Round elbow, 45 deg	0.3	—
Tee or 90 deg breeching	1.25	1.0 to 4.0
Y breeching	0.75	0.5 to 1.5
Cap, top,		
Open straight	0.0	—
Low resistance (UL)	0.5	0.0 to 1.5
Other	—	1.5 to 4.5
Spark Screen	0.5	
Converging exit cone	$(d_{i1}/d_{i2})^4 - 1$	System designed using d_{i1}
Tapered reducer (d_{i1} to d_{i2})	$1 - (d_{i2}/d_{i1})^4$	System designed using d_{i2}
Increaser		See Chapter 4, Fluid Flow, 1977 FUNDAMENTALS VOLUME
Piping k_L	$0.4 \dfrac{L, \text{ft (mm)}}{d_i, \text{in. (mm)}}$	Numerical coefficient from 0.2 to 0.5, see Fig. 4, Chapter 4, 1977 FUNDAMENTALS VOLUME for size, roughness, and velocity effects.

checked using the following equation [3]:

$$I = 413\ [(d_c)^2/M][\Delta P\ B/(k_t T_{cm})] \quad \text{(Btu/h)} \tag{3.34}$$

Table 3.7 relates the furnace input, flue pipe sizes, and chimney size and height.

The discussion above outlines how the pressure drop in a chimney system is calculated for any given furnace input.

The furnace exit temperature is important, because it helps in calculating other temperatures throughout the furnace-chimney system. The heat balance for flue gases passing over the furnace heat exchanger, can be written as

$$\eta I = Q_o \quad \text{(Btu/h)}, \tag{3.35}$$

where η is the overall furnace efficiency, which includes the combustion efficiency and the furnace heat exchanger efficiency, and Q_o the furnace output. The furnace output can also be written as

$$Q_o = \dot{m}C_p(T_{f1} - T_{f2}) \quad \text{(Btu/h)}, \tag{3.36}$$

where $(T_{f1} - T_{f2})$ is the flue gas temperature drop (^oF) within the furnace, C_p is the flue gas specific heat (Btu/lbm-^oF) at constant pressure, which is evaluated at the mean furnace temperature, and $\dot{m}$ is the flue gas mass flow rate. Combining equations (3.26) and (3.35) gives

$$T_{f2} = T_{f1} - 1000\eta/(M\ C_p) \quad (^oF), \tag{3.37}$$

where M is defined in equation (3.26). This relationship shows that the furnace outlet temperature is determined only by the fuel type, and not the system size or furnace input. If the furnace steady-state efficiency is assumed to be 80% (design conditions for a draft analysis), then

Table 3.7 Residential Chimney Capacities (3)

Capacity of Masonry Chimneys and Single-Wall Vent Connectors Serving a Single Appliance[a,b,*]

Height H, Feet	Lateral L, Feet	3	4	5	6	7	8	10	12
		Single-Wall Vent Connector Diameter, D, Inches (To be used with chimney areas not less than those at bottom) — Maximum Appliance Input Rating, Thousands of Btu/h							
6	2	28	52	86	130	180	247	400	580
	5	25*	48	81	118	164	230	375	560
8	2	29	55	93	145	197	265	445	650
	5	26*	51	87	133	182	246	422	638
	10	22*	44*	79	123	169	233	400	598
10	2	31	61	102	161	220	297	490	722
	5	28*	56	95	147	203	276	465	710
	10	24*	49*	86	137	189	261	441	665
	15	NR	42*	79*	125	175	246	421	634
15	2	35*	67	113	178	249	335	560	840
	5	32*	61	106	163	230	312	531	825
	10	27*	54*	96	151	214	294	504	774
	15	NR	46*	87*	138	198	278	481	738
	20	NR	NR	73*	128*	184	261	459	706
20	2	38*	73	123	200	273	374	625	950
	5	35*	67*	115	183	252	348	594	930
	10	NR	59*	105*	170	235	330	562	875
	15	NR	NR	95*	156	217	311	536	835
	20	NR	NR	80*	144*	202	292	510	800
30	2	41*	81*	136	215	302	420	715	1110
	5	NR	75*	127*	196	279	391	680	1090
	10	NR	66*	113*	182*	260	370	644	1020
	15	NR	NR	105*	168*	240*	349	615	975
	20	NR	NR	88*	155*	223*	327	585	932
	30	NR	NR	NR	NR	182*	281*	544	865
50	2	NR	91*	160*	250*	350*	475	810	1240
	5	NR	NR	149*	228*	321*	442	770	1220
	10	NR	NR	136*	212*	301*	420*	728	1140
	15	NR	NR	124*	195*	278*	395*	695	1090
	20	NR	NR	NR	180*	258*	370*	660*	1040
	30	NR	NR	NR	NR	NR	318*	610*	970
Minimum Internal Area of Chimney, A, Square Inches		19	19	28	38	50	68	95	132

[a]See Table 10 for Masonry Chimney Liner Sizes.
[b]See Fig. 19 and text section, Notes for Single Appliance Vents.
*SI conversions: W = Btu/h × 0.293; m = ft × 0.3048; mm = in. × 25.4; mm^2 = $in.^2$ × 645.

equation (3.37) becomes

$$T_{f2} = 2287 - 500/C_p \quad (^{\circ}F) \tag{3.38a}$$

for natural gas, where M = 1.6, and

$$T_{f2} = 2918 - 645.2/C_p \quad (^{\circ}F) \tag{3.38b}$$

for heating oil, where M = 1.24. Equations (3.38a) and (3.38b) are solved by trial and error, and it was found that for an 80% efficient furnace, the furnace exit temperature T_{f2} was 622°F, for natural gas heating systems, and 823°F, for heating oil systems.

The chimney exit temperature is also important, because it is the lowest temperature that the flue gases should experience throughout the heating system. The chimney exit temperature depends on many factors, such as the chimney size and height, the flue pipe geometry, the fuel burned, and the the flue gas mass flow rate, to name a few.

The chimney exit temperature was determined from the mean chimney temperature assuming that the flue gas temperature decreases linearly along its height. The average air temperature surrounding the chimney was assumed to be 68°F, because most of the chimney is in the heated area of the house. Also, it was assumed that the average overall heat-transfer coefficient for the chimney wall was 1.0 (Btu/h-ft^2-$^{\circ}$F) [3].

An energy balance for the chimney gave the following relation:

$$(\dot{m}C_p)\ (T_{c1} - T_{c2}) = U_c A_c (T_{cm} - T_{co}) \quad \text{(Btu/h)}, \tag{3.39}$$

where $(T_{c1} - T_{c2})$ is the difference in the chimney flue gas inlet and outlet temperatures ($^{\circ}$F), $(T_{cm} - T_{co})$ the difference in the mean chimney temperature and the exterior air temperature ($^{\circ}$F), U_c the chimney overall heat-transfer coefficient (Btu/h-ft^2-$^{\circ}$F), and A_c the inside chimney surface area based on the chimney hydraulic diameter (ft^2). The mean chimney temperature was approximated as

$$T_{cm} = (T_{c1} + T_{c2})/2 \qquad (^{o}F) \tag{3.40}$$

Equations (3.39) and (3.40) were combined to solve for T_{c1} and T_{c2}.

$$T_{c1} = [U_c A_c /(2\dot{m}C_p)](T_{cm} - T_{co}) + T_{cm} \qquad (^{o}F), \tag{3.41}$$

and

$$T_{c2} = 2T_{cm} - T_{c1} \qquad (^{o}F) \tag{3.42}$$

It will be shown in Chapter V that the addition of a heat reclaimer lowers the mean chimney temperature, which is equivalent to increasing the length of the horizontal flue pipe to the system. It will also be shown that lowering the furnace steady-state efficiency (below the design value of 80%) increases the furnace exit temperature, and the system flue gas temperature, which affect many of the design variables outlined.

CHAPTER IV

HEAT TRANSFER AND PRESSURE DROP IN THE TUBE-BANK HEAT RECLAIMER

Heat transfer and pressure drop are the two most important aspects of a tube bank heat exchanger analysis. Equations correlated from experimental data emphasize that the analysis of tube bank heat exchangers be performed for a specific flow through a specific tube configuration. The amount of heat transfer and the pressure drop are largely determined by the tube bank geometry, fluid properties, and flow velocity.

In general, an analysis of pressure drop for flow through tube-bank heat exchangers cannot be presented until the various temperatures are known or estimated. For best results, a heat transfer analysis is required to establish various temperatures in the reclaimer.

An energy balance for the heat reclaimer (Fig. 4.1) may be written as

$$Q_g = Q_t + Q_w, \tag{4.1}$$

where Q_g is the heat given up by the flue gas, Q_t is the heat gain by the tube bank, and Q_w is the heat loss from the exterior reclaimer surface to the surroundings. The modes of heat transfer include forced convection inside the reclaimer and tube bank, and natural convection and radiation from the exterior surface of the reclaimer. Radiation within the reclaimer, and the temperature drop across the (metal) walls and inside tubes of the reclaimer are assumed to be negligible.

Heat transfer from the exterior reclaimer surface to the

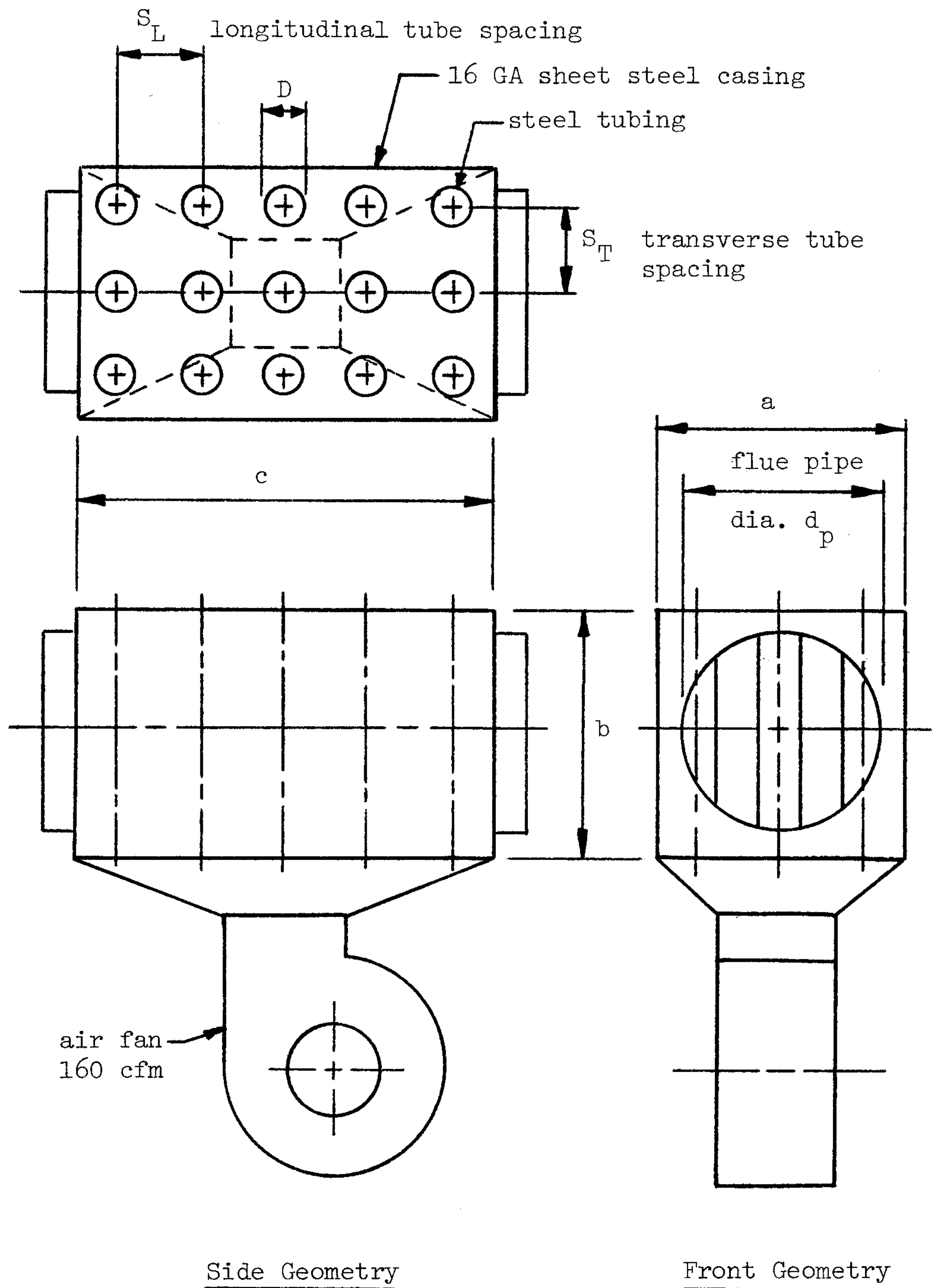


Fig. 4.1 Residential Heat Reclaimer

surroundings is given as

$$Q_w = h_{wo} A_w (T_w - T_{a1}) + \varepsilon A_w \sigma (T_w^4 - T_{a1}^4), \tag{4.2}$$

where

h_{wo} = average exterior reclaimer surface heat-transfer coefficient

A_w = exterior reclaimer surface area

T_w = average reclaimer surface area temperature

T_{a1} = surrounding air temperature

ε = emmissivity of the exterior reclaimer surface

σ = Stefan-Boltzmann constant

The heat loss by natural convection from the exterior surfaces of the reclaimer may be modeled as those from flat plates [1].

For the top surface,

$$h_{wo} = 0.27[(T_w - T_{a1})/c]^{1/4} \quad (\text{Btu/h-ft}^2\text{-}^{\circ}\text{F}), \tag{4.3}$$

for the bottom surface,

$$h_{wo} = 0.12[(T_w - T_{a1})/c]^{1/4} \quad (\text{Btu/h-ft}^2\text{-}^{\circ}\text{F}), \tag{4.4}$$

and for the sides,

$$h_{wo} = 0.29[(T_w - T_{a1})/b]^{1/4} \quad (\text{Btu/h-ft}^2\text{-}^{\circ}\text{F}), \tag{4.5}$$

where $(T_w - T_{a1})$ represents the temperature difference between the reclaimer casing and the surrounding air ($^{\circ}$F), and b and c are reclaimer characteristic lengths as shown in Fig. 4.1. The total heat transfer by natural convection is the sum of the heat transfer from the individual surfaces.

The analysis assumes that natural convection occurs on the six rectangular surfaces of the reclaimer, and that the surface temperatures are constant. Heat transfer from the blower and attachments was assumed negligible.

Heat transfer from the exterior reclaimer surface to the surroundings by radiation is assumed to occur from the six flat sides of the casing. The exterior reclaimer surface is assumed to be coated with an aluminum base paint that gives a normal emissivity ε of 0.6 [15].

Heat transfer from the flue gases to the interior surface of the reclaimer is considered as that due to forced convection through a short duct, which is given by

$$Q_{wi} = h_{wi} A_w (T_{gm} - T_w), \tag{4.6}$$

where h_{wi} is the interior reclaimer surface heat-transfer coefficient, A_w the interior reclaimer duct area, based on the hydraulic diameter (excluding the front and rear plates), and T_{gm} the bulk-gas temperature. Equation (4.6) assumes negligible heat transfer from flue gases to the front and rear casing plates.

The Nusselt number for flow in short ducts is given by [9]

$$Nu = 0.036 Re^{0.8} Pr_f^{0.33} (D_H/c), \tag{4.7}$$

where Re is the Reynolds number at the interior surface, Pr_f the Prandtl number (at the average film temperature), and c the reclaimer length.

The average flue gas Reynolds number is influenced by the presence of the tube bank. This was accounted for by defining a modified average Reynolds number

$$Re = Re_d (V_d)/(V_d - V_t), \tag{4.8}$$

where Re_d is the flue gas Reynolds number assuming no tube bank, V_d the total unobstructed interior volume of the reclaimer, and V_t the volume occupied by the tube bank.

The heat-transfer coefficient (h_{wi}) is calculated from the Nusselt number by

$$h_{wi} = Nu(k/D_H), \tag{4.9}$$

where Nu is the average Nusselt number, k the flue gas conductivity, and D_H the reclaimer hydraulic diameter. The flue gas properties used in equation (4.9) are evaluated at the film temperature $(T_w + T_{gm})/2$.

The heat transfer in the tube bank is described in the following equations:

For the exterior tube surface,

$$Q_t = h_{to}A_{to}(T_{gm} - T_t), \tag{4.10}$$

and for the interior tube surface,

$$(\dot{m}C_p)_a(T_{a2} - T_{a1}) = h_{ti}A_{ti}(T_t - T_{am}), \tag{4.11}$$

where

$(T_{gm} - T_t)$ = average temperature difference between the flue gas and the tube surface

$(T_t - T_{am})$ = average temperature difference between the air and the tube surface

$(T_{a2} - T_{a1})$ = difference between inlet and outlet temperatures of air in the tube

$(\dot{m}C_p)$ = product of air mass flow rate and constant pressure specific heat (evaluated at the tube air bulk-temperature)

h_{to}, h_{ti} = average exterior and interior tube heat-transfer coefficients

A_{to}, A_{ti} = exterior and interior tube surface areas

The air flow inside the tubes is similarly considered as that through a short duct, and the interior tube heat-transfer coefficient (h_{ti}) is estimated from equations (4.7), (4.8), and (4.9). The only exceptions are that the inside tube diameter, tube length, and air properties must be appropriately used, and that the flow Reynolds number is represented by

$$Re = 4\,\dot{m}_a/(\pi d_i\ \ \mu_m\ n_t), \tag{4.12}$$

where $\dot{m}_a$ is the total air mass flow rate through the tubes, μ_m the bulk-air viscosity, and n_t the number of tubes.

It is crucial in the heat transfer analysis that the fluid properties of the flue gas and the tube air are evaluated at the proper temperatures. Film and bulk temperatures are used to estimate many of the previous heat transfer relationships. However, the reference temperature used for the evaluation of the exterior heat-transfer coefficient (h_{to}) must be at the reclaimer upstream, or inlet temperature [29]. Choice of the reference temperature has been the cause of much controversy in the use of the correlations developed for tube bank heat exchangers.

The heat-transfer coefficient (h_{to}) is determined from correlations for flow through tube banks [29].

As the Reynolds number of a viscous fluid increases, the boundary layer separates from the tube and vorticities are shed. This separation point, measured at an angle ϕ from the leading edge of the tube, is a function of the flow Reynolds number and it varies for different types of flows [29]. In transitional Reynolds number flows, stability depends on the laminar boundary layer thickness, and the separation point does not vary much with the Reynolds number. The separation point is important when determining the average heat-transfer coefficient through a boundry layer.

Flow past a tube bank in a restricted channel is determined largely by the surrounding geometry, which influences the flow velocity and pressure distribution. Flow through in-line tube banks is comparable to flow through straight channels, whereas flow through staggered tube banks is comparable to flow through a series of converging and diverging nozzles. In both cases, flow past the leading tube row is similar to flow around single tubes, and flow through deeper rows is significantly shaded by the wakes of upstream tubes.

Heat transfer for flow through tube banks has been investigated by various researchers; some of the results are shown in Fig. 4.2a and 4.2b, for in-line and staggered tube arrangements respectively. Equations relating the Nusselt number for various tube banks of less than 20 rows are outlined as follows [29]:

For in-line tube banks, $10^2 < Re < 2x10^5$, $S_T/S_L > 0.7$,

$$Nu = C_n [0.27Re^{0.63} Pr_1^{0.36} (Pr_1/Pr_t)^{0.25}]. \tag{4.13}$$

For staggered tube banks, $10^2 < Re < 2x10^5$,

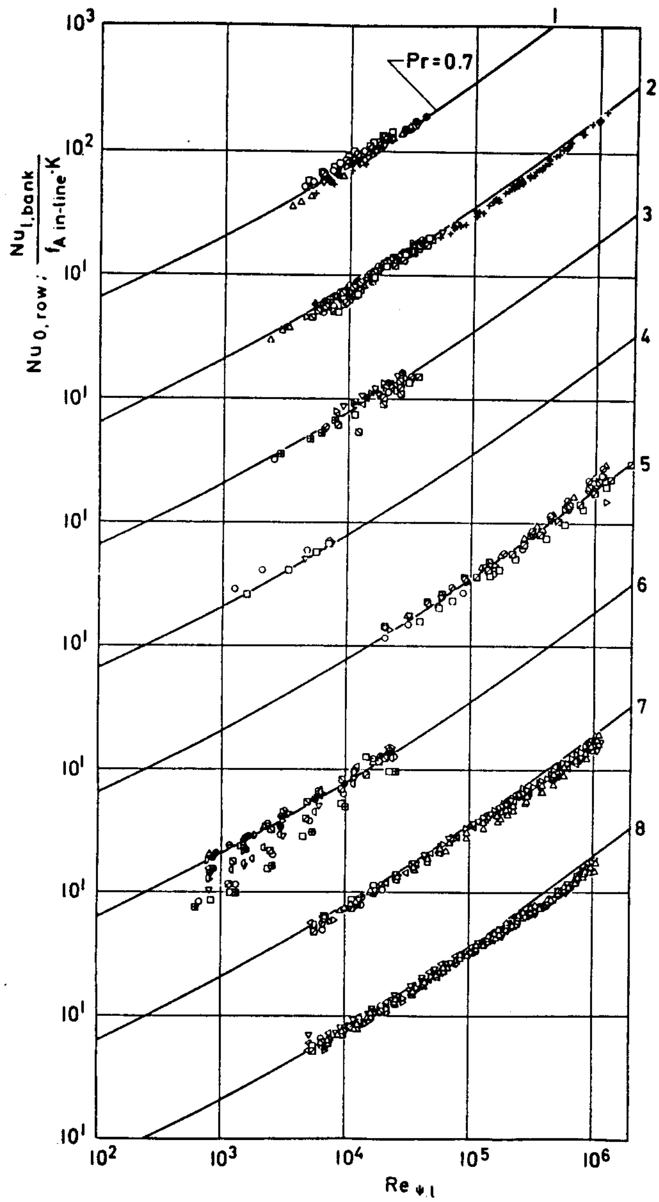
$$Nu = C_n [0.35(S_T/S_L)^{0.2} Re^{0.60} Pr_1^{0.36} (Pr_1/Pr_t)^{0.25}], \tag{4.14}$$

for $0.7 < (S_T/S_L) < 2.0$, and

$$Nu = C_n [0.40Re^{0.60} Pr_1^{0.36} (Pr_1/Pr_t)^{0.25}], \tag{4.15}$$

for $S_T/S_L > 2.0$,

where S_T and S_L are the transverse and longitudinal tube spacing respectively, and C_n the correction variable for less than 20 rows of tubes. Pr_1 and Pr_t are the Prandtl numbers at the upstream and average



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	▷		2.00	3.00	
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3	□	Žukauskas et al [5]	1.26	2.00	
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	■		1.30	2.60	
	○		1.65	2.00	
	●		1.65	2.00	
	●		1.95	1.30	
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	●		2.00	1.25	
	●		2.00	2.00	
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	D		4.00	1.25	
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	▣		1.50	1.50	0.25
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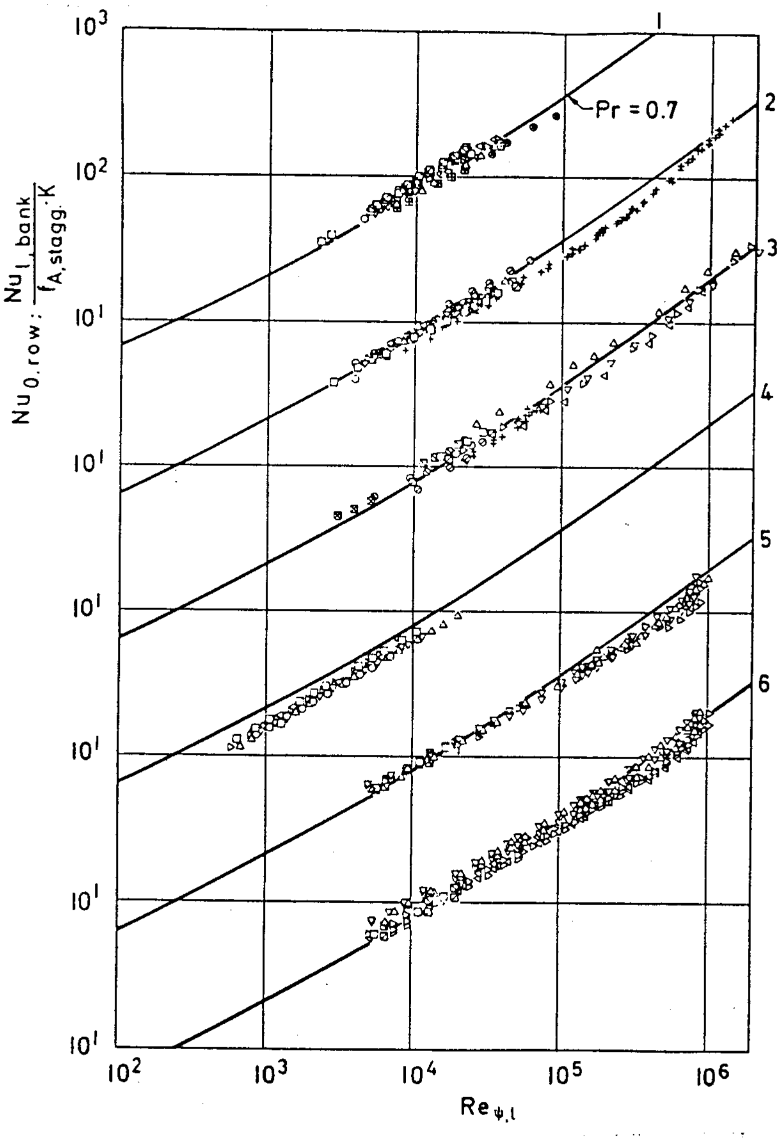
Fig. 4.2a In-line Tube Bank Arrangements

Fig. 4.2 Experimental Results for Heat Transfer for Flow Through Tube Banks (30)



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	⊗	Huge [12]	2.00	3.00	
	○		2.00	3.00	
	+	Hammecke et al. [13]	2.06	1.38	
3	□	Žukauskas et al. [5]	1.30	1.30	
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4	□	Kays et al. [19]	1.25	1.25	
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	▽		3.00	1.50	
	▷		3.00	2.00	1.50
	□	Bressler [17]	1.50	1.25	0.75
	⊠		1.50	1.50	0.75
	⊠		2.00	1.25	1.00
	■		2.00	2.00	1.00
6	△	Niggeschmidt [9]	1.50	3.00	0.50
	▽		1.50	3.00	1.00
	▷		2.00	1.50	0.75
	◁		3.00	2.00	1.12
	□	Bressler [17]	1.50	1.25	0.50
	⊠		2.00	1.25	0.50
	⊠		2.00	2.00	0.75

Fig. 4.2b Staggered Tube Bank Arrangements

Fig. 4.2 continued Experimental Results for Heat Transfer in Flow Through Tube Banks (30) tube wall temperatures respectively. Tube banks with $S_T/S_L < 0.7$ were not included in the analysis, because staggered banks of this type are not considered effective heat exchangers [29].

The flue gas Reynolds number used in equations (4.13), (4.14), and (4.15) is based on the tube-gap maximum velocity (u_m), given by the following [15]:

For in-line tube banks, and staggered tube banks where $S_T > S_D$,

$$u_m = u_1 S_T/(S_T - D), \tag{4.16}$$

and for staggered banks where $S_T < S_D$,

$$u_m = u_1(S_T)/2(S_D - D), \tag{4.17}$$

where u_m is the upstream flow velocity, D the outside tube diameter, and S_D the diagonal tube spacing, given by

$$S_D = [(S_T/2)^2 + S_L^2] \tag{4.18}$$

In general, the exterior tube heat-transfer coefficient is calculated by the average velocity integrated over the perimeter of each tube [29]. In most cases, however, the maximum flow velocity differs insignificantly from this integrated velocity. In compact tube banks, where tube gaps are less than about 5/32 inch, the maximum velocity may substantially increase and differ from the average velocity, resulting in high heat-transfer coefficients due to high local heat transfer over small portions of the tube. The corrected flow velocity is related to the separation angle ϕ by

$$u_\phi = u_1 S_T/(S_T - D \sin\phi). \tag{4.19}$$

Tube gaps of less than 5/32 inch were not considered in this thesis because of the adverse pressure losses associated with tightly spaced tube banks.

There are several pressure losses that occur in the heat reclaimer. The first two losses occur at the entrance and exit regions of the reclaimer. The flows in these regions can be considered as those through a duct with a sudden expansion, and contraction, for which relationships are readily found [1]:

For flow through a sudden enlargement,

$$\Delta P = C_{o1}(V_{p1}/4005)^2 \quad \text{(inches of water column)}, \qquad (4.22a)$$

and for flow through a sudden contraction,

$$\Delta P = C_{o2}(V_{p2}/4005)^2 \quad \text{(inches of water column)}, \qquad (4.22b)$$

where C_{o1} and C_{o2} are correction variables for sudden enlargements and contractions respectively, V_{p1} and V_{p2} are the flue pipe flow velocities at the reclaimer inlet and exit respectively, and are given by

$$V_{pj} = 2.4\,\dot{m}\,/(\rho_j A_p) \quad \text{(fpm)}, \qquad (4.23)$$

where the subscript j refers to reclaimer inlet or exit conditions.

Pressure drop through a bank of tubes is estimated by summing the individual pressure losses on each tube, which is impractical for most cases. Pressure drop is a function of the tube bank geometry (spacing and tube size), the number of tube rows, the flow velocity, and the fluid properties [20]. Pressure drop through tube banks is related to the dimensionless Euler number by

$$Eu = Eu(\,Re,\ S_T/D,\ S_L/D,\ n), \qquad (4.24)$$

where Re is the dimensionless Reynolds number, S_T/D and S_L/D the dimensionless transverse and longitudinal tube spacings respectively, and n the number of tube rows. The Euler number is related to a flow friction factor f by [20]

$$f = 2 \; \Delta P \rho / (4 \; \dot{m}^2 n) = 1/4 \; Eu, \tag{4.25}$$

and by the power law as

$$Eu = K_1 \; Re^K, \tag{4.26}$$

where K_1 and K are constants which are functions of the tube bank geometry.

Much of the available data solves K_1 and K only for typical industrial applications, where flow Reynolds numbers are well into the turbulent range. For most residential natural draft heating systems, maximum tube gap velocities are often in the transition Reynolds number region. For this reason the pressure drop analysis was based on correlations outlined by Gnielinski et al. [20], as shown in Fig. 4.3, where k_1 is a geometry variable.

The general equation for the pressure drop through in-line and staggered tube banks of less than 10 rows is [20]

$$\Delta P = C_m \; Eu_T \, \rho_m (u_m)^2 \, n, \tag{4.27}$$

where C_m is the correction variable for less than 10 rows of tubes, Eu_T the total Euler number, ρ_m the flue gas density, and u_m the maximum flue gas velocity. The flue gas properties are estimated at the bulk flue gas temperature between the reclaimer inlet and exit regions [20].

Estimating values for Eu for in-line and staggered tube banks of 10 or more rows can be accomplished either graphically, using Fig. 4.3, or through a series of correlated equations [20]. The graphical approach

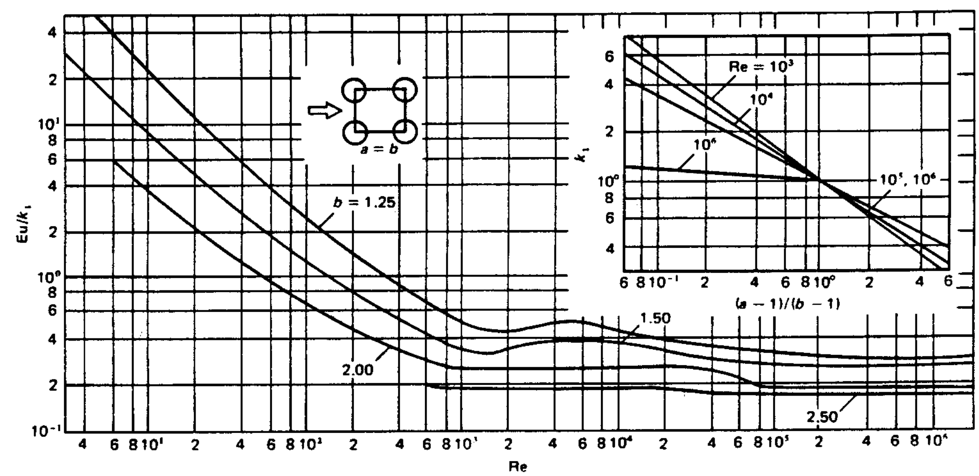


Fig. 4.3a In-line Tube Bank Arrangements

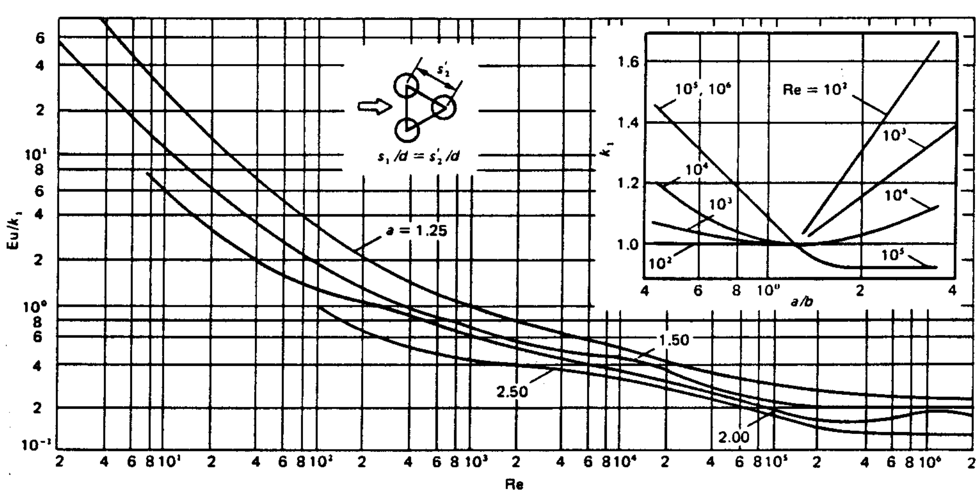


Fig. 4.3b Staggered Tube Arrangements

Fig. 4.3 Experimental Results for Pressure Drop in Flow Through Tube Banks (20) is ideal for initial pressure drop estimates, whereas the equations enable the user to enter the information on a computer, and evaluate specific flows and tube bank geometries in detail. The equations can treat a variety of flows and tube bank configurations which include transition Reynolds number flows.

Other influences affecting pressure drop through tube bank heat exchangers include property variation and flow acceleration, due to large temperature changes [20]. One such property variation is the wall-to-bulk viscosity, in which fluid flow near the exterior tube surface behaves differently (in the sense of flow friction) from the bulk-temperature fluid. Although the effect of wall-to-bulk viscosity does not substantially affect pressure drop in residential heat reclaimers, it is included here for completeness, and should be included in the flow Euler number.

Fluid acceleration or deceleration from a sudden temperature change should be added separately to the overall pressure loss in the reclaimer, and is given by

$$\Delta P = \rho_2 u_2^2 - \rho_1 u_1^2, \tag{4.28}$$

where the subscripts 1 and 2 refer to the reclaimer inlet and exit conditions.

An additional pressure loss would occur if the heat reclaimer were installed vertically, due to a fluid change of elevation, but this is usually negligible.

The total pressure drop in the heat reclaimer is the sum of the pressure losses at the entrance, through the tube bank, at the exit, and due to flow deceleration.

CHAPTER V

COMPUTER ANALYSIS OF THE HEAT RECLAIMER

The computer program generated solutions for various heat reclaimer designs and heating systems. The first step in developing the program was to define the design variables, which included environmental conditions, heating systems, tube bank geometries, and the reclaimer casing dimensions. The average air temperature surrounding the chimney was assumed to be 68^{o}F, and the average air temperature surrounding furnace 69^{o}F. Also, it was assumed that the outside barometric pressure was 28.86 inches of mercury (at 1000 feet elevation). The 1000 feet elevation was chosen as an average value for U.S. urban areas. The ambient temperature was 60^{o}F and lower.

The heating system consisted of the furnace, flue pipe, and chimney. The combustion heating systems were natural draft, warm-air furnaces that burn either natural gas or No.2 heating oil. The furnace input capacities ranged from 50,000 to 170,000 Btu/h, and had a steady-state efficiency of 80% and lower.

The flue pipe diameters ranged from 4 to 6 inches for the furnace inputs specified [3]. The flue pipe which was connected to the furnace, heat reclaimer, and chimney, included a draft regulator and a 90^{o} elbow, as shown in Fig. 3.1. Flue pipe horizontal breeching ranged from 2 to 4 feet.

The chimney was masonry-type with a rectangular lining, and served a single appliance as described in Table 3.5. The chimney inside hydraulic diameter ranged from 5 to 7 inches, and the chimney height ranged from 15 to 35 feet (where the height is measured from the draft regulator to the top of the chimney as shown in Fig. 3.1).

Correlations for various chimney capacities used in this analysis assumed 20% system oversizing, and are given in the following Table:

TABLE 5.1

DESIGN CHIMNEY SYSTEM CAPACITIES [3]

Chimney height (feet)	Flue dia. (inches)	Chimney hydr. dia. (inches)	Flue breeching (feet)	Furnace input (Btu/h)
15	4	5	2	54,000
15	5	6	2	90,000
15	6	7	2	142,000
20	4	5	2	58,000
20	5	6	2	98,000
20	6	7	2	160,000
25	4	5	3	60,000
25	5	6	3	101,000
25	6	7	3	161,000
30	4	5	3	63,000
30	5	6	3	106,000
30	6	7	3	167,000
35	4	5	4	---
35	5	6	4	109,000
35	6	7	4	169,000

Maximum chimney capacities are shown in Table 3.7. Oversizing flue pipes and chimneys by at least 20% is common practice in system design, and decreases both the system steady-state efficiency and the seasonal efficiency [3]. However, the amount of decrease does not affect the analysis significantly.

The reclaimer tube bank geometry considered tube bank variations such as tube configuration, tube diameter, and tube spacing. In-line and staggered tube bank arrangements were used, and staggered tube banks were assumed to always have the fewest number of tubes in the first row. Both in-line and staggered arrangements were restricted to fewer than 9 rows or fewer than 20 tubes, due to pressure loss. Special heat exchanger tubes were used, which have a thin tube wall thickness, and good heat transfer characteristics, as shown in the following Table:

TABLE 5.2

STEEL HEAT EXCHANGER TUBES [11]

Tube O.D. (inches)	Tube wall thickness (inches)	Tube I.D. (inches)	Weight per linear foot (lbs/ft)
0.625	0.035	0.555	0.2205
0.750	0.050	0.650	0.3738
0.875	0.050	0.775	0.4406
1.000	0.050	0.900	0.5073
1.250	0.050	1.150	0.6408
1.500	0.050	1.400	0.7743
1.750	0.650	1.620	1.1700

Both the tube bank and the reclaimer casing were constructed of steel from considerations of material and manufacturing costs. Other materials such as copper and aluminum were considered, but it was determined that the additional heat recovery was minimal and the manufacturing costs higher.

The transverse and longitudinal tube spacings were determined largely by the heat transfer and pressure drop equations, and were established as in Table 5.3:

TABLE 5.3

DIMENSIONLESS TUBE SPACING FOR COMPUTER ANALYSIS

	In-line tube banks	Staggered tube banks
S_T/D	1.75, 2.00, 2.25, 2.50	1.25, 1.50, 2.00, 2.50
S_L/D	1.25, 1.50, 2.00, 2.50	0.90, 1.20, 1.50, 1.80

The reclaimer casing size (Fig. 4.1) was determined by the tube spacing and the specifications given in Table 5.4:

TABLE 5.4

HEAT RECLAIMER CASING DIMENSION RESTRICTIONS

1. a = (maximum number of tubes per row) x (S_T)
2. a (minimum) = (d_p) + (1/4 inch), d_p = flue pipe diameter
3. a (maximum) = (d_p) x 1.5
4. b = a
5. c = [(n - 1) x (S_L)] + (2 x D), n = number of tube rows in the direction of flow
6. c (minimum) = a (minimum)
7. c (maximum) = d_p + (8 inches)

Restriction (1) suggests that if equations (4.16) and (4.17) are used to calculate the maximum inner-tube gap flow velocity, then the reclaimer side walls must be spaced close enough to the tubes so as not to hamper the flow through the tube bank. Keeping the reclaimer side walls close to the tube bank also forces flow through the tube bank as opposed to around the tube bank. Restrictions (2) through (7) were assumed.

Air properties such as specific heat at constant pressure, dynamic viscosity, and thermal conductivity are presented in Appendix B. Assuming linear variation between 60 and 120^{o}F,

$$C_p = 5.00 \times 10^{-5}\ T + 0.2401 \quad (\text{Btu/lbm-}^{o}\text{F}), \tag{5.1}$$

$$\mu = 6.3167 \times 10^{-5}\ T + 0.0396 \quad (\text{lbm/h-ft}), \tag{5.2}$$

and

$$k = 2.4833 \times 10^{-5}\ T + 0.0132 \quad (\text{Btu/h-ft-}^{o}\text{F}), \tag{5.3}$$

where T is the temperature (^{o}F). These equations agree favorably with tabulated data and were used throughout the analysis.

The reclaimer inlet temperature was assumed to be the mid-temperature between the calculated furnace and the chimney inlet temperatures. This assumption was made because the exact location of the heat reclaimer along the flue pipe was not readily known.

The core of the computer program analyzed the heat transfer in the reclaimer. From a heat balance in and around the reclaimer,

$$h_{wi}A_w(T_{gm} - T_w) = A_w h_{wo}(T_w - T_{a1}) + A\varepsilon\sigma\,(T_w^4 - T_{a1}^4), \tag{5.4}$$

$$(\dot{m}C_p)_a(T_{a2} - T_{a1}) = h_{ti}A_{ti}(T_t - T_{am}), \tag{5.5}$$

$$h_{to}A_{to}(T_{gm} - T_t) = U_{to}A\,\varepsilon\sigma(T_{gm} - T_{am}), \tag{5.6}$$

and

$$(\dot{m}C_p)_g(T_{g1} - T_{g2}) = U_{to}A_{to}(T_{gm} - T_{am}) + h_{wo}A_w(T_w - T_{a1}) + A_w\varepsilon\sigma(T_w^4 - T_{a1}^4), \tag{5.7}$$

where

T_{g1}, T_{gm}, T_{g2} = reclaimer flue gas inlet, bulk, and exit temperatures

T_{a1}, T_{am}, T_{a2} = tube air inlet, bulk, and outlet temperatures

T_w, T_t = reclaimer casing and tube wall temperatures

U_{to} = overall heat-transfer coefficient based on the exterior tube surface area

h_{to}, h_{ti} = exterior and interior tube heat-transfer coefficients

h_{wo}, h_{wi} = exterior and interior reclaimer casing heat-transfer coefficients

Equations (5.4) through (5.7) contain 4 unknowns, which are the flue gas bulk-temperature (T_{gm}), the tube air bulk-temperature (T_{am}), the tube wall temperature (T_t), and the reclaimer casing temperature (T_w). The unknown fluid properties are related to the unknown temperatures. Solution of equations (5.4) through (5.7) involved a multi-stage iteration process that was accomplished with the help of a computer.

Subroutines were created for the various flue gas and air properties. There are many occasions in both the heat transfer and the pressure drop analyses that the fluid properties at various temperatures were used.

The unknown reclaimer casing temperature T_w was assumed and equation (5.4) was solved for the flue gas bulk-temperature (T_{gm}). It may be noted that the interior reclaimer heat-transfer coefficient (h_{wi}) is a function of T_{gm}, and therefore, was independently iterated for proper convergence. Also, if equation (5.4) diverged, a new value for the reclaimer wall temperature (T_w) was assumed, and the process of solving equation (5.4) repeated.

Next, equations (5.5) and (5.6) were solved for the tube air bulk-temperature (T_{am}) and the tube wall temperature (T_t), respectively.

These equations were converged using the calculated flue gas bulk-temperature and the assumed values of T_{am} and T_t. If equation (5.5) or (5.6) diverged, then the initial reclaimer casing temperature (T_w) was incremented and equation (5.4) solved as above. In general, T_w causes rapid divergence in the solutions, and was initially assumed small and increased in small increments.

When solutions for T_{gm}, T_{am}, and T_t finally converged properly, equation (5.7) was used to check the casing wall temperature T_w. Equation (5.7) was independently iterated because of the nonlinearity of the term, which stems from the natural convection and radiation involved. The new T_w from equation (5.7) was compared to the previous T_w and the entire procedure was repeated until the difference in T_w was minimal.

The following were the criterion for temperature convergence:

For the flue gas Bulk-temperature,

$$\left| T_{gm} \text{ (calculated)} - T_{gm} \text{ (previous)} \right| < 0.01 \quad (^{o}F).$$

For the tube air bulk-temperature,

$$\left| T_{am} \text{ (calculated)} - T_{am} \text{ (previous)} \right| < 0.1 \quad (^{o}F).$$

For the average tube wall temperature,

$$\left| T_{t} \text{ (calculated)} - T_{t} \text{ (previous)} \right| < 0.1 \quad (^{o}F)$$

For the average reclaimer casing temperature,

$$\left| T_{w} \text{ (calculated)} - T_{w} \text{ (previous)} \right| < 1.0 \quad (^{o}F)$$

In short, the program numerically solved for the unknown temperatures through tedious iteration procedures, in which statement counters were introduced to prevent infinite loops. Also, several time-saving measures were inserted into the program, such as proper incrementing of the reclaimer casing temperature T_w.

The main program followed a sequence of steps which was intended to choose various reclaimers for typical heating systems, and to disregard unsafe reclaimer designs. The first step was to choose a specific reclaimer design for a specific heating system. Reclaimer designs were required to meet the geometry restrictions outined previously, or the program chose a new reclaimer design.

Next, a reclaimer temperature drop was assumed so that a pressure drop analysis could be performed. The intent of this step was to estimate whether or not the reclaimer was safe from a pressure drop point of view before a heat transfer analysis was performed, because the heat transfer analysis consumed most of the computer time. The reclaimer temperature drop was assumed at a minimal value ($30^{o}F$) to ensure that safe designs are not disregarded. The resulting system updraft, which is the difference between the system theoretical draft and the pressure losses, was to be above a value of 0.050 (inches of water column), to be considered safe. If the reclaimer did not meet this criterion, then the program established a new design, and if the criterion is met, then the program proceeded to the heat transfer analysis.

A new mean chimney temperature was calculated due to the effects of the temperature drop through the reclaimer. This method compared the reclaimer to an equivalent length of flue pipe having a similar temperature drop. The heat balance matching the reclaimer temperature loss with that of an equal length of flue pipe was written as

$$(\dot{m}C_p)_g(T_{g1} - T_{g2}) = U_p \pi d_p L_e(T_{gm} - T_{a1}), \tag{5.8}$$

where

$(T_{gm} - T_{a1})$ = temperature difference between the interior and exterior of the flue pipe

U_p = flue pipe overall heat-transfer coefficient [3]

d_p = flue pipe diameter

L_e = flue pipe equivalent length

Solving for the equivalent horizontal flue pipe length (L_e) gave

$$L_e = (\dot{m}C_p)_g(T_{g1} - T_{g2})/[U_p \pi\ d_p(T_{gm} - T_{a1})]. \quad (5.9)$$

This provided a method of determining the actual mean chimney temperature, which was needed for calculating the chimney inlet and exit temperatures, and consequently for determining whether the flue gases condense in the chimney. Also, the mean chimney temperature was needed for calculating a new theoretical draft and system pressure losses.

The system updraft could be estimated with the correct reclaimer temperature drop and was again checked for safety against the design criteria. Once the reclaimer design was proven safe, heat recovery was determined.

Finally, the total weight was calculated by

$$W_T = A_w\gamma_w + n_t\gamma_t b + W_b \quad \text{(lbs)}, \quad (5.10)$$

where γ_w is the reclaimer weight per square foot of 16 GA. steel (2.5 lbs/ft), γ_t the tube weight per linear foot (see Table 5.2), and W_b the approximated weight of the blower and accessories (6 lbs.).

The program printed in two modes. In the initial mode, the input variables were the transverse and longitudinal tube spacings (Table 5.3), outside tube diameters (Table 5.2), flue pipe and chimney hydraulic diameters, and chimney heights (Table 5.1). This amounted to 1568 different reclaimer designs per tube configuration, of which many were discarded due to restrictions on pressure drop and reclaimer dimensions. The output was tabulated and the optimum designs chosen.

In the final mode, a specific reclaimer was selected, usually the optimum reclaimer design from the initial mode, and additional information about the reclaimer was printed. The purpose of this mode was to concentrate on one tube bank design and evaluate it under various conditions, such as lower furnace steady-state efficiencies and ambient temperatures.

It is important to note here that ASHRAE assumes a constant chimney temperature rise for oil and natural gas furnaces under design conditions [3]. However, as the furnace efficiency decreases, the temperature rise in the chimney will clearly increase, due to the substantial increase of the furnace exit temperature. To recognize this effect, the chimney temperature rise was assumed only as a function of the furnace exit temperature, which is described by

$$T_R = (T_{f2} - T_s)/C_R, \tag{5.11}$$

where T_{f2} is the furnace exit temperature, and T_s the design temperature, which is $60^\circ F$ [3]. The constant C_R depends on the fuel burned and was estimated by

$$C_R = (\hat{T}_{f2} - T_s)/\hat{T}_R, \tag{5.12}$$

where $\hat{T}_{f2}$ is the furnace exit temperature at the design conditions, and $\hat{T}_R$ the chimney temperature rise stipulated by ASHRAE, for natural gas and oil-fired equipment. The value of C_R for natural gas furnaces was estimated to be 1.8733, and for oil-fired furnaces, 1.544. These constants helped compensate for the increase in chimney temperature rise due to inefficient furnaces, found typically in existing homes.

Flow charts describing the the main program and heat transfer calculations are shown on Figures (5.1) through (5.3). The program and a list of program variables are given in Appendix C and D, respectively.

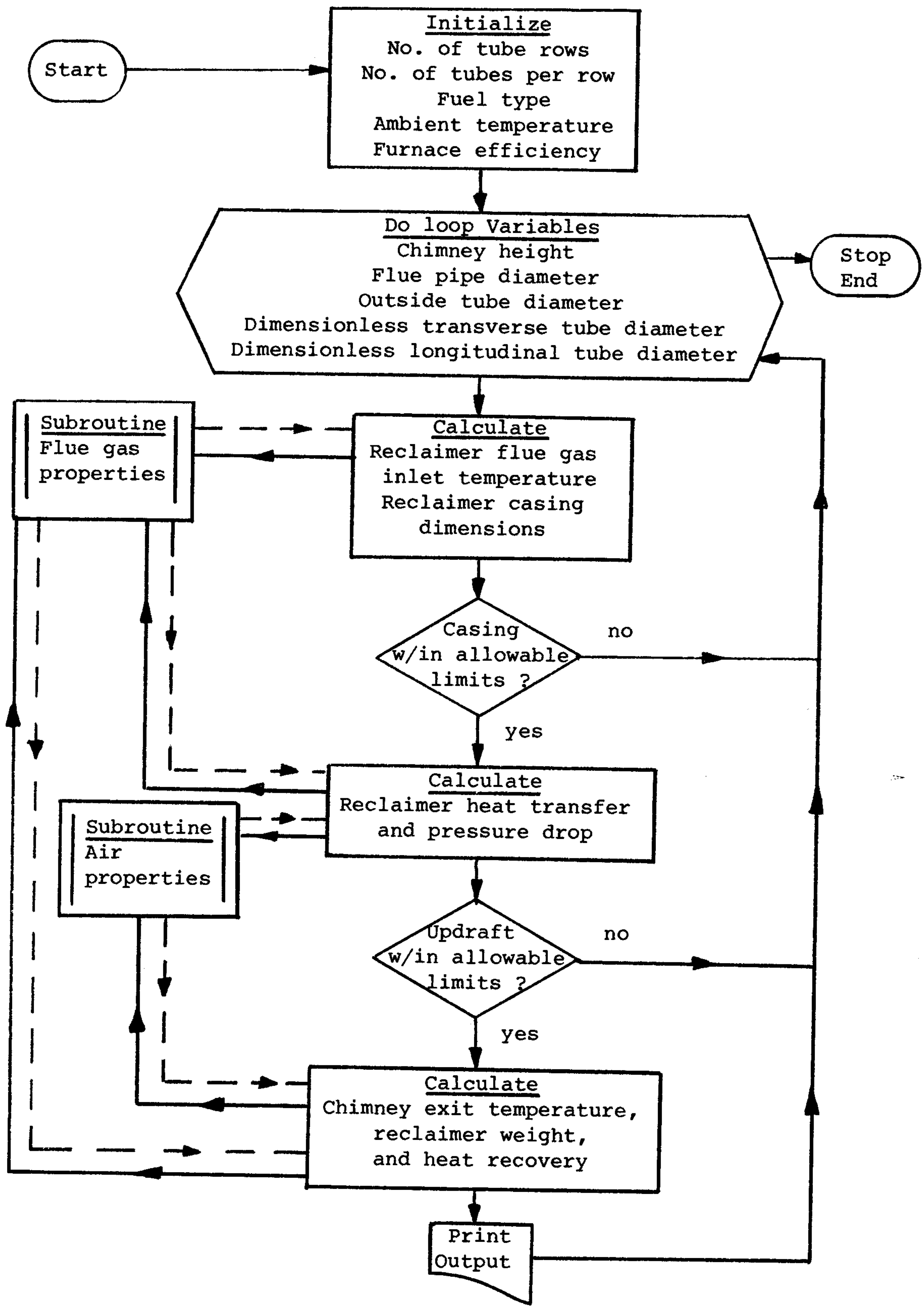


Fig. 5.1 Simplified Program Flow Diagram for Initial Execution Mode

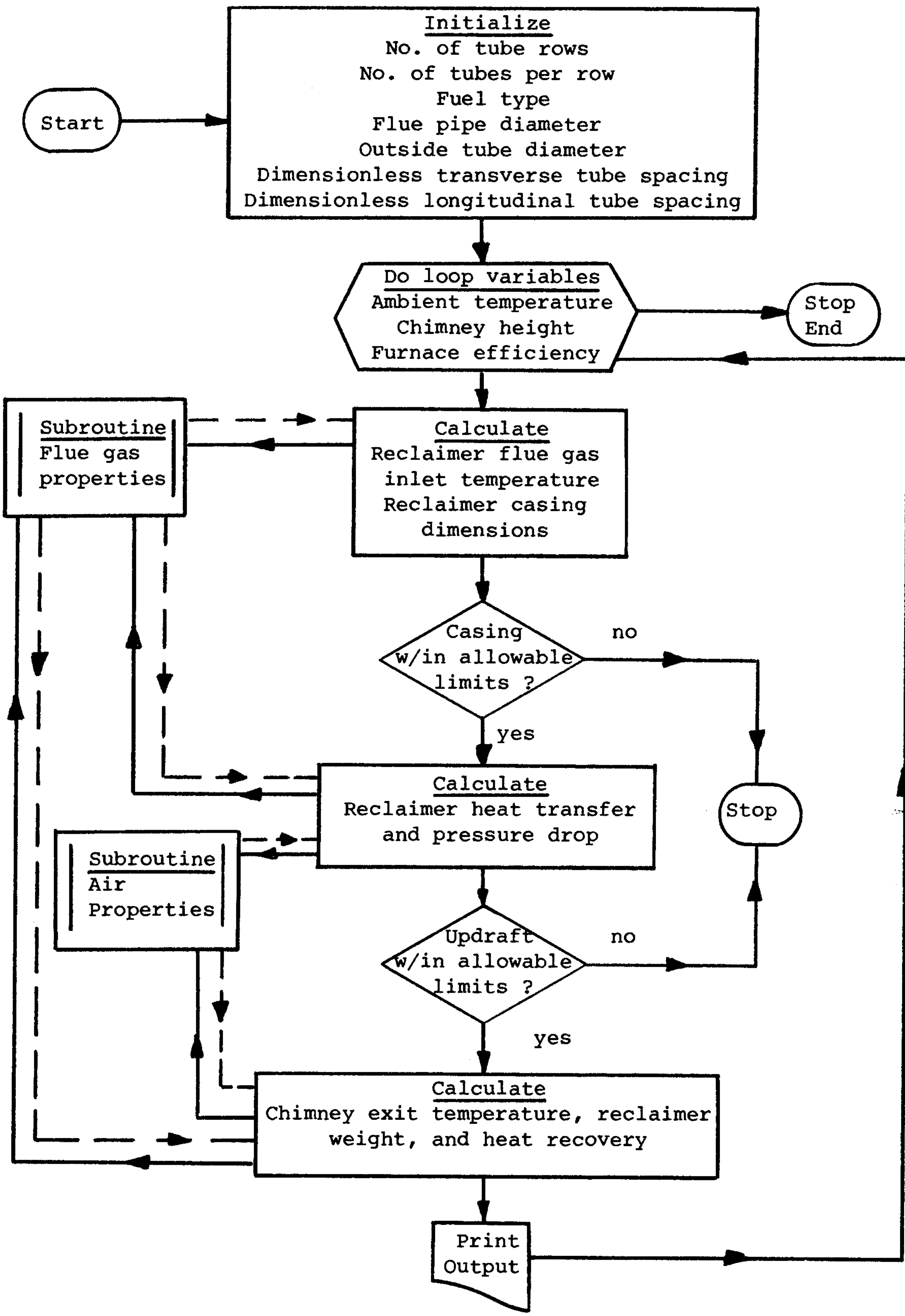
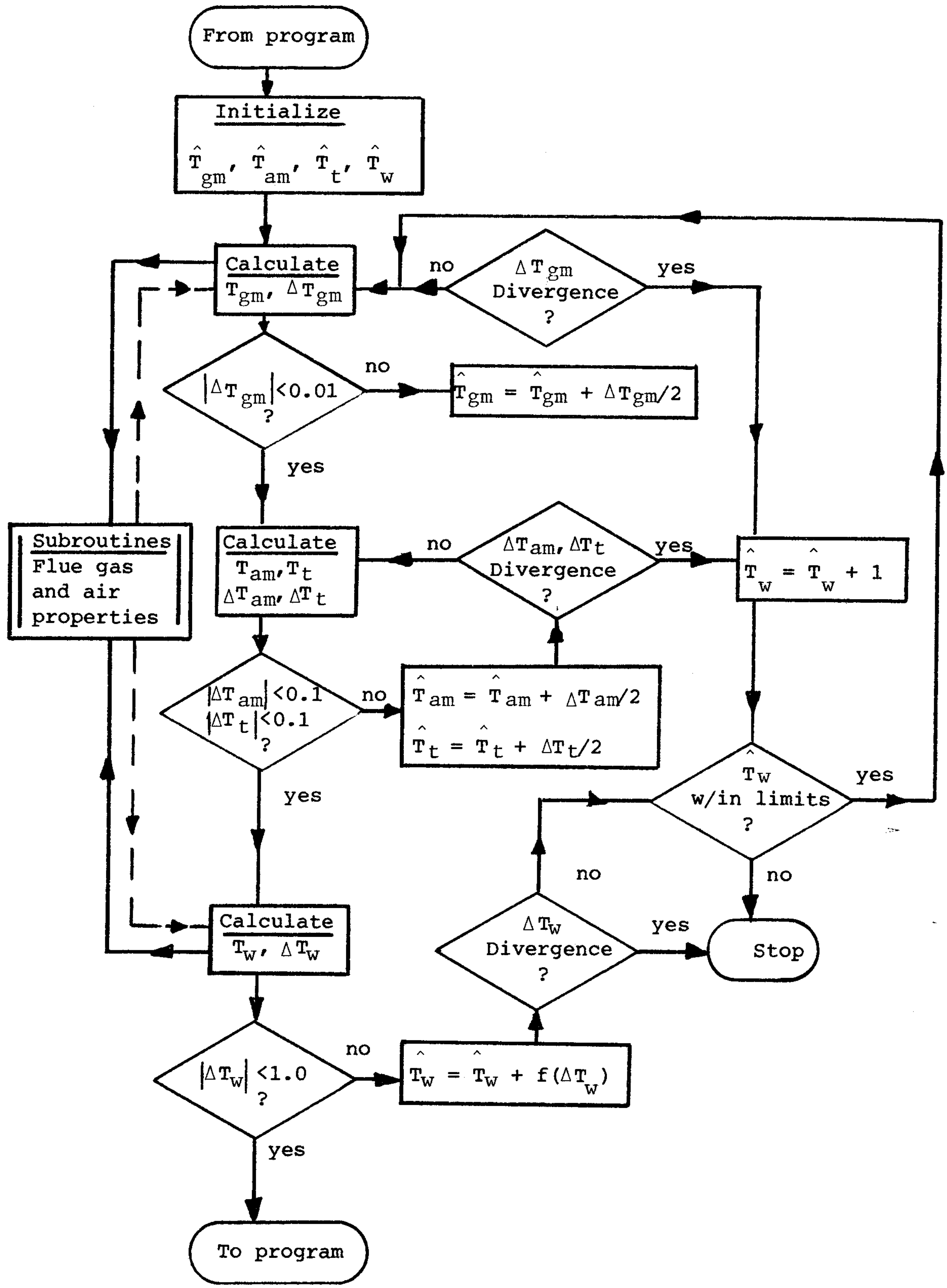


Fig. 5.2 Simplified Program Flow Diagram for Final Reclaimer Design Mode



5.3 Simplified Heat Transfer Flow Diagram

CHAPTER VI

RESULTS AND OPTIMIZATION

The results brought out several important facts about tube bank heat reclaimers. Among the most important findings was the fact that in-line tube bank arrangements were generally superior to comparable staggered tube bank arrangements. The pressure drop was clearly the greatest deterrent in the use of staggered tube banks. Most staggered tube banks of 7 or more tubes did not meet satisfactory updraft requirements for chimney heights of less than 25 feet. It was discovered in certain cases of in-line and staggered arrangements, using the same number of tubes, that heat recovery for in-line arrangements was slightly better than that for staggered arrangements. These cases occurred in higher chimneys where pressure drop was not an important factor.

The important factors which influenced heat recovery were tube spacing and the number of tube rows. For in-line tube arrangements, the transverse tube spacing determines the maximum flow velocity and Reynolds number, which the pressure drop and heat transfer are partially based on. The number of tube rows in the direction of flow also affects the pressure drop and heat transfer, but does not affect the pressure drop as much as the number of transverse tube spacings does. For example, when comparing in-line tube banks, 3 x 4 (rows x columns) and 4 x 3, the 4 x 3 tube banks recovered, on the average, 6% more heat than similar 3 x 4 tube banks. The trade-off was only a 4% decrease in updraft. Staggered tube banks showed similar trends.

Heat reclaimers used with a heating system having a chimney height of less than 20 feet (single-story, slab foundation residence) are not recommended, primarily because of the inadequate draft produced with the reclaimer installed. A few arrangements of less than 7 tubes were found safe for pressure drop, but they did not reclaim a justifiable amount of heat. For example, an in-line tube bank (2 x 3) was found safe for 15 foot high chimneys using a 5 or 6 inch flue pipe diameter, but it recovered less than 10% of the lost heat, which amounted to less than 1.5% fuel savings. In fact, it was not until chimney heights researched above 20 feet (two-story homes and above) that significant amounts of heat were reclaimed.

For chimney heights of 25 feet and above, it was discovered that for flue pipe diameters of 4 inches the heat reclaimers would cause flue gas condensation in the chimney. Some reclaimer designs were found to prevent flue gas condensation, but were much smaller than the ones which recovered significant amounts of heat. This, however, does not pose much of a problem in light of the fact that most two-story residences, with chimney heights of 25 feet and above, require greater furnace inputs, and consequently, larger flue pipe diameters, as shown in Fig. 3.7.

In addition, when choosing an optimum reclaimer design for each heating system, certain correlations were looked for in the results which could be used for grouping like reclaimers for various heating systems. The versatility of a particular reclaimer design was carefully searched for, and Table 6.1 shows that for chimney heights of 25 feet and above, using flue pipe diameters of 5 and 6 inches, one tube bank (5 x 3) dominates as the most effective reclaimer design. The design, using an outside tube diameter of 1.25 inches, and a transverse and longitudinal tube spacing of 2.50 inches, was within 5% of the maximum heat recovery for any of the six designs that it could be used on. Other reclaimers,

Table 6.1 Results for Typical Residential Heating Systems (furnace eff. = 80%, ambient temp. = 60°F)

In-line tube array (rows x col.)	Chimney height (ft)	Flue pipe dia. (in.)	Furnace input rate (Btu/h)	Tube O.D. (in.)	Trans. tube spacing (in.)	Longit. tube spacing (in.)	Reclaimer casing size a = b (in.)	Reclaimer casing size c (in.)	Total weight (lb)	Lost heat recovered Oil %	Lost heat recovered N.G. %
4 x 3	20	4	58,000	0.75	1.688	0.938	5.06	4.31	9.7	27.5	19.1
4 x 3	20	5	98,000	1.00	2.250	1.250	6.75	5.75	12.7	22.9	15.7
4 x 3	20	6	160,000	1.00	2.500	1.250	7.50	5.75	13.4	16.9	11.9
5 x 3	25	5	101,000	1.25	2.500	2.500	7.50	12.50	19.2	30.9	20.9
5 x 3	25	6	161,000						18.9	27.6	18.6
5 x 3	30	5	106,000						19.2	31.1	20.8
5 x 3	30	6	167,000						18.9	23.3	16.2
5 x 3	35	5	109,000						19.2	30.0	20.6
5 x 3	35	6	169,000						18.9	23.4	16.1

using tube bank arrangements of 6 x 3 and 5 x 4, were found safe for chimney heights of 25 feet and above, but universal designs were not discovered. Similar grouping of designs for chimney heights of 20 feet could not be accomplished due to the strong influence of pressure drop. Such designs for 4, 5, and 6-inch flue pipe diameters used 4 x 3 tube arrangements and had various tube diameters, and transverse and longitudinal tube spacings.

As a special note, (4 x 3) reclaimer designs did not meet the pressure drop restriction for heating systems using a 20 foot high chimney and 6-inch flue pipe diameter. However, after reducing the overall reclaimer length restriction by 1/4 inch, it was found that the pressure drop was significantly reduced, and a safe reclaimer design was established.

It was discovered that the tube bank was responsible for approximately 85% of the total heat recovery and the reclaimer casing was responsible for the rest, where the tube bank heat output Q and the reclaimer casing heat output Q were calculated from equations (4.10) and (4.2) respectively.

Figures 6.1a-c and 6.2a-c show some of the final designs as they appeared in the initial computer output. Optimum designs were chosen under several considerations using the output for in-line and staggered tube banks ranging from 4 to 20 tubes. The four main considerations were safety, material and manufacturing cost, heat recovery, and weight. Safety was defined in terms of allowable pressure drop and chimney flue gas exit temperature. The manufacturing cost was determined by the manufacturing processes involved for a given choice of material. Steel was used for obvious reasons. The heat recovery was defined in terms of the per cent of heat loss recovered and the fuel saved.

A S S U M E D   C O N D I T I O N S

NUMBER OF TUBES: 12

TUBE CONFIGURATION

```
|0 0 0|
|0 0 0|
|0 0 0|
|0 0 0|
```

TUBES: STEEL HEAT EXCHANGER TYPE

CASING: CONSTRUCTED OF 16 GA STEEL

FUEL: NATURAL GAS

BAROMETRIC PRESSURE: 28.86 INCHES MERCURY

OUTSIDE AIR TEMPERATURE: 60DEGREES FAHRENHEIT

FURNACE ROOM TEMPERATURE: 69 DEGREES FAHRENHEIT

FURNACE STEADY-STATE EFFICIENCY: 80%

FURNACE EXIT TEMPERATURE: 622 DEGREES FAHRENHEIT

GAS MASS FLOW RATE(POUND-MASS/HR): APPROXIMATELY 1.6000 TIMES THE FURNACE INPUT

AIR FLOW THROUGH TUBE BANK: 160 CFM

CHIMNEY CONSTRUCTION: MASONRY TYPE WITH LINER

K E Y   N O T E S
****

FOR ALL CHIMNEY HEIGHTS, 4,5,6 INCH DIA FLUE PIPES CORRESPOND TO 5,6,7 INCH EQUIVALENT CHIMNEY DIAMETERS.

LATERAL BREECHING OF 2 FEET CORRESPONDS TO CHIMNEY HEIGHTS OF 15 AND 20 FEET.

LATERAL BREECHING OF 3 FEET CORRESPONDS TO CHIMNEY HEIGHTS OF 25 AND 30 FEET.

LATERAL BREECHING OF 4 FEET CORRESPONDS TO A CHIMNEY HEIGHT OF 35 FEET.

Fig. 6.1a Assumed Conditions and Key Notes

Fig. 6.1 Example of Initial Execution Mode Output for 4 x 3 Tube Arrays

			TUBE BANK ARRAY			BOX DIMENSIONS				ALL TEMPS DEGREES FAHRENHEIT								
MASON CHIM HEIGHT FEET	FURN INPUT *1000 BTU/HR	FLUE PIPE DIAM INCH	TUBE O.D. INCHES	TRANS SPCG INCHES	LONGIT SPCG INCHES	16 GA WIDTH INCHES	SHEET DEPTH INCHES	STEEL LENGTH INCHES	GAS MAX RE NO.	BOX GAS INLET	BOX GAS OUTLET	TUBE AIR EXIT	BOX WALL TEMP	CHIM GAS EXIT	APPROX TOTAL WEIGHT LBS	DRAFT W/UNIT INCHES WATER	TOTAL HEAT GAIN BTU/HR	% HEAT LOSS RECOV
20.	58.	4.	1.000	1.750	1.500	5.25	5.25	6.50	***INSUFFICIENT DRAFT***									
20.	58.	4.	1.000	1.750	2.000	5.25	5.25	8.00	***INSUFFICIENT DRAFT***									
20.	58.	4.	1.000	1.750	2.500	5.25	5.25	9.50	***INSUFFICIENT DRAFT***									
20.	58.	4.	1.000	2.000	1.250	6.00	6.00	5.75	***INSUFFICIENT DRAFT***									
20.	58.	4.	1.000	2.000	1.500	6.00	6.00	6.50	***INSUFFICIENT DRAFT***									
20.	58.	4.	1.000	2.000	2.000	6.00	6.00	8.00	***INSUFFICIENT DRAFT***									
20.	58.	4.	1.000	2.000	2.500	6.00	6.00	9.50	***INSUFFICIENT DRAFT***									
20.	98.	5.	0.750	1.875	1.500	5.63	5.63	6.00	1282.	498.3	431.1	82.3	232.2	177.6	10.7	0.0494	2786.	14.2
20.	98.	5.	0.750	1.875	1.875	5.63	5.63	7.13	1283.	498.3	429.9	82.3	232.5	177.1	11.1	0.0494	2836.	14.5
20.	98.	5.	0.875	1.750	1.313	5.25	5.25	5.69	***INSUFFICIENT DRAFT***									
20.	98.	5.	0.875	1.750	1.750	5.25	5.25	7.00	***INSUFFICIENT DRAFT***									
20.	98.	5.	0.875	1.750	2.188	5.25	5.25	8.31	***INSUFFICIENT DRAFT***									
20.	98.	5.	0.875	1.969	1.313	5.91	5.91	5.69	1468.	498.3	426.5	83.4	225.1	175.6	11.2	0.0493	2977.	15.2
20.	98.	5.	0.875	1.969	1.750	5.91	5.91	7.00	***INSUFFICIENT DRAFT***									
20.	98.	5.	0.875	1.969	2.188	5.91	5.91	8.31	***INSUFFICIENT DRAFT***									
20.	98.	5.	0.875	2.188	1.313	6.56	6.56	5.69	1100.	498.3	428.7	82.9	204.7	176.6	12.0	0.0516	2885.	14.7
20.	98.	5.	0.875	2.188	1.750	6.56	6.56	7.00	1101.	498.3	427.3	82.9	205.6	176.0	12.6	0.0507	2944.	15.0
20.	98.	5.	0.875	2.188	2.188	6.56	6.56	8.31	1101.	498.3	425.9	82.8	206.6	175.4	13.2	0.0505	3001.	15.3
20.	98.	5.	1.000	1.750	1.250	5.25	5.25	5.75	***INSUFFICIENT DRAFT***									
20.	98.	5.	1.000	1.750	1.500	5.25	5.25	6.50	***INSUFFICIENT DRAFT***									
20.	98.	5.	1.000	1.750	2.000	5.25	5.25	8.00	***INSUFFICIENT DRAFT***									
20.	98.	5.	1.000	1.750	2.500	5.25	5.25	9.50	***INSUFFICIENT DRAFT***									
20.	98.	5.	1.000	2.000	1.250	6.00	6.00	5.75	***INSUFFICIENT DRAFT***									
20.	98.	5.	1.000	2.000	1.500	6.00	6.00	6.50	***INSUFFICIENT DRAFT***									
20.	98.	5.	1.000	2.000	2.000	6.00	6.00	8.00	***INSUFFICIENT DRAFT***									
20.	98.	5.	1.000	2.000	2.500	6.00	6.00	9.50	***INSUFFICIENT DRAFT***									
20.	98.	5.	1.000	2.250	1.250	6.75	6.75	5.75	1286.	498.3	423.9	83.9	202.3	174.6	12.7	0.0502	3083.	15.7

Fig. 6.1b Output Sample (4 x 3 tube array)

Fig. 6.1 continued Example of Initial Execution Mode Output for 4 x 3 Tube Arrays

MASON CHIM HEIGHT FEET	FURN INPUT *1000 BTU/HR	FLUE PIPE DIAM INCH	TUBE BANK ARRAY TUBE O.D. INCHES	TRANS SPCG INCHES	LONGIT SPCG INCHES	BOX DIMENSIONS 16 GA SHEET STEEL WIDTH INCHES	DEPTH INCHES	LENGTH INCHES	GAS MAX RE NO.	ALL TEMPS DEGREES FAHRENHEIT BOX GAS INLET	BOX GAS OUTLET	TUBE AIR EXIT	BOX WALL TEMP	CHIM GAS EXIT	APPROX TOTAL WEIGHT LBS	DRAFT W/UNIT INCHES WATER	TOTAL HEAT GAIN BTU/HR	% HEAT LOSS RECOV
20.	58.	4.	0.625	1.563	1.250	4.69	4.69	5.00	909.	508.9	425.7	78.9	222.0	141.8	8.9	0.0502	2043.	17.6
20.	58.	4.	0.625	1.563	1.563	4.69	4.69	5.94	909.	508.9	424.5	78.8	222.5	141.4	9.2	0.0501	2073.	17.9
20.	58.	4.	0.750	1.500	0.938	4.50	4.50	4.31	*** INSUFFICIENT DRAFT ***									
20.	58.	4.	0.750	1.500	1.125	4.50	4.50	4.88	*** INSUFFICIENT DRAFT ***									
20.	58.	4.	0.750	1.500	1.500	4.50	4.50	6.00	*** INSUFFICIENT DRAFT ***									
20.	58.	4.	0.750	1.500	1.875	4.50	4.50	7.13	*** INSUFFICIENT DRAFT ***									
20.	58.	4.	0.750	1.688	0.938	5.06	5.06	4.31	1013.	508.9	418.8	80.0	209.9	139.7	9.7	0.0503	2212.	19.1
20.	58.	4.	0.750	1.688	1.125	5.06	5.06	4.88	1013.	508.9	417.9	79.9	209.9	139.4	9.9	0.0493	2232.	19.2
20.	58.	4.	0.750	1.688	1.500	5.06	5.06	6.00	*** INSUFFICIENT DRAFT ***									
20.	58.	4.	0.750	1.688	1.875	5.06	5.06	7.13	*** INSUFFICIENT DRAFT ***									
20.	58.	4.	0.750	1.875	0.938	5.63	5.63	4.31	758.	508.9	422.4	79.5	190.4	140.8	10.3	0.0504	2124.	18.3
20.	58.	4.	0.750	1.875	1.125	5.63	5.63	4.88	759.	508.9	421.7	79.5	191.1	140.6	10.5	0.0499	2141.	18.5
20.	58.	4.	0.750	1.875	1.500	5.63	5.63	6.00	759.	508.9	420.0	79.4	192.3	140.1	10.9	0.0492	2182.	18.8
20.	58.	4.	0.750	1.875	1.875	5.63	5.63	7.13	759.	508.9	418.4	79.4	193.4	139.6	11.4	0.0492	2221.	19.1
20.	58.	4.	0.875	1.531	1.094	4.59	4.59	5.03	*** INSUFFICIENT DRAFT ***									
20.	58.	4.	0.875	1.531	1.313	4.59	4.59	5.69	*** INSUFFICIENT DRAFT ***									
20.	58.	4.	0.875	1.531	1.750	4.59	4.59	7.00	*** INSUFFICIENT DRAFT ***									
20.	58.	4.	0.875	1.531	2.188	4.59	4.59	8.31	*** INSUFFICIENT DRAFT ***									
20.	58.	4.	0.875	1.750	1.094	5.25	5.25	5.03	*** INSUFFICIENT DRAFT ***									
20.	58.	4.	0.875	1.750	1.313	5.25	5.25	5.69	*** INSUFFICIENT DRAFT ***									
20.	58.	4.	0.875	1.750	1.750	5.25	5.25	7.00	*** INSUFFICIENT DRAFT ***									
20.	58.	4.	0.875	1.750	2.188	5.25	5.25	8.31	*** INSUFFICIENT DRAFT ***									
20.	58.	4.	0.875	1.969	1.094	5.91	5.91	5.03	*** INSUFFICIENT DRAFT ***									
20.	58.	4.	0.875	1.969	1.313	5.91	5.91	5.69	*** INSUFFICIENT DRAFT ***									
20.	58.	4.	0.875	1.969	1.750	5.91	5.91	7.00	*** INSUFFICIENT DRAFT ***									
20.	58.	4.	0.875	1.969	2.188	5.91	5.91	8.31	*** INSUFFICIENT DRAFT ***									
20.	58.	4.	1.000	1.750	1.250	5.25	5.25	5.75	*** INSUFFICIENT DRAFT ***									

Fig. 6.1c Output Sample (4 x 3 tube array)

Fig 6.1 continued Example of Initial Execution Mode Output for 4 x 3 Tube Arrays

```
A S S U M E D   C O N D I T I O N S

     NUMBER OF TUBES: 15                    |0 0 0|
                                            |0 0 0|
     TUBE CONFIGURATION                     |0 0 0|
                                            |0 0 0|
TUBES: STEEL HEAT EXCHANGER TYPE            |0 0 0|
CASING: CONSTRUCTED OF 16 GA STEEL
FUEL: NATURAL GAS
BAROMETRIC PRESSURE: 28.86 INCHES MERCURY
OUTSIDE AIR TEMPERATURE: 60DEGREES FAHRENHEIT
FURNACE ROOM TEMPERATURE: 69 DEGREES FAHRENHEIT
FURNACE STEADY-STATE EFFICIENCY: 80%
FURNACE EXIT TEMPERATURE:  622 DEGREES FAHRENHEIT
GAS MASS FLOW RATE(POUND-MASS/HR): APPROXIMATELY 1.6000 TIMES THE FURNACE INPUT
AIR FLOW THROUGH TUBE BANK: 160 CFM
CHIMNEY CONSTRUCTION: MASONRY TYPE WITH LINER

        K E Y   N O T E S
        *****************

FOR ALL CHIMNEY HEIGHTS, 4,5,6 INCH DIA FLUE PIPES CORRESPOND TO 5,6,7 INCH EQUIVALENT CHIMNEY DIAMETERS.
LATERAL BREECHING OF 2 FEET CORRESPONDS TO CHIMNEY HEIGHTS OF 15 AND 20 FEET.
LATERAL BREECHING OF 3 FEET CORRESPONDS TO CHIMNEY HEIGHTS OF 25 AND 30 FEET.
LATERAL BREECHING OF 4 FEET CORRESPONDS TO A CHIMNEY HEIGHT OF 35 FEET.
```

Fig. 6.2a Assumed Conditions and Key Notes

Fig. 6.2 Example of Initial Execution Mode Output for 5 x 3 Tube Arrays

MASON CHIM HEIGHT FEET	FURN INPUT *1000 BTU/HR	FLUE PIPE DIAM INCH	TUBE TUBE O.D. INCHES	BANK ARRAY TRANS SPCG INCHES	LONGIT SPCG INCHES	BOX DIMENSIONS 16 GA WIDTH INCHES	SHEET DEPTH INCHES	STEEL LENGTH INCHES	GAS MAX RE NO.	ALL BOX GAS INLET	TEMPS BOX GAS OUTLET	DEGREES TUBE AIR EXIT	FAHRENHEIT BOX WALL TEMP	CHIM GAS EXIT	APPROX TOTAL WEIGHT LBS	DRAFT W/UNIT INCHES WATER	TOTAL HEAT GAIN BTU/HR	% HEAT LOSS RECOV
30.	106.	5.	1.000	2.500	1.250	7.50	7.50	7.00	1038.	508.9	425.4	87.1	193.3	140.4	15.3	0.0714	3747.	17.7
30.	106.	5.	1.000	2.500	1.500	7.50	7.50	8.00	1039.	508.9	424.5	87.0	193.9	140.1	15.8	0.0705	3789.	17.9
30.	106.	5.	1.000	2.500	2.000	7.50	7.50	10.00	1039.	508.9	422.5	87.1	195.1	139.4	16.8	0.0694	3875.	18.3
30.	106.	5.	1.000	2.500	2.500	7.50	7.50	12.00	1040.	508.9	420.6	87.0	195.9	138.8	17.9	0.0689	3963.	18.7
30.	106.	5.	1.250	2.188	1.563	6.56	6.56	8.75	2386.	508.9	410.9	90.1	224.1	135.4	15.4	0.0616	4394.	20.7
30.	106.	5.	1.250	2.188	1.875	6.56	6.56	10.00	2387.	508.9	409.7	90.0	222.7	134.9	16.0	0.0575	4447.	21.0
30.	106.	5.	1.250	2.188	2.500	6.56	6.56	12.50	2389.	508.9	407.4	89.9	220.8	134.1	17.1	0.0532	4551.	21.5
30.	106.	5.	1.250	2.500	1.563	7.50	7.50	8.75	1564.	508.9	414.2	89.3	199.2	136.5	17.2	0.0662	4248.	20.0
30.	106.	5.	1.250	2.500	1.875	7.50	7.50	10.00	1565.	508.9	413.0	89.3	199.0	136.1	17.8	0.0646	4299.	20.3
30.	106.	5.	1.250	2.500	2.500	7.50	7.50	12.50	1566.	508.9	410.7	89.2	198.7	135.3	19.2	0.0624	4401.	20.8
30.	167.	6.	0.875	2.188	1.313	6.56	6.56	7.00	1863.	503.9	440.2	90.7	253.2	178.7	13.0	0.0744	4504.	13.5
30.	167.	6.	0.875	2.188	1.750	6.56	6.56	8.75	1864.	503.9	438.7	90.6	252.6	178.1	13.8	0.0705	4611.	13.8
30.	167.	6.	0.875	2.188	2.188	6.56	6.56	10.50	1865.	503.9	437.2	90.5	252.2	177.4	14.6	0.0705	4715.	14.1
30.	167.	6.	1.000	2.250	1.250	6.75	6.75	7.00	2176.	503.9	436.9	91.8	250.7	177.3	13.8	0.0734	4737.	14.2
30.	167.	6.	1.000	2.250	1.500	6.75	6.75	8.00	2177.	503.9	436.1	91.8	249.9	176.9	14.2	0.0702	4795.	14.4
30.	167.	6.	1.000	2.250	2.000	6.75	6.75	10.00	2178.	503.9	434.4	91.8	248.9	176.2	15.2	0.0663	4913.	14.7
30.	167.	6.	1.000	2.250	2.500	6.75	6.75	12.00	2179.	503.9	432.7	91.7	248.2	175.4	16.1	0.0663	5028.	15.1
30.	167.	6.	1.000	2.500	1.250	7.50	7.50	7.00	1631.	503.9	438.0	91.5	228.2	177.8	15.0	0.0784	4659.	14.0
30.	167.	6.	1.000	2.500	1.500	7.50	7.50	8.00	1632.	503.9	437.2	91.4	228.2	177.4	15.5	0.0766	4716.	14.1
30.	167.	6.	1.000	2.500	2.000	7.50	7.50	10.00	1633.	503.9	435.5	91.3	228.6	176.7	16.5	0.0742	4833.	14.5
30.	167.	6.	1.000	2.500	2.500	7.50	7.50	12.00	1634.	503.9	434.0	91.3	228.9	176.0	17.6	0.0740	4941.	14.8
30.	167.	6.	1.250	2.188	1.563	6.56	6.56	8.75	3741.	503.9	429.2	94.1	263.3	173.8	15.1	0.0510	5279.	15.8
30.	167.	6.	1.250	2.188	1.875	6.56	6.56	10.00	***INSUFFICIENT DRAFT***									
30.	167.	6.	1.250	2.188	2.500	6.56	6.56	12.50	***INSUFFICIENT DRAFT***									
30.	167.	6.	1.250	2.500	1.563	7.50	7.50	8.75	2454.	503.9	430.5	93.7	234.5	174.4	16.9	0.0712	5187.	15.5
30.	167.	6.	1.250	2.500	1.875	7.50	7.50	10.00	2455.	503.9	429.6	93.6	233.6	174.0	17.5	0.0677	5251.	15.7
30.	167.	6.	1.250	2.500	2.500	7.50	7.50	12.50	2456.	503.9	427.5	93.5	232.5	173.1	18.9	0.0641	5396.	16.2

Fig. 6.2b Output Sample (5 x 3 tube array)

Fig. 6.2 continued Example of Initial Execution Mode Output for 5 x 3 Tube Arrays

			TUBE BANK ARRAY			BOX DIMENSIONS 16 GA SHEET STEEL				ALL TEMPS DEGREES FAHRENHEIT								
MASON CHIM HEIGHT FEET	FURN INPUT *1000 BTU/HR	FLUE PIPE DIAM INCH	TUBE O.D. INCHES	TRANS SPCG INCHES	LONGIT SPCG INCHES	WIDTH INCHES	DEPTH INCHES	LENGTH INCHES	GAS MAX RE NO.	BOX GAS INLET	BOX GAS OUTLET	TUBE AIR EXIT	BOX WALL TEMP	CHIM GAS EXIT	APPROX TOTAL WEIGHT LBS	DRAFT W/UNIT INCHES WATER	TOTAL HEAT GAIN BTU/HR	% HEAT LOSS RECOV
30.	106.	5.	0.750	1.875	1.125	5.63	5.63	6.00	1381.	508.9	432.2	85.8	242.3	142.7	11.2	0.0730	3446.	16.3
30.	106.	5.	0.750	1.875	1.500	5.63	5.63	7.50	1382.	508.9	430.5	85.7	241.9	142.2	11.7	0.0703	3519.	16.6
30.	106.	5.	0.750	1.875	1.875	5.63	5.63	9.00	1383.	508.9	428.9	85.7	241.7	141.6	12.3	0.0701	3590.	16.9
30.	106.	5.	0.875	1.750	1.094	5.25	5.25	6.13	2226.	508.9	424.3	87.6	261.5	140.1	11.1	0.0629	3794.	17.9
30.	106.	5.	0.875	1.750	1.313	5.25	5.25	7.00	2227.	508.9	423.5	87.6	259.7	139.8	11.4	0.0576	3834.	18.1
30.	106.	5.	0.875	1.750	1.750	5.25	5.25	8.75	2228.	508.9	421.6	87.5	257.3	139.1	12.0	0.0517	3918.	18.5
30.	106.	5.	0.875	1.750	2.188	5.25	5.25	10.50	2230.	508.9	419.8	87.5	255.6	138.5	12.7	0.0519	3998.	18.9
30.	106.	5.	0.875	1.969	1.094	5.91	5.91	6.13	1581.	508.9	426.9	87.0	235.7	140.9	12.0	0.0725	3682.	17.4
30.	106.	5.	0.875	1.969	1.313	5.91	5.91	7.00	1582.	508.9	425.9	87.0	235.2	140.6	12.3	0.0701	3726.	17.6
30.	106.	5.	0.875	1.969	1.750	5.91	5.91	8.75	1583.	508.9	424.1	86.9	234.6	140.0	13.1	0.0669	3807.	18.0
30.	106.	5.	0.875	1.969	2.188	5.91	5.91	10.50	1584.	508.9	422.3	86.9	234.1	139.3	13.8	0.0666	3888.	18.3
30.	106.	5.	0.875	2.188	1.094	6.56	6.56	6.13	1185.	508.9	428.9	86.4	214.4	141.6	12.9	0.0750	3590.	16.9
30.	106.	5.	0.875	2.188	1.313	6.56	6.56	7.00	1186.	508.9	428.1	86.5	214.7	141.3	13.3	0.0738	3627.	17.1
30.	106.	5.	0.875	2.188	1.750	6.56	6.56	8.75	1186.	508.9	426.2	86.4	215.3	140.7	14.1	0.0722	3710.	17.5
30.	106.	5.	0.875	2.188	2.188	6.56	6.56	10.50	1187.	508.9	424.6	86.3	215.8	140.2	14.9	0.0718	3782.	17.8
30.	106.	5.	1.000	1.750	1.250	5.25	5.25	7.00	*** INSUFFICIENT DRAFT ***									
30.	106.	5.	1.000	1.750	1.500	5.25	5.25	8.00	*** INSUFFICIENT DRAFT ***									
30.	106.	5.	1.000	1.750	2.000	5.25	5.25	10.00	*** INSUFFICIENT DRAFT ***									
30.	106.	5.	1.000	1.750	2.500	5.25	5.25	12.00	*** INSUFFICIENT DRAFT ***									
30.	106.	5.	1.000	2.000	1.250	6.00	6.00	7.00	1950.	508.9	420.6	88.2	235.8	138.7	12.9	0.0685	3964.	18.7
30.	106.	5.	1.000	2.000	1.500	6.00	6.00	8.00	1951.	508.9	419.6	88.2	234.8	138.4	13.3	0.0651	4008.	18.9
30.	106.	5.	1.000	2.000	2.000	6.00	6.00	10.00	1952.	508.9	417.6	88.2	233.5	137.7	14.1	0.0605	4094.	19.3
30.	106.	5.	1.000	2.000	2.500	6.00	6.00	12.00	1954.	508.9	415.5	88.1	232.6	137.0	15.0	0.0603	4187.	19.8
30.	106.	5.	1.000	2.250	1.250	6.75	6.75	7.00	1386.	508.9	423.1	87.6	212.4	139.6	14.1	0.0718	3849.	18.2
30.	106.	5.	1.000	2.250	1.500	6.75	6.75	8.00	1386.	508.9	422.2	87.6	212.4	139.3	14.5	0.0703	3892.	18.4
30.	106.	5.	1.000	2.250	2.000	6.75	6.75	10.00	1387.	508.9	420.2	87.5	212.5	138.6	15.5	0.0682	3979.	18.8
30.	106.	5.	1.000	2.250	2.500	6.75	6.75	12.00	1388.	508.9	418.2	87.5	212.6	137.9	16.4	0.0678	4068.	19.2

Fig. 6.2c Output Sample (5 x 3 tube array)

Fig. 6.2 continued Example of Initial Execution Mode Output for 5 x 3 Tube Arrays

Figures 6.3a-b show a typical design as it appears in the computer output. Information about the reclaimer, such as various heat-transfer coefficients, and the heat gain by the reclaimer casing are included.

It is useful to evaluate the performance of the heat reclaimer under normal operating conditions. Since many urban North American homes have chimney heights above 20 feet and flue pipe diameters greater than 4 inches, the 5 x 3 reclaimer design was used to study the effects of changing ambient temperature and furnace steady-state efficiency on six heating systems (furnace inputs ranging from 101,000 to 169,000 Btu/h). Figures 6.4a-f, 6.5a-f, and 6.6a-f show the heat recovered, the per cent of heat loss recovered, and the per cent of fuel saved respectively, for the reclaimer under typical conditions. Inefficient oil-fired furnaces proved to be the best sources of waste heat.

Varying the furnace steady-state efficiency and/or the ambient temperature affected the system updraft and heat recovery. It was assumed throughout this thesis that the effect of additional air infiltration into the draft regulator produces negligible changes in the system pressure drop or heat transfer.

The reclaimer is not resricted to heating the near surroundings. In fact, a heating duct attached to the reclaimer could be connected to some other portion of the house, thus providing localized space-heat. For most typical cases, however, the furnace exists in the basement and the heat reclaimer would add heat there.

The basement heat loss depends primarily on the size and construction of the basement, and the ambient temperature. Assuming the conditions in Fig. 6.7, for a basement in a two-story house (25-ft chimney and a 5 or 6-inch flue pipe), the following analysis describes

G E N E R A L   N O T E S

THE FURNACE IS A RESIDENTIAL, NATURAL DRAFT TYPE.

THE FURNACE ROOM TEMPERATURE IS 69F.

THE OUTSIDE AMBIENT TEMPERATURE IS 60F.

THE BAROMETRIC PRESSURE AT 1000 FEET ELEVATION IS 28.86 INCHES OF MERCURY.

THE CHIMNEY IS OF MASONRY CONSTRUCTION WITH LINER.

THE FLUE PIPE IS SINGLE WALL STEEL WITH VARIOUS ATTACHMENTS.

INFILTRATION THROUGH THE BAROMETRIC DRAFT DIVERTER IS ASSUMED NEGLIGIBLE.

THE DEVICE CASING IS CONSTRUCTED OF 16 GA CARBON STEEL.

THE TUBE BANK CONSISTS OF THIN-WALL, HEAT EXCHANGER TYPE STEEL TUBES.

Fig. 6.3a General Notes

Fig. 6.3 Example of Final Design Mode Output

```
H E A T   R E C O V E R Y   P E R F O R M A N C E
FUEL: NO. 2 FUEL OIL    FURNACE EXIT TEMPERATURE:  832F
FURNACE EFF:  80%    INPUT:  101000. BTU/HR    OUTPUT  80800. BTU/HR
CHIMNEY HEIGHT............... 25. FEET
INSIDE EQUIV. CHIMNEY DIA.... 6. INCHES
FLUE PIPE DIA................ 5. INCHES
FLUE PIPE LATERAL BREECHING.. 3. FEET
SYSTEM UPDRAFT W/O DEVICE.... 0.1374 INCHES WATER
GAS MASS FLOW RATE........... 125.2 POUND-MASS/HR
TUBE AIR VOLUME FLOW RATE.... 160. CFM
DEVICE CASING WIDTH..........  7.50 INCHES                    |0 0 0|
                                             TUBE             |0 0 0|
DEVICE CASING DEPTH..........  7.50 INCHES                    |0 0 0|
                                         CONFIGURATION        |0 0 0|
DEVICE CASING LENGTH......... 12.50 INCHES                    |0 0 0|
                                           15 TUBES
TUBE O.D..................... 1.250 INCHES
TUBE I.D..................... 1.150 INCHES
TRANSVERSE SPACING........... 2.500 INCHES
LONGITUDINAL SPACING......... 2.500 INCHES
GAS INLET TEMP: 738.9 F   BULK GAS TEMP: 646.5 F   GAS EXIT TEMP: 554.0 F
AIR INLET TEMP:  69.0 F   BULK AIR TEMP:  83.6 F   AIR EXIT TEMP:  98.2 F
DEVICE CASING WALL TEMPERATURE.......... 226.6 F  TUBE WALL TEMP: 264.1 F
TUBE INSIDE HEAT TRANSFER COEFFICIENT... 10.58 BTU/HR/SQFT/F
TUBE OUTSIDE HEAT TRANSFER COEFFICIENT..  4.59 BTU/HR/SQFT/F
TOTAL WEIGHT OF DEVICE.................. 19.2 POUNDS
CHIMNEY EXIT TEMPERATURE WITH DEVICE.... 167.1 F
MAXIMUM REYNOLDS NUMBER.................  1005.
THEORETICAL DRAFT WITH DEVICE........... 0.1186 INCHES WATER
UPDRAFT OF SYSTEM WITH DEVICE........... 0.0970 INCHES WATER
HEAT GAIN BY TUBE BANK:  5385.BTU/HR    GAIN BY EXPOSED CASING:  866.BTU/HR
TOTAL HEAT GAIN BY DEVICE:  6251.BTU/HR     30.9% OF HEAT LOSS RECOVERED
FUEL SAVED:  6.2%
```

Fig. 6.3b Output Sample

Fig. 6.3 continued Example of Final Design Mode Output

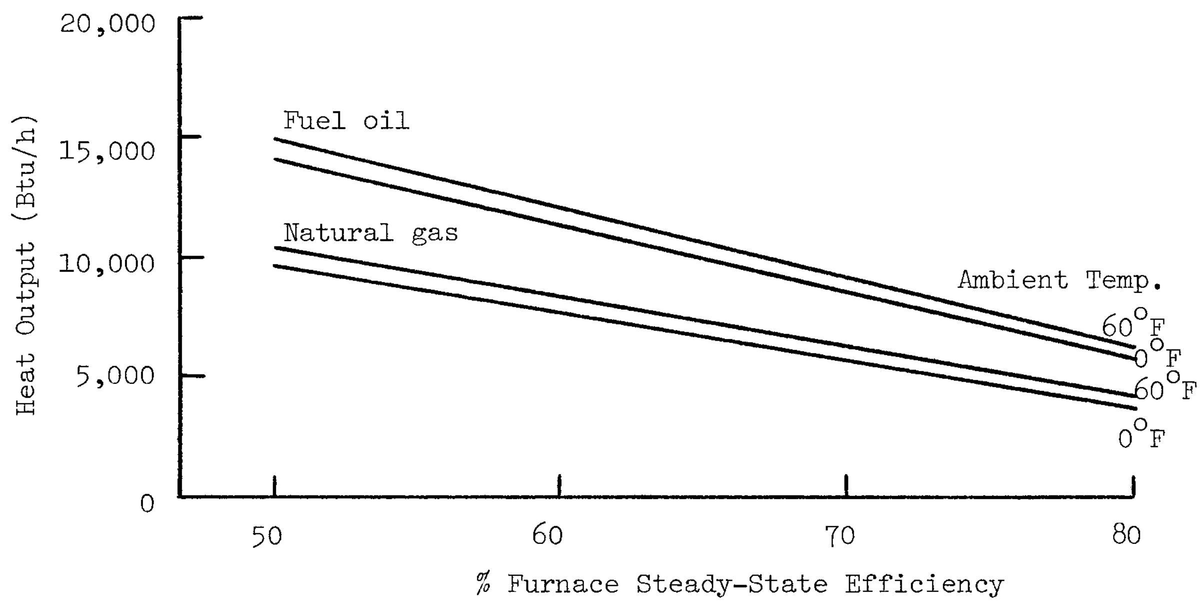


Fig. 6.4a 25-ft. chimney, 5-in. flue pipe

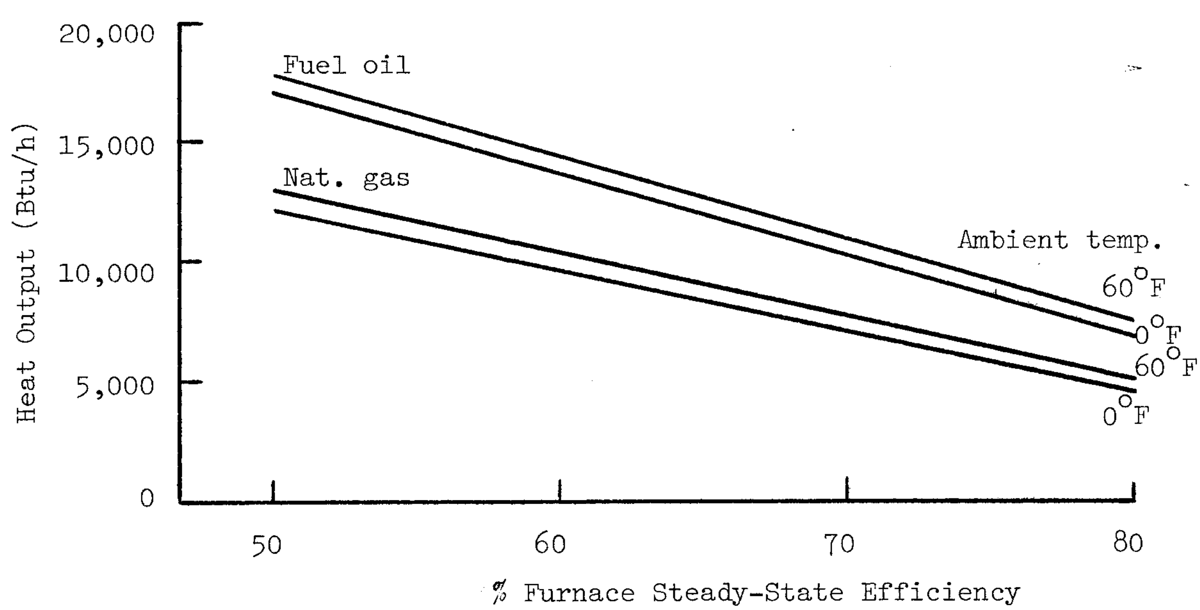


Fig. 6.4b 25-ft. chimney, 6-in. flue pipe

Fig. 6.4 Furnace Efficiency vs. Heat Output

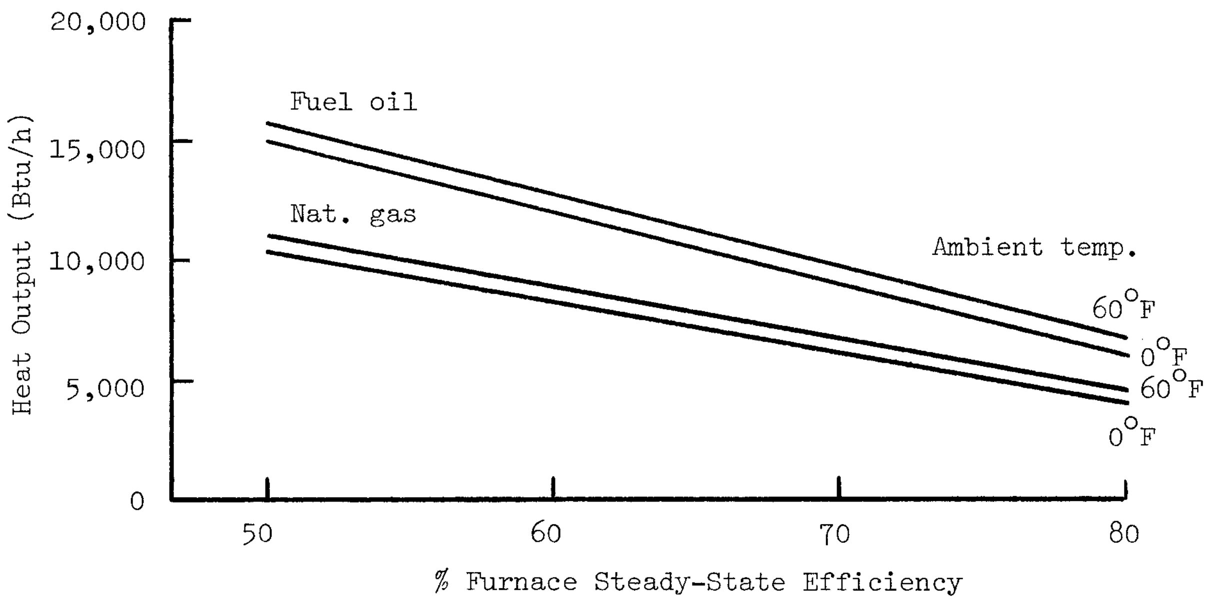


Fig. 6.4c 30-ft. chimney, 5-in. flue pipe

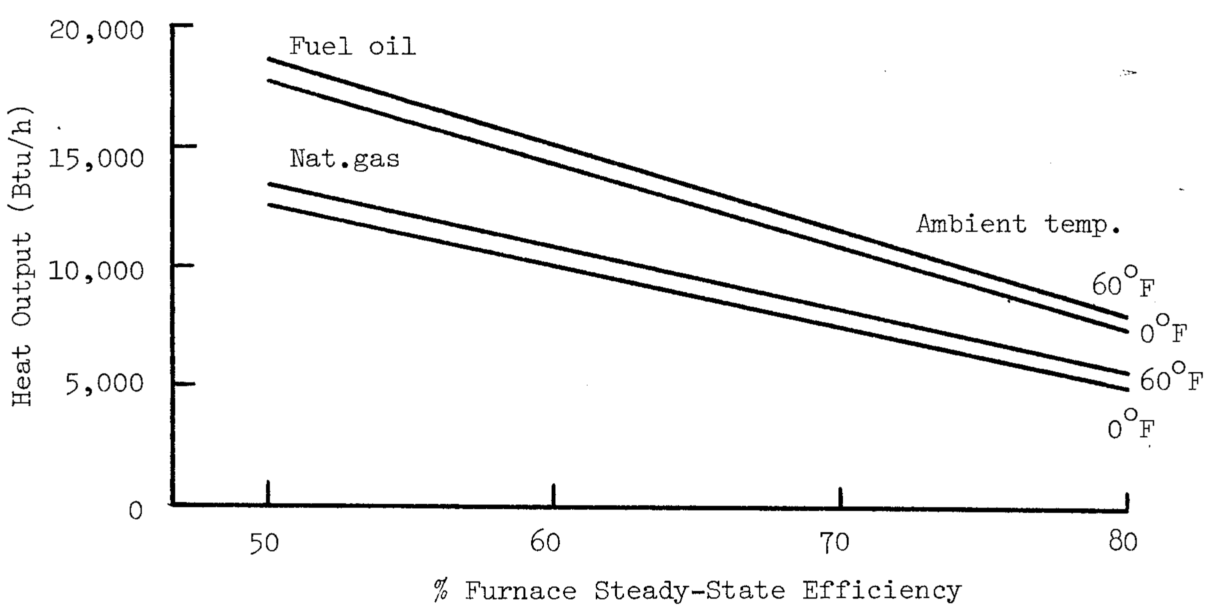


Fig. 6.4d 30-ft. chimney, 6-in. flue pipe

Fig. 6.4 continued Furnace Efficiency vs. Heat Output

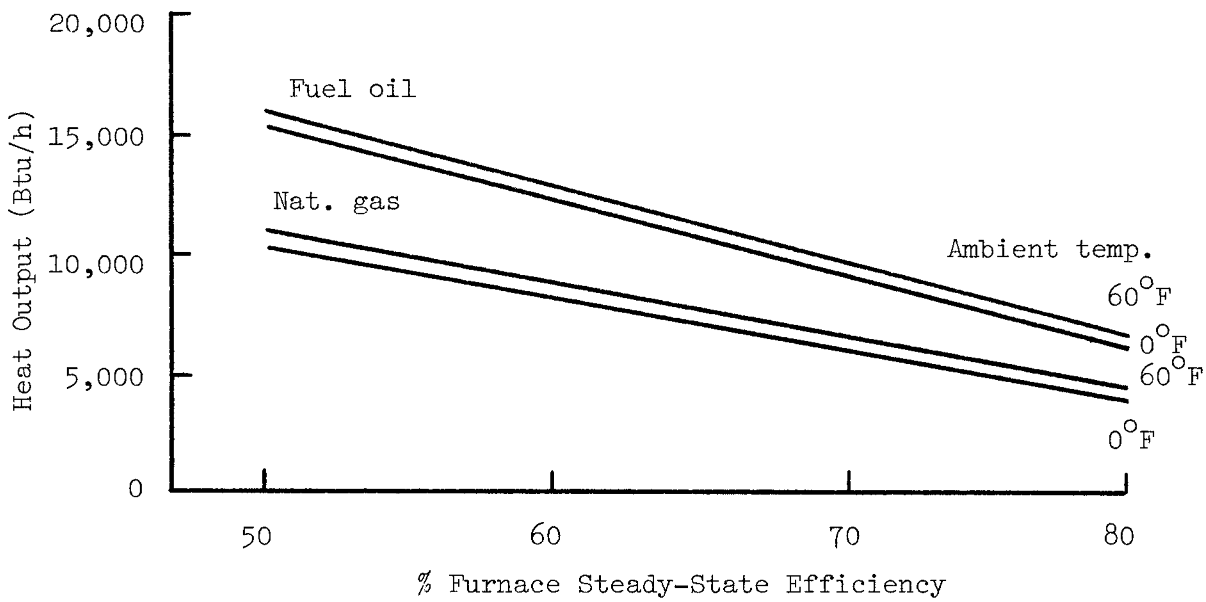


Fig. 6.4e 35-ft. chimney, 5-in. flue pipe

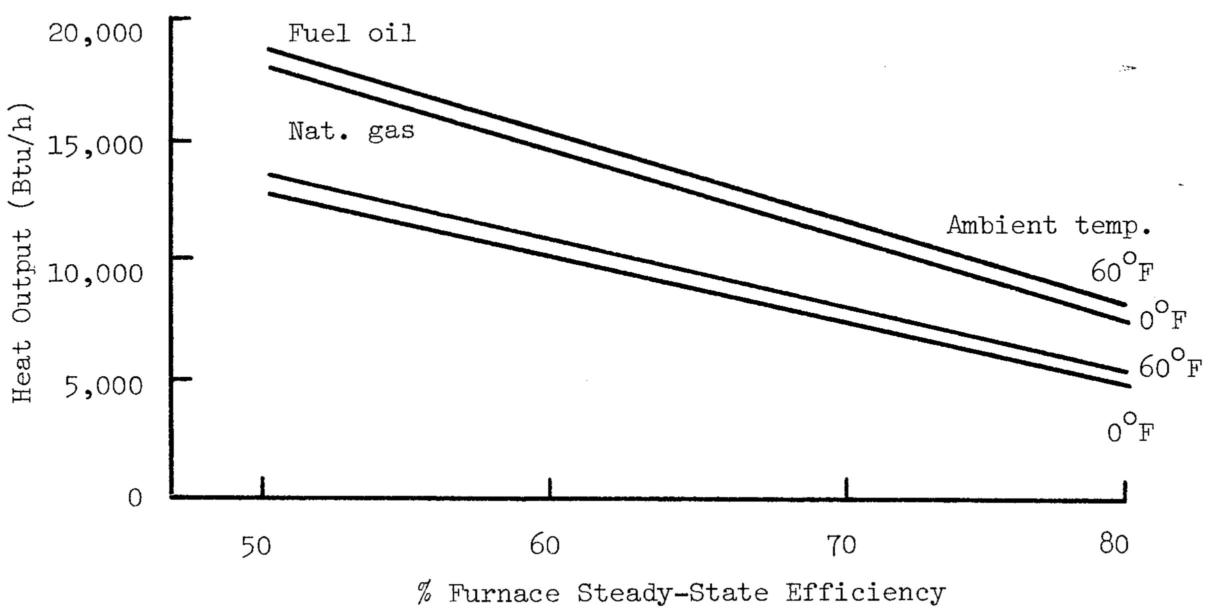


Fig. 6.4f 35-ft. chimney, 6-in. flue pipe

Fig. 6.4 continued Furnace Efficiency vs. Heat Output

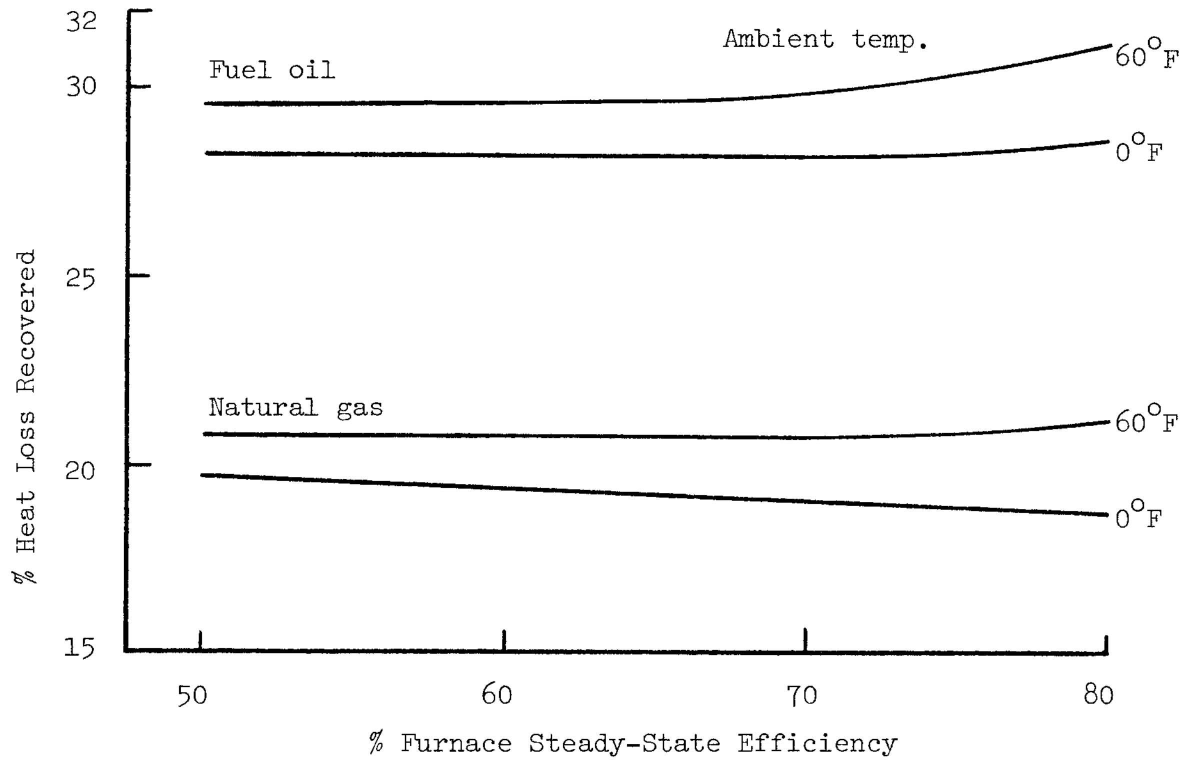


Fig. 6.5a 25-ft. chimney, 5-in. flue pipe

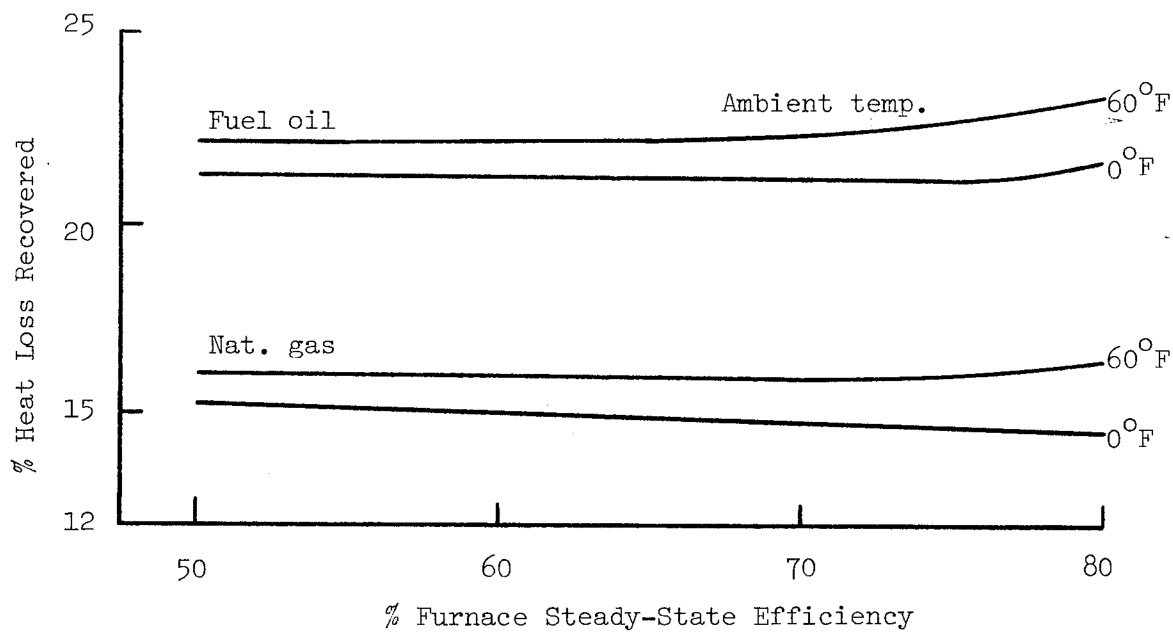


Fig. 6.5b 25-ft. chimney, 6-in. flue pipe

Fig. 6.5 Furnace Efficiency vs. Heat Recovery

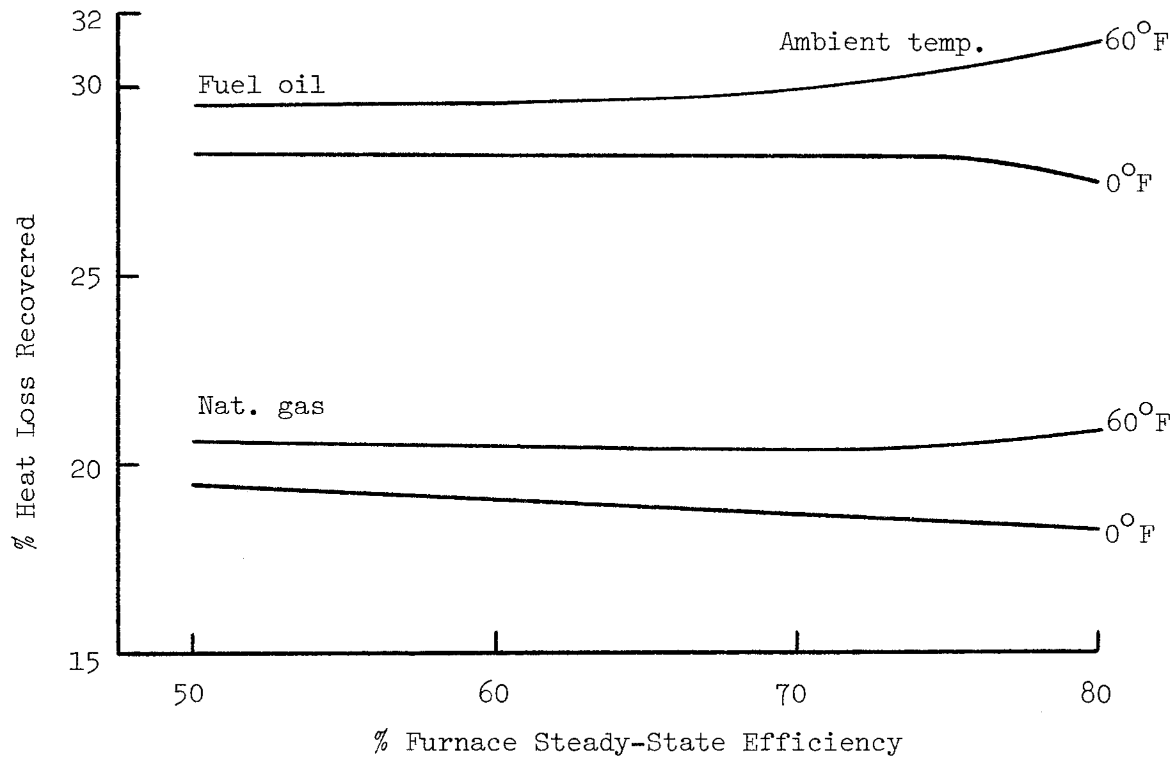


Fig. 6.5c 30-ft. chimney, 5-in. flue pipe

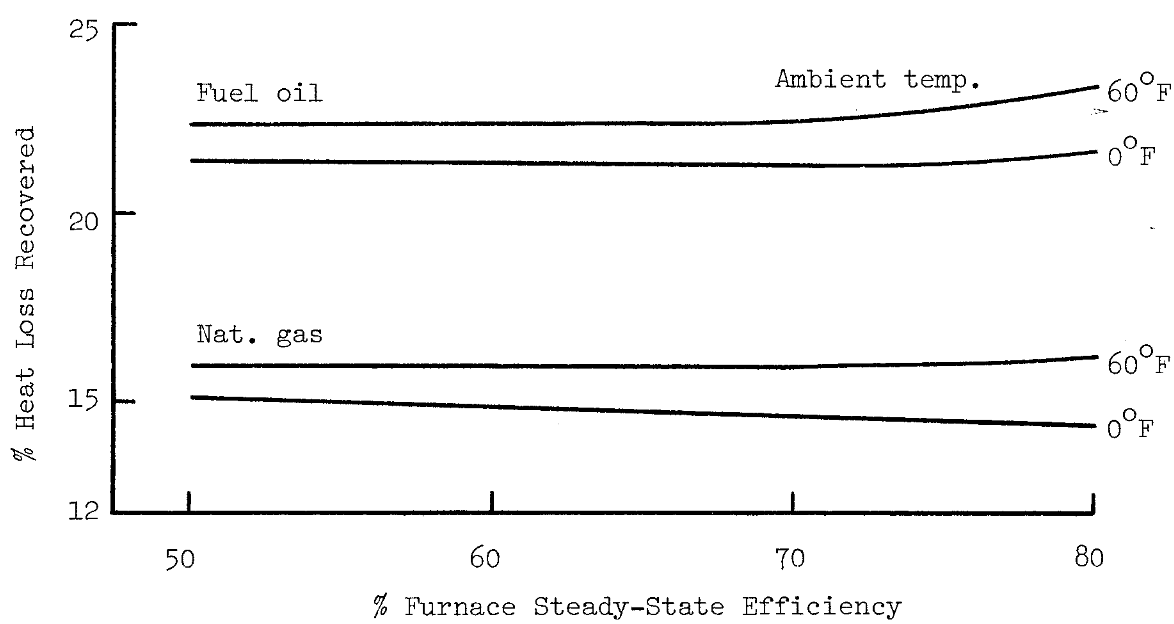


Fig. 6.5d 30-ft. chimney, 6-in. flue pipe

Fig. 6.5 continued Furnace Efficiency vs. Heat Recovery

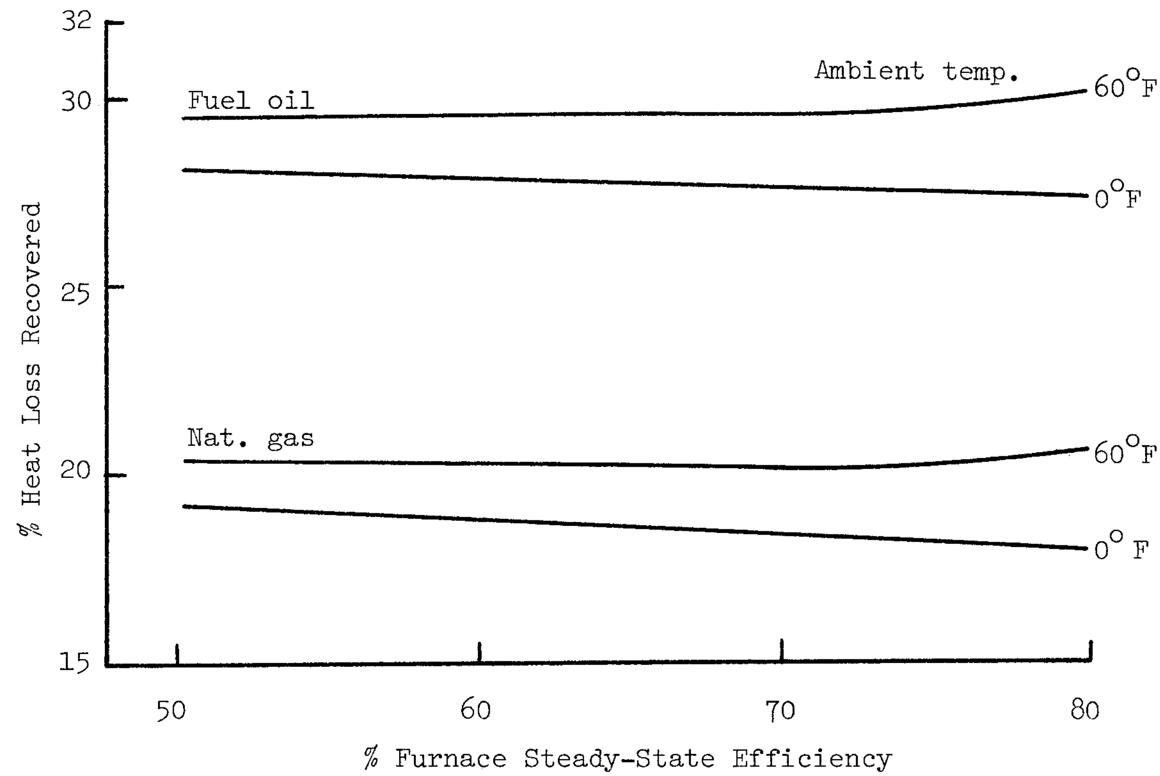


Fig. 6.5e 35-ft. chimney, 5-in. flue pipe

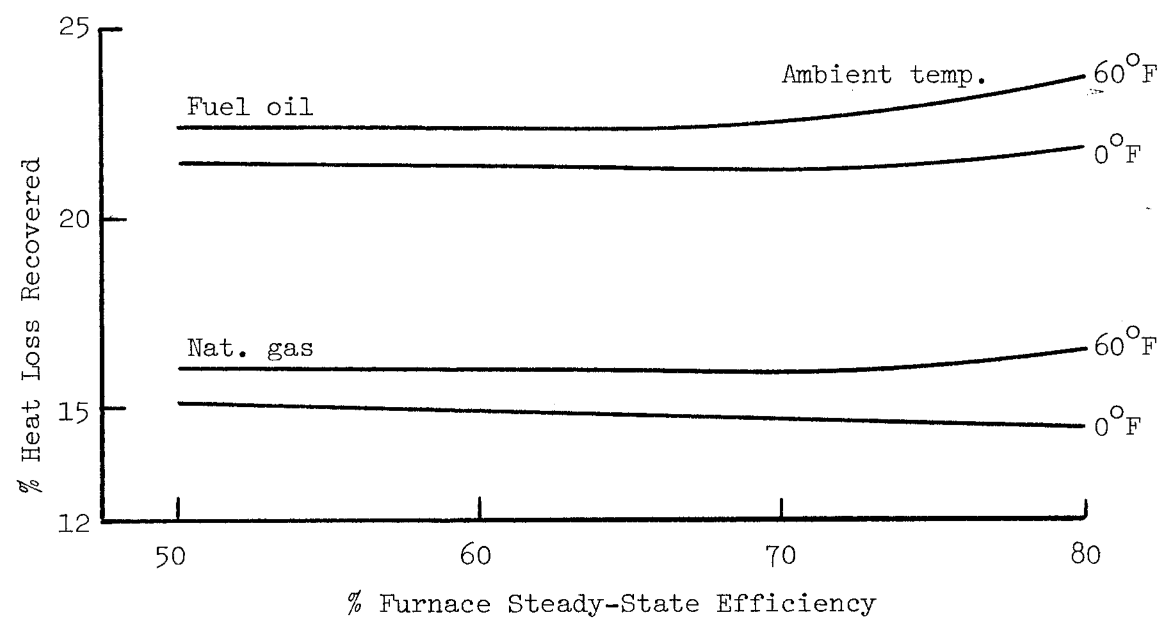


Fig. 6.5f 35-ft. chimney, 6-in. flue pipe

Fig. 6.5 continued Furnace Efficiency vs. Heat Recovery

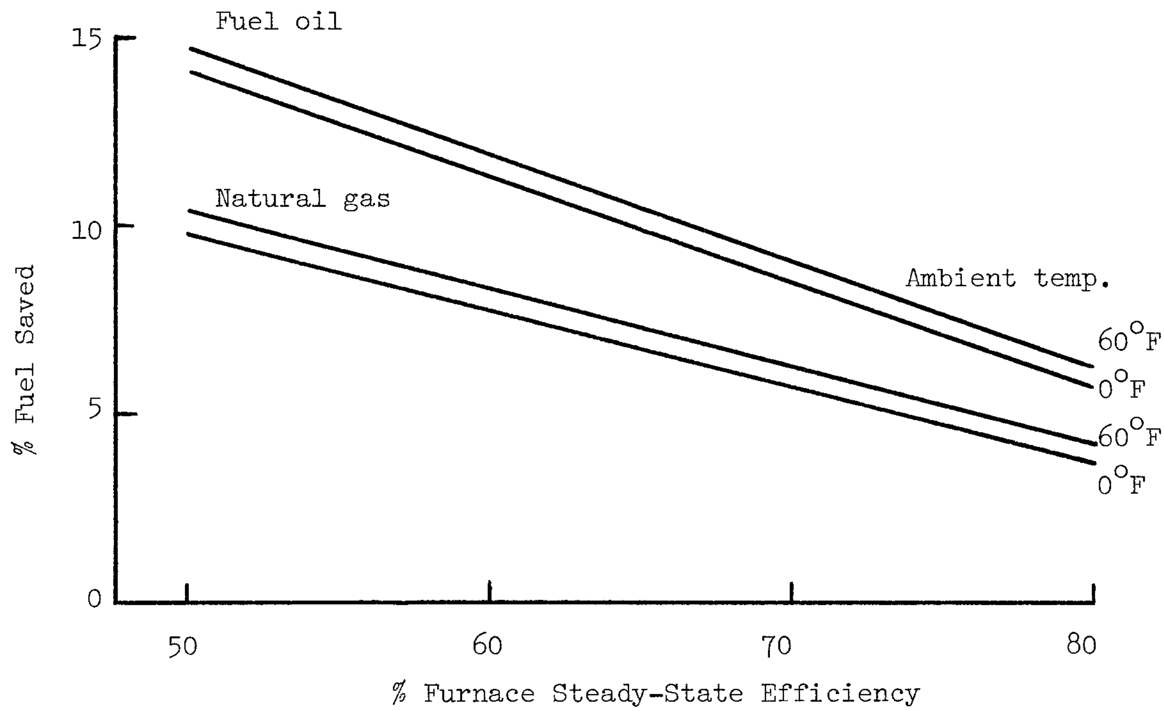


Fig. 6.6a 25-ft. chimney, 5-in. flue pipe

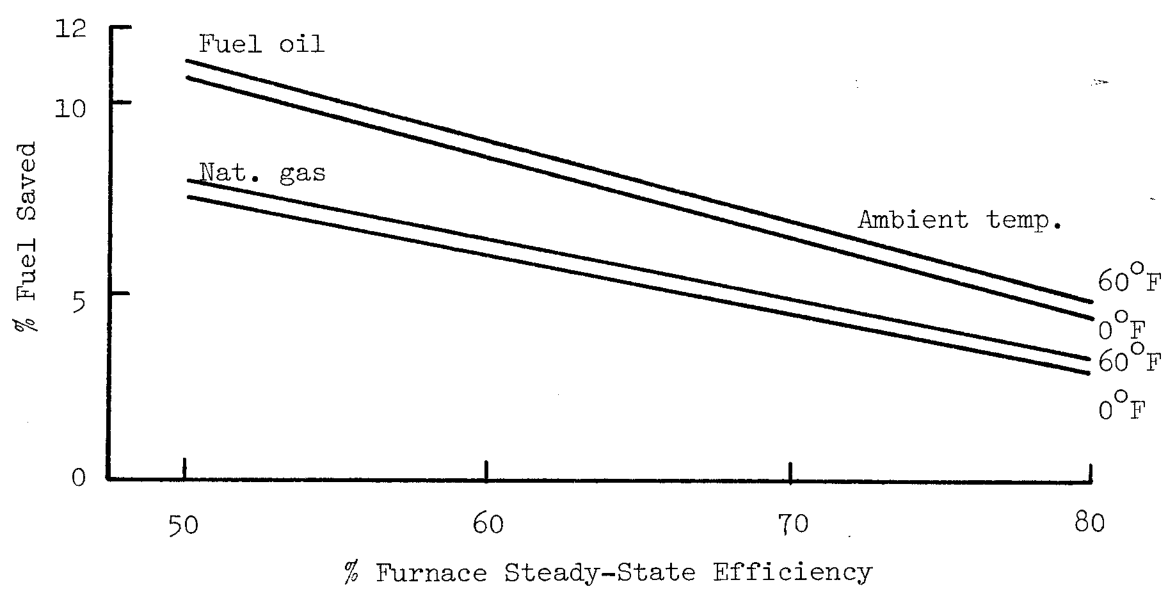


Fig. 6.6b 25-ft. chimney, 6-in. flue pipe

Fig. 6.6 Furnace Efficiency vs. Fuel Saved

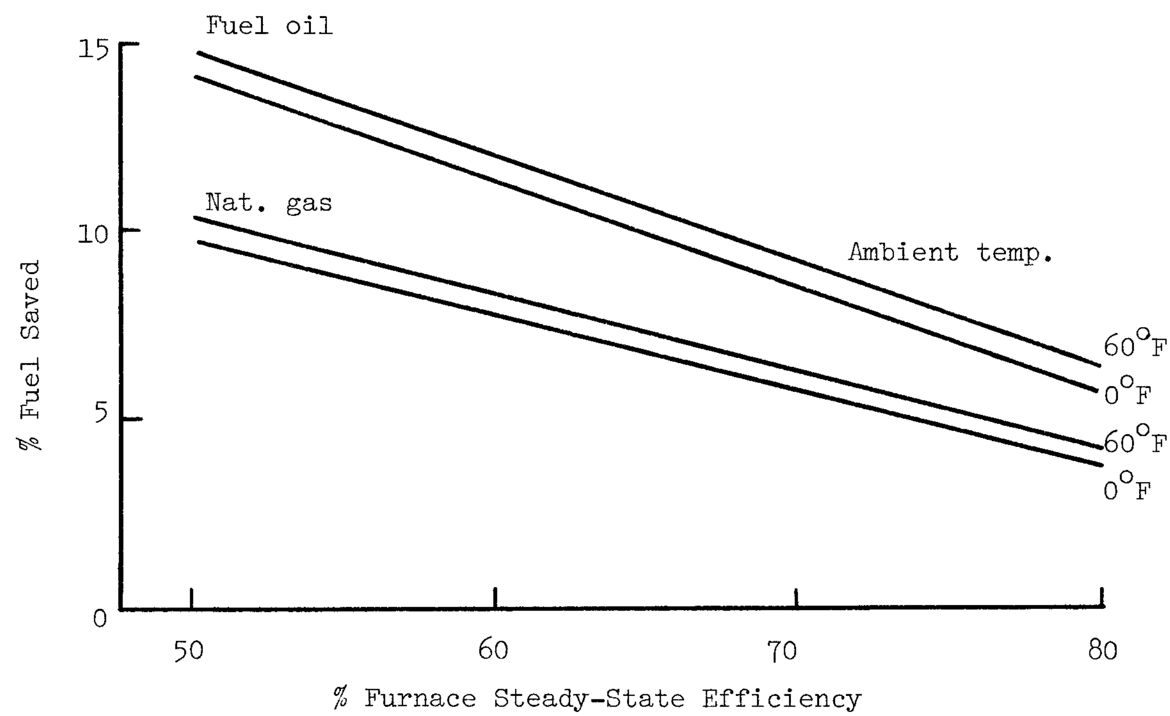


Fig. 6.6c 30-ft. chimney, 5-in. flue pipe

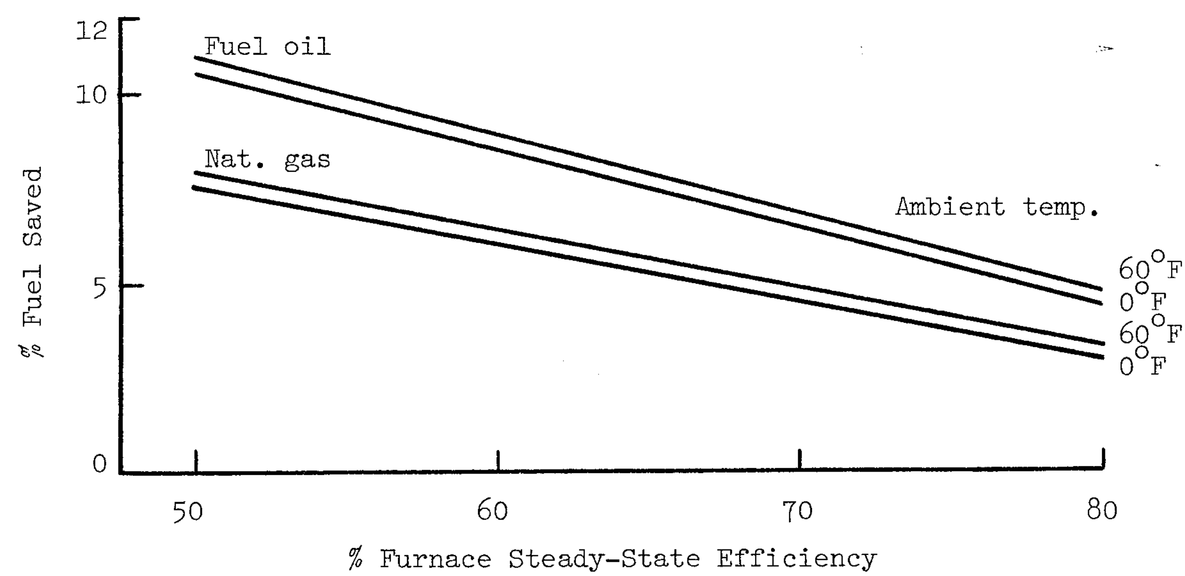


Fig. 6.6d 30-ft. chimney, 6-in. flue pipe

Fig. 6.6 continued Furnace Efficiency vs. Fuel Saved

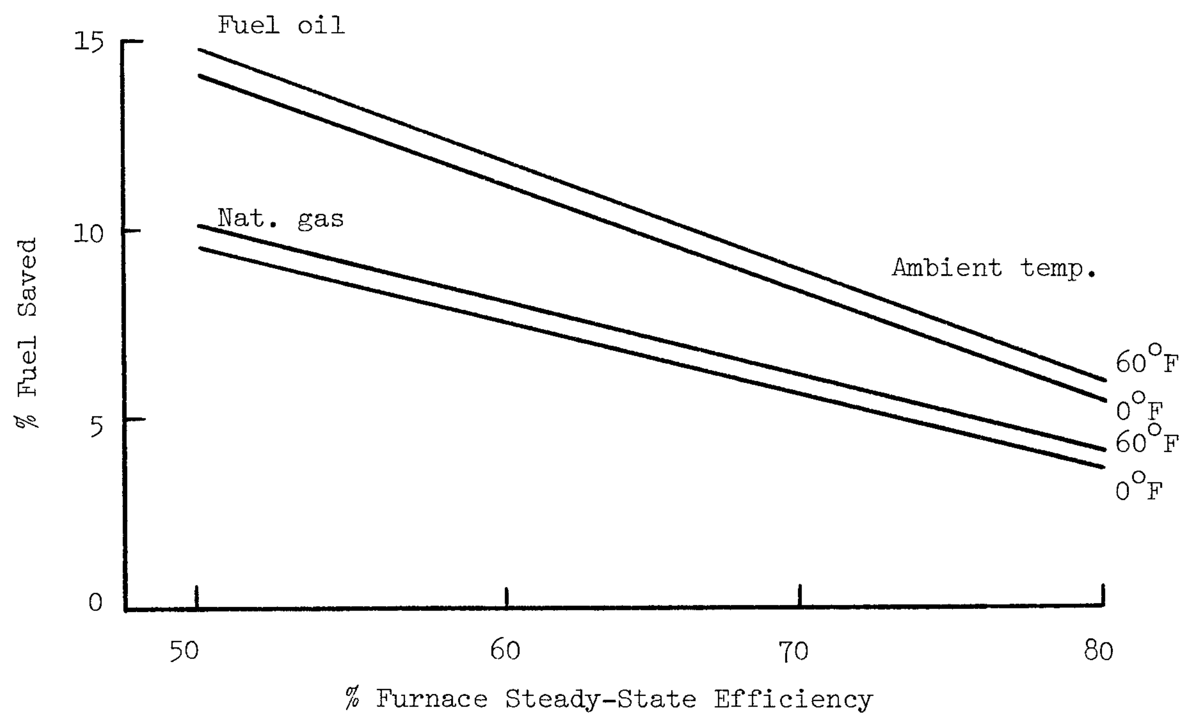


Fig. 6.6e 35-ft. chimney, 5-in. flue pipe

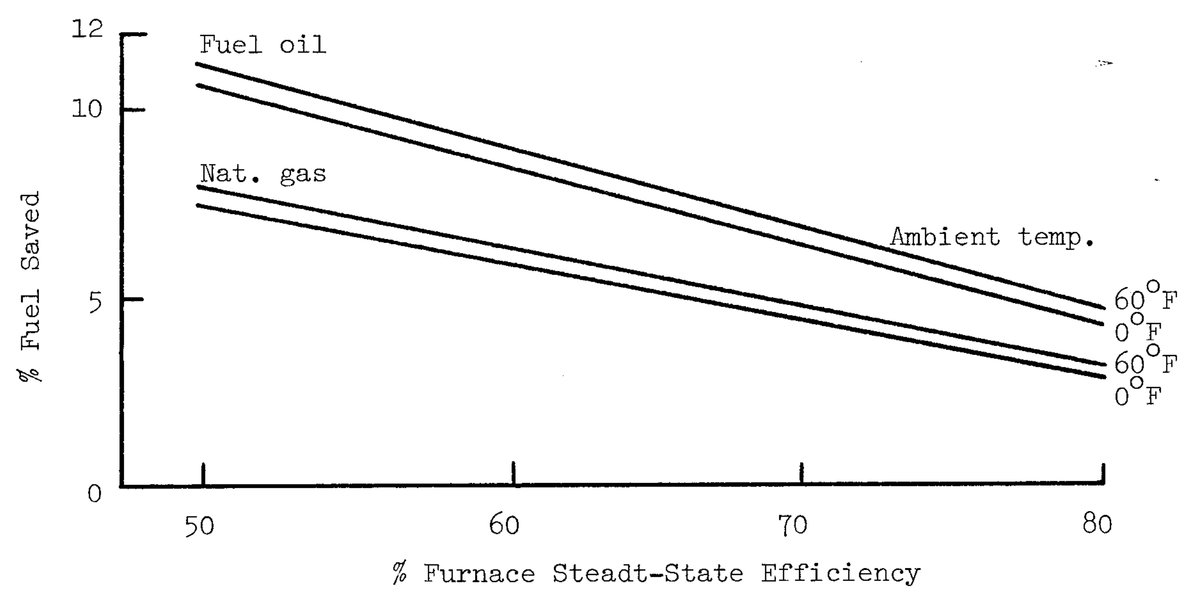


Fig. 6.6f 35-ft. chimney, 6-in. flue pipe

Fig. 6.6 continued Furnace Efficiency vs. Fuel Saved how much basement heat is needed. For the total heat loss in the basement (through the walls and floor),

$$Q_b = U_b A_b (\Delta T)_b , \qquad (6.1)$$

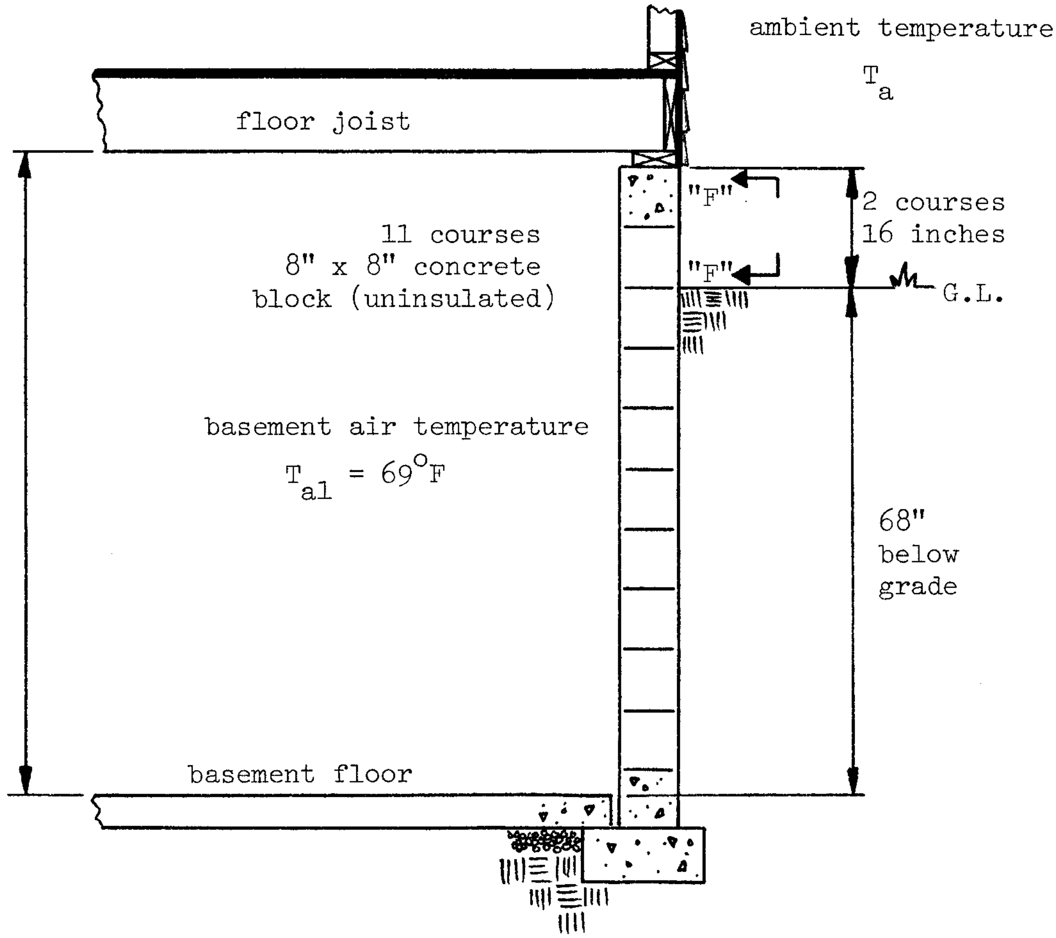
where U_b is the average overall heat-transfer coefficient, A_b the area of the walls and floor, and $(\Delta T)_b$ the temperature difference between the inside and outside (wall or floor). Basement heat loss analysis can be readily found [1] and is summarized in the following:

$$Q_b = (UA)_{bo}(T_{al} - T_a) + (UA)_{bi}(\Delta T)_{bi}, \qquad (6.2)$$

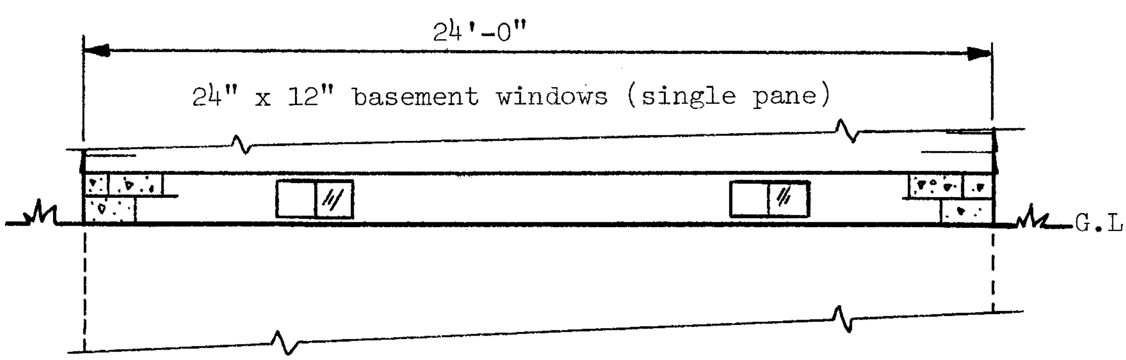
where the first and second terms on the right-hand side represent the heat loss above and below-grade, respectively. The loss above-grade consists of the loss which passes through the top two courses of uninsulated concrete block and basement windows. The average overall heat-transfer coefficient, based on the composite above-grade basement wall, was calculated to be 0.60 (Btu/h-ft^2 - $^\circ$F), and the amount of heat loss depends on the ambient temperature.

The loss below-grade consists of the heat transmitted through the floor and the walls (below grade). The overall heat-transfer coefficient for the basement floor [1] was found to be 0.029 (Btu/h-ft^2 - $^\circ$F). The heat loss through the walls is primarily determined by the number of concrete courses below grade, and the average overall heat-transfer coefficient was calculated to be 1.081 (Btu/h-ft^2 - $^\circ$F). The temperature change used in calculating the below-grade heat loss is the difference between the yearly mean average temperature and the temperature amplitude, which for Northeast Ohio is estimated at 49 $^\circ$F and 18 $^\circ$F, respectively [1].

The basement heat loss can now be expressed in the following



Typical Basement Wall Section



View "F - F" Typical Basement Side Elevation (reduced scale)

Fig. 6.7 Typical Basement of a Two-Story Home

Fig. 6.8 Furnace Efficiency vs. Per Cent Heat Supplied to Basement

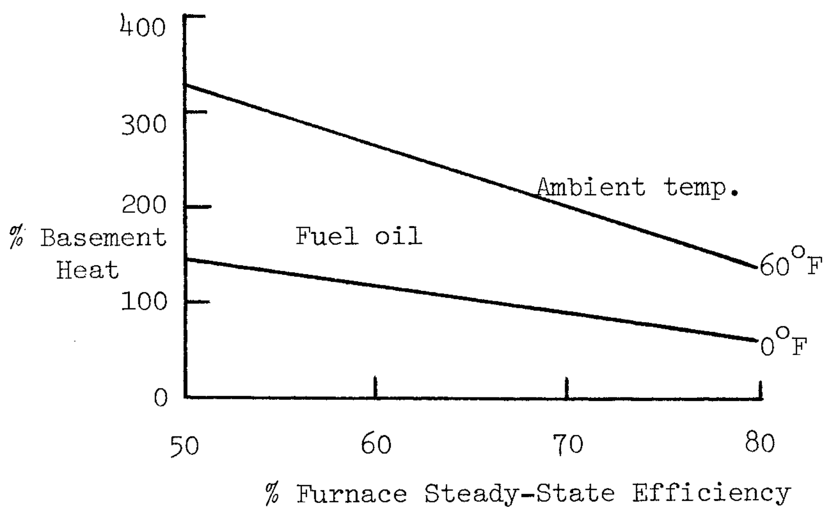


Fig. 6.8a 25-ft. chimney, 5-in. flue pipe

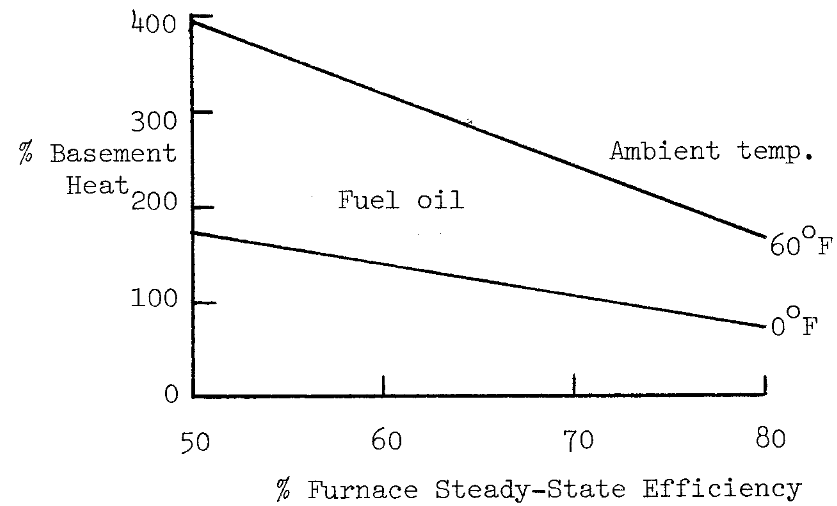


Fig. 6.8b 25-ft. chimney, 6-in. flue pipe

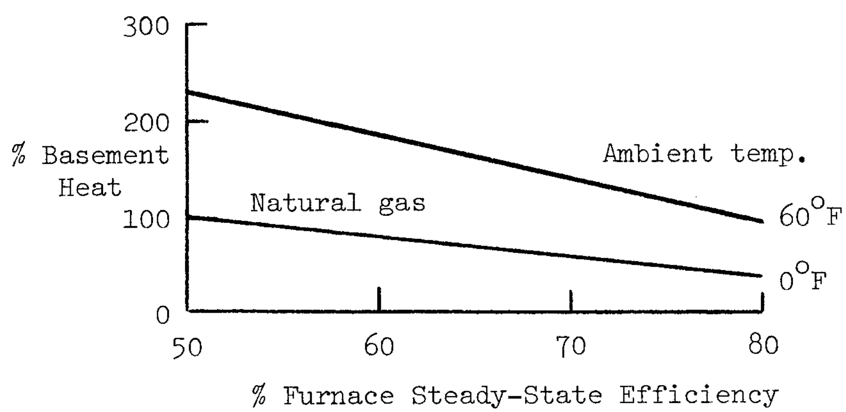


Fig. 6.8c 25-ft. chimney, 5-in. flue pipe

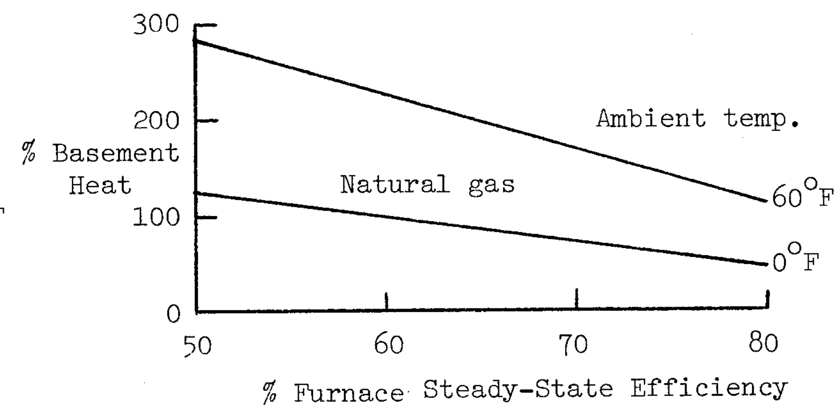


Fig. 6.8d 25-ft. chimney, 6-in. flue pipe

simplified equation:

$$Q_b = 87.7(69 - T_a) + 3500 \qquad \text{(Btu/h)}, \qquad (6.3)$$

where T_a is the ambient temperature in (^oF).

Figures 6.8a-d represent the per cent of basement heat supplied by the heat reclaimer as a function of furnace steady-state efficiency for typical two-story homes. They show that the heat reclaimer can supply all or a significant portion of the basement heat, even on very cold days.

The results clearly show that the greatest potential for heat recovery exists in older, two-story (or higher) homes that burn fuel oil. The main reason is that higher chimneys produce sufficient draft, and flue gases from oil-fired furnaces are generally about 200 ^oF warmer than those from natural gas furnaces.

CHAPTER VII

DISCUSSION AND CONCLUSION

Optimized designs for residential heat reclaimers have been presented. The results have shown the importance of in-line tube bank arrangements for heat recovery in natural draft, residential heating systems, using natural gas or oil-fired equipment.

It was found that for the trade-off between pressure drop and heat recovery, the best reclaimer tube bank arrangements were the in-line arrangements that had a greater number of tube rows than columns. For the majority of heating systems analyzed, reclaimers using tube arrangements of 5 x 3 (rows x columns) proved to be the most effective. The corresponding outside tube diameter was 1.25 inches, and the transverse and longitudinal tube spacings were 2.50 inches.

In general, decreasing the transverse tube spacing increases the heat transfer, but at the cost of an increased pressure drop. Increasing the longitudinal tube spacing of in-line tube banks also improves heat transfer, in that the flow is able to circulate more completely around tubes in deeper rows.

Increasing the number of tube rows in in-line tube banks results in the most effective heat reclaimers. The dominant term in the pressure drop correlations is the flow velocity, and therefore, increasing the number of tube rows in the direction of flow does not substantially increase the pressure drop (compared to reducing the transverse tube spacing), but significantly increases the heat recovery.

Heat reclaimers used with inefficient residential heating systems are recommended for chimney heights greater than 20 feet, and flue pipe diameters greater than 4 inches. It is these systems that the reclaimers will perform most effectively. Heat reclaimer outputs of 9,000 (Btu/h) for natural gas systems, and 13,000 (Btu/h) for oil-fired systems (based on a furnace steady-state efficiency of 60%), can be achieved for typical two-story homes with the furnace in the basement. This corresponds to supplying 100% of the basement heat load, and fuel savings of 6 to 12%. Also, if additional space heat is not required in the basement, over 80% of the reclaimed heat may be ducted to other areas of the house.

In general, the performance of the reclaimer depends upon the fuel burned, the steady-state efficiency of the furnace, and the sizing of the flue pipe and chimney system. Some heating systems which have chimney heights that cannot produce sufficient draft, or small flue pipe and chimney hydraulic diameters that substantially reduce flue gas temperatures, are shown not to be effective sources of waste heat. Heat reclaimers can be safely designed for these systems, but with very little amount of heat reclaimed.

In closing, heat recovery in residential combustion heating systems is very possible. As the technology of heat recovery equipment improves, new methods of creating an energy-efficient environment will develop. Presently, however, existing North American residential heating systems waste a significant amount of energy in the form of flue gas waste heat. Implementing residential heat recovery measures save both fuel costs and valuable energy resources.

APPENDIX A

ENTHALPY OF FORMATION AND CHANGE OF ENTHALPY AT 77 ^{o}F AND 1 (ATM) [14]

Nitrogen Diatomic (N_2)

$(h_f^o)_{537} = 0$ Btu/lbmol

$M_i = 28.013$

Temp. ^{o}F	Temp. ^{o}R	$(h^o - h^o_{537})$ Btu/lbmol
77	537	0
80	540	23
260	720	1278
440	900	2543
620	1080	3825
800	1260	5135
980	1440	6473
1160	1620	7839
1340	1800	9232
1520	1980	10651
1700	2160	12092
1880	2340	13552
2060	2520	15030
2240	2700	16522
2420	2880	18027
2600	3060	19544
2780	3240	21073
2960	3420	22608
3140	3600	24152
3320	3780	25704
3500	3960	27263

APPENDIX A continued

ENTHALPY OF FORMATION AND CHANGE OF ENTHALPY AT 77^{o}F AND 1 (ATM)

Oxygen Diatomic (O_2)

$(h_f^o)_{537}$ = 0 Btu/lbmol

M_i = 31.999

Temp. ^{o}F	Temp. ^{o}R	$(h^o - h^o_{537})$ Btu/lbmol
77	537	0
80	540	23
260	720	1303
440	900	2619
620	1080	3978
800	1260	5378
980	1440	6815
1160	1620	8280
1340	1800	9769
1520	1980	11279
1700	2160	12805
1880	2340	14348
2060	2520	15903
2240	2700	17471
2420	2880	19049
2600	3060	20637
2780	3240	22237
2960	3420	23848
3140	3600	25468
3320	3780	27097
3500	3960	28739

APPENDIX A continued

ENTHALPY OF FORMATION AND CHANGE OF ENTHALPY AT 77 ^{o}F AND 1 (ATM)

Carbon Dioxide (CO_2)

$(h_f^o)_{537}$ = -169297 Btu/lbmol

M_i = 44.01

Temp. ^{o}F	Temp. ^{o}R	$(h^o - h^o_{537})$ Btu/lbmol
77	537	0
80	540	29
260	720	1724
440	900	3577
620	1080	5557
800	1260	7641
980	1440	9815
1160	1620	12064
1340	1800	14371
1520	1980	16733
1700	2160	19138
1880	2340	21578
2060	2520	24052
2240	2700	26550
2420	2880	29074
2600	3060	31617
2780	3240	34177
2960	3420	36752
3140	3600	39343
3320	3780	41945
3500	3960	44559

APPENDIX A continued

ENTHALPY OF FORMATION AND CHANGE OF ENTHALPY AT $77^{o}F$ AND 1 (ATM)

Water (H_2O)

$(h_f^o)_{537}$ = -104036 Btu/lbmol

M_i = 18.015

Temp. ^oF	Temp. ^oR	$(h^o - h_{537}^o)$ Btu/lbmol
77	537	0
80	540	27
260	720	1485
440	900	2977
620	1080	4516
800	1260	6102
980	1440	7740
1160	1620	9432
1340	1800	11176
1520	1980	12978
1700	2160	14832
1880	2340	16736
2060	2520	18691
2240	2700	20691
2420	2880	22734
2600	3060	24817
2780	3240	26935
2960	3420	29088
3140	3600	31271
3320	3780	33484
3500	3960	35723

APPENDIX B

THERMAL PROPERTIES OF VARIOUS GASES AT 1 (ATM) [15]

Temp. ^{o}F	Temp. ^{o}R	Specific heat Btu/lbm-^{o}F	Absolute viscosity lbm/ft-h	Thermal conductivity Btu/ft-h-^{o}F
		Dry Air		
60	520	0.2404	0.04339	0.01466
80	540	0.2405	0.04467	0.01516
100	560	0.2406	0.04594	0.01566
120	580	0.2407	0.04718	0.01615
140	600	0.2409	0.04839	0.01664
160	620	0.2411	0.04959	0.01712
180	640	0.2413	0.05077	0.01759
200	660	0.2415	0.05193	0.01806

APPENDIX B continued

THERMAL PROPERTIES OF VARIOUS GASES AT 1 (ATM)

Temp. °F	Temp. °R	Absolute viscosity $\times 10^6$ lbm/ft-s	Thermal conductivity Btu/ft-h-°F
		Carbon Dioxide (CO_2)	
0	460	0.88	0.0076
100	560	1.05	0.0100
200	660	1.22	0.0125
500	960	1.67	0.0198
1000	1460	2.30	0.0318
		Nitrogen (N_2)	
0	460	1.055	0.0132
100	560	1.222	0.0154
200	660	1.380	0.0174
400	860	1.660	0.0212
600	1060	1.915	0.0252
800	1260	2.145	0.0291
1000	1460	2.355	0.0330
		Oxygen (O_2)	
0	460	1.215	0.0131
100	560	1.420	0.0159
200	660	1.610	0.0179
400	860	1.955	0.0228
600	1060	2.260	0.0277
800	1260	2.530	0.0324
1000	1460	2.780	0.0366

APPENDIX B continued

THERMAL PROPERTIES OF VARIOUS GASES AT 1 (ATM)

Temp. °F	Temp. °R	Absolute viscosity x 10^6 lbm/ft-s	Thermal conductivity Btu/ft-h-°F
	Water Vapor (H_2O)		
212	672	0.870	0.0145
300	760	1.000	0.0171
400	860	1.130	0.0200
500	960	1.265	0.0228
600	1060	1.420	0.0257
700	1160	1.555	0.0288
800	1260	1.700	0.0321
900	1360	1.810	0.0355
1000	1460	1.920	0.0388

APPENDIX C

COMPUTER PROGRAM LISTING

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```
$JOB
C  **************************************************************
C  * IT IS CRUCIAL FOR PROGRAM OPERATION THAT THE USER HAVE     *
C  * ADDITIONAL INSTRUCTIONS, SUPPLIED IN THE THESIS PAPER.     *
C  * THIS IS NOT AN INTERACTIVE-TYPE (USER FRIENDLY) PROGRAM.   *
C  **************************************************************
      REAL CH(5),AI(15),DC(3),D(7),STD(4),TG,THETA,GCP,GK,GU,
     * TA,ACP,AK,AU,DI(7),BRE(5),DP(3),WEIT(7),CZ(9),CEU1(4),
     * CEU2(4),CEU3(4),CEU4(4),CEU5(4),CU1(3),CU2(3),SLD(4),CEU6(4),
     * CEU7(4),CEU8(4),CEU9(4),CEU10(4),CZL(9),SLDL(4),STDL(4),FEFF(7),
     * C91(9),C92(9),C93(9),TO(7)
      READ, (CH(I1), I1=1,5)
      READ, (AI(I2), I2=1,15)
      READ, (DC(I3), I3=1,3)
      READ, (D(I4), I4=1,7)
      READ, (STD(I5), I5=1,4)
      READ, (SLD(I6), I6=1,4)
      READ, (STDL(I7), I7=1,4)
      READ, (SLDL(I8), I8=1,4)
      READ, (DI(I9), I9=1,7)
      READ, (BRE(I10), I10=1,5)
      READ, (DP(I11), I11=1,3)
      READ, (WEIT(I12), I12=1,7)
      READ, (CZ(I13), I13=1,9)
      READ, (CEU1(I14), I14=1,4)
      READ, (CEU2(I15), I15=1,4)
      READ, (CEU3(I16), I16=1,4)
      READ, (CEU4(I17), I17=1,4)
      READ, (CEU5(I18), I18=1,4)
      READ, (CU1(I19), I19=1,3)
      READ, (CU2(I20), I20=1,3)
      READ, (CEU6(I21), I21=1,4)
      READ, (CEU7(I22), I22=1,4)
      READ, (CEU8(I23), I23=1,4)
      READ, (CEU9(I24), I24=1,4)
      READ, (CEU10(I25), I25=1,4)
      READ, (CZL(I26), I26=1,9)
      READ, (FEFF(I27), I27=1,7)
      READ, (C91(I28), I28=1,9)
      READ, (C92(I29), I29=1,9)
      READ, (C93(I30), I30=1,9)
      READ, (TO(I31), I31=1,7)
      M222=?
      M666=?
      M777=?
      N=?
      L2=?
      I11F=?
      ITEM=?
CSTOP1  INPUT: PRINT MODE, TUBE CONFIGURATION, FUEL, FURNACE EFF.
C             M222_____PRINT ONE BANK "0" OR A SERIES OF BANKS "1".
C             M666_____STAGGERED "0" OR IN-LINE "1" TUBE ARRGT.
C             M777_____OIL "0" OR NAT GAS "1" TYPE FUEL.
C             N________NUMBER OF TUBE ROWS.
C             L2_______NUMBER OF TUBES PER ROW (MIN IN LEADING ROW)
```

```
C             I11F_____FURNACE STEADY-STATE EFFICIENCY:
C                      "1"--50%    "2"--55%    "3"--60%
C                      "4"--65%    "5"--70%    "6"--75%    "7"--80%
C             ITEM_____OUTSIDE AMBIENT TEMPERATURE:
C                      "1"--0F     "2"--10F    "3"--20F
C                      "4"--30F    "5"--40F    "6"--50F    "7"--60F
CSTOP2   LOOP CONTROL FOR OUTSIDE TEMP, CHIM HEIGHT, AND FURN EFF.
      DO 990 ITEM=?,?
      M888=0
      NTO=TO(ITEM)
      DO 980 I111=?,?
      DO 970 I11F=?,?
      NEFF=FEFF(I11F)*100.+.01
      NTFO=0
      IF(M777 .EQ. 0)THEN
       TFL=2918
       AM=1.24
      ELSE
       TFL=2287
       AM=1.6
      ENDIF
      TFI=600.
 10   TFO1=(TFI+TFL)/2
      CALL GAS(TFO1,THETA,GCPF,M777)
      TFO=TFL-FEFF(I11F)*1000./AM/GCPF
      TFOD=ABS(TFI-TFO)
      IF(TFOD .LT. 1)GO TO 20
       TFI=TFO
       NTFO=NTFO+1
      IF(NTFO .LT. 10)GO TO 10
       PRINT,'THE FURNACE EXIT TEMPERATURE IS NOT CONVERGING WITHIN 1 DE
     *GREE.'
       GO TO 999
 20   ITFO=TFO
      IF(M666 .EQ. 0)THEN
       L1=L2*N+N/2
      ELSE
       L1=L2*N
      ENDIF
      IF(M222 .EQ. 0)GO TO 70
      PRINT 30
 30   FORMAT('1','      ')
       PRINT,'              A S S U M E D    C O N D I T I O N S '
CSTOP3   FILL IN TUBE CONFIGURATION FOR A SERIES OF TUBE DESIGNS.
       PRINT,' '
       PRINT,' '
       PRINT,'                                                          '
       PRINT 35,L1
 35    FORMAT(' ',14X,'NUMBER OF TUBES:',I3,1X,'              *******    ')
       PRINT,'                                                *******    '
       PRINT,'              TUBE CONFIGURATION                *******    '
       PRINT,'                                                *******    '
       PRINT,'       TUBES: STEEL HEAT EXCHANGER TYPE         *******    '
       PRINT,'                                                *******    '
       PRINT,'       CASING: CONSTRUCTED OF 16 GA STEEL       *******    '
```

```
      PRINT,'                                                    *******    '
      IF(M777 .EQ. 1)THEN
      PRINT,'       FUEL: NATURAL GAS'
      ELSE
      PRINT,'       FUEL: NO. 2 FUEL OIL'
      ENDIF
      PRINT,' '
      PRINT,'       BAROMETRIC PRESSURE: 28.86 INCHES MERCURY '
      PRINT 37,NTO
 37   FORMAT('0','       OUTSIDE AIR TEMPERATURE:',I3,'DEGREES FAHRENHE
     *IT')
      PRINT,' '
      PRINT,'       FURNACE ROOM TEMPERATURE: 69 DEGREES FAHRENHEIT'
      PRINT 40,NEFF
 40   FORMAT('0','       FURNACE STEADY-STATE EFFICIENCY:',I3,'%')
      PRINT 50,ITFO
 50   FORMAT('0','       FURNACE EXIT TEMPERATURE: ',I4,' DEGREES FAHRE
     *NHEIT')
      PRINT 60,AM
 60   FORMAT('0','       GAS MASS FLOW RATE(POUND-MASS/HR): APPROXIMATE
     *LY',F7.4,' TIMES THE FURNACE INPUT')
      PRINT,' '
      PRINT,'       AIR FLOW THROUGH TUBE BANK: 160 CFM'
      PRINT,' '
      PRINT,'       CHIMNEY CONSTRUCTION: MASONRY TYPE WITH LINER'
      PRINT,' '
      PRINT,' '
      PRINT,'               K E Y    N O T E S'
      PRINT,'               *****************'
      PRINT,' '
      PRINT,' '
      PRINT,' FOR ALL CHIMNEY HEIGHTS, 4,5,6 INCH DIA FLUE PIPES CORRES
     *POND TO 5,6,7 INCH EQUIVALENT CHIMNEY DIAMETERS.'
      PRINT,' '
      PRINT,' LATERAL BREECHING OF 2 FEET CORRESPONDS TO CHIMNEY HEIGHT
     *S OF 15 AND 20 FEET.'
      PRINT,' '
      PRINT,' LATERAL BREECHING OF 3 FEET CORRESPONDS TO CHIMNEY HEIGHT
     *S OF 25 AND 30 FEET.'
      PRINT,' '
      PRINT,' LATERAL BREECHING OF 4 FEET CORRESPONDS TO A CHIMNEY HEIG
     *HT OF 35 FEET.'
 70   PI=3.14159
      IF(M777 .EQ. 1)THEN
       TR=(TFO-60)/1.8733
       C81=1.5
      ELSE
       TR=(TFO-60)/1.544
       C81=.5
      ENDIF
      TAA=69.
      TAR=TAA+460.
      TLIM=600.
      CKSTL=21.
      E1=.6
```

```
      BP=28.86
      C82=.75
      C83=1.25
      WAA=160.
      WA1=WAA*60.*.08
      WA=WA1/L1
      U1=1.
      NCVRG=1
      M444=1
C  ***************** K E Y   I N P U T   D A T A ********************
C  ARRAY VARIABLE             1     2     3     4     5     6     7
C  ------------------------------------------------------------------
C  CHIMNEY HT (FT)  CH(I111)  15    20    25    30    35    -     -
C  LAT BREECH (FT) BRE(I111)  2     2     3     3     4     -     -
C  FLUE DIA (IN)    DP(I112)  4     5     6     -     -     -     -
C  CHIM DIA (IN)    DC(I112)  5     6     7     -     -     -     -
CSTOP4
C  TUBE O.D. (IN)    D(I113) .625  .75   .875  1.00  1.25  1.50  1.75
C  TRANS SPC-TUBE O.D. RATIO:
C  IN-LINE ARRGT  STDL(I115) 1.75  2.00  2.25  2.50   -     -     -
C  STAGG ARRGT     STD(I115) 1.25  1.50  2.00  2.50   -     -     -
C  LONGT SPC-TUBE O.D. RATIO:
C  IN-LINE ARRGT  SLDL(I116) 1.25  1.50  2.00  2.50   -     -     -
C  STAGG ARRGT     SLD(I116) 0.90  1.20  1.50  1.80   -     -     -
       I111=?
       I112=?
       I113=?
       I115=?
       I116=?
C                       I111-------------CH,BRE
C                       I112-------------DP,DC,CU1,CU2
CSTOP5  KEY INPUT       I113-------------D,DI
C                       I114-------------AI
C                       I115-------------STD,STDL,CEU1 THRU CEU10
C                       I116-------------SLD,SLDL
      DO 600 I111=?,?
      DO 590 I112=?,?
      I114=5*(I112-1)+I111
      IF(I114 .EQ. 5 .AND. M222 .EQ. 1)GO TO 590
      IF(I114 .EQ. 5 .AND. M222 .NE. 1)GO TO 980
      DO 580 I113=?,?
      DO 570 I115=?,?
      DO 560 I116=?,?
      TW=???
      IF(M666 .EQ. 0)THEN
       SL=SLD(I116)*D(I113)
       ST=STD(I115)*D(I113)
       A=(L2+1)*ST
      ELSE
       SL=SLDL(I116)*D(I113)
       ST=STDL(I115)*D(I113)
       A=L2*ST
      ENDIF
      BOXMAX=1.5*DP(I112)
      BOXMIN=DP(I112)+.25
```

```
      DROP1=CO1*(VOI/4005)**2
      CO3=-.5*AR+.5
      UINF=W/ROI/AHH/3600.
      SST1=ST-D(I113)
      IF(M666 .NE. 0)GO TO 80
      SD=((ST/2)**2+SL**2)**.5
      SSD1=SD-D(I113)
      SSD11=2*SSD1
      IF(SST1 .LT. SSD11)THEN
       UMAX=UINF*ST/SST1
      ELSE
       UMAX=UINF*ST/2/SSD1
      ENDIF
      IF(SRDO .GT. 1.25)THEN
       CK1=.951*SRDO**.284
      ELSE
       CK1=1/SRDO**.048
      ENDIF
      GO TO 90
80    UMAX=UINF*ST/SST1
      CK1=1.009/((STDL(I115)-1.)/(SLDL(I116)-1.))**.744
90    SUMZ=0
       DO 100 KZZ=1,N
       IF(M666 .EQ. 0)THEN
        SUMZ=SUMZ+CZ(KZZ)
       ELSE
        SUMZ=SUMZ+CZL(KZZ)
       ENDIF
100    CONTINUE
      CZZ=SUMZ/N
110   DT=.2554*BP*CH(I111)*(1/(TO(ITEM)+460)-1/(TM2+460))
      IF(M333 .EQ. 1)DTN=DT
      RO=1.325*BP/(TM2+460)
      VM=W/(19.63*RO*DC(I112)**2)
      DROP0=C86*RO*VM**2/334.88
      TOTAL0=DT-DROP0
      IF(M333 .EQ. 1)TOTALN=TOTAL0
      ROO=1.325*BP/(TGOO+460)
      VOO=W*2.4/A23/ROO
      DROP3=CO3*(VOO/4005)**2
      PPPPP=DROP0+DROP1+DROP3
      PPT=DT-PPPPP
      IF(M222 .EQ. 0)GO TO 120
      IF(PPT .LT. .049)GO TO 480
120    ROB=1.325*BP/(TB+460)
      CALL GAS1(TB,GK,GUTB,M777)
      UBINF=W/ROB/AHH/3600.
      IF(M666 .EQ. 1)GO TO 130
       IF(SST1 .LT. SSD11)THEN
        UBMAX=UBINF*ST/SST1
       ELSE
        UBMAX=UBINF*ST/2/SSD1
       ENDIF
       GO TO 140
130   UBMAX=UBINF*ST/SST1
```

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```
 140  REMAX=UBMAX*ROB*D(I113)*300./GUTB
      IF(M666 .NE. 0)GO TO 220
      IF(I115 .LE. 2)GO TO 150
       GO TO 160
 150  IF(REMAX .LT. 1000)GO TO 190
       GO TO 200
 160  IF(I115 .EQ. 3)GO TO 170
       GO TO 180
 170  IF(REMAX .GT. 10000.)GO TO 190
       GO TO 200
 180  IF(REMAX .GT. 5000.)GO TO 190
       GO TO 200
 190  CONST1=CEU6(I115)
      CONST2=CEU7(I115)
      CONST3=CEU8(I115)
      CONST4=CEU9(I115)
      CONST5=CEU10(I115)
      GO TO 210
 200  CONST1=CEU1(I115)
      CONST2=CEU2(I115)
      CONST3=CEU3(I115)
      CONST4=CEU4(I115)
      CONST5=CEU5(I115)
 210  IF(REMAX .GT. 10000.)THEN
       CK1=1.28-.708/SRDO+.55/SRDO**2-.113/SRDO**3
      ENDIF
      EUK1=CONST1+CONST2/REMAX-CONST3/REMAX**2+CONST4/REMAX**3
     *-CONST5/REMAX**4
      GO TO 230
 220  IF(I116 .EQ. 1 .AND. REMAX .LT. 2000.)THEN
       EUK1=.272+207./REMAX+102/REMAX**2-286./REMAX**3
      ENDIF
      IF(I116 .EQ. 1 .AND. REMAX .GE. 2000.)THEN
       EUK1=.267+2490./REMAX-9.27E6/REMAX**2+1.E10/REMAX**3
      ENDIF
      IF(I116 .EQ. 2 .AND. REMAX .LT. 2000.)THEN
       EUK1=.263+86.7/REMAX-2.02/REMAX**2
      ENDIF
      IF(I116 .EQ. 2 .AND. REMAX .GE. 2000.)THEN
       EUK1=.235+1970./REMAX-1.24E7/REMAX**2+3.12E10/REMAX**3-2.74E13
     */REMAX**4
      ENDIF
      IF(I116 .EQ. 3 .AND. REMAX .LT. 800.)THEN
       EUK1=.188+56.6/REMAX-646./REMAX**2+6010./REMAX**3-1.83E4/
     *REMAX**4
      ENDIF
      IF(I116 .EQ. 3 .AND. REMAX .GE. 800.)THEN
       EUK1=.247-.595*REMAX/1.E6+.15*REMAX**2/1.E11-.137*REMAX**3/1.E17
     *+.396*REMAX**4/1.E24
      ENDIF
      IF(I116 .EQ. 4)THEN
       EUK1=.177-3.11E-7*REMAX+1.17E-12*REMAX**2
      ENDIF
 230  IF(REMAX .LT. 1000.)THEN
       EU2=1.
```

```
      ELSE
       C98=REMAX**.196*1.076
       C97=.968/2.71828**C98
       CALL GAS1(TTWB,GK,GUEU,M777)
       EU2=(GUEU/GUTB)**C97
      ENDIF
       UONF=W/ROO/AHH/3600.
       DROP4=(ROO*UONF**2-ROI*UINF**2)*.19248/32.2
      EUZ=EUK1*CK1*CZZ*EU2
      DROP2=EUZ*ROB*UBMAX**2*N*.19248/64.4
      TOTALD=PPT-DROP2+DROP4
      IF(M222 .EQ. 0)GO TO 250
      IF(TOTALD .LT. .049)GO TO 480
 250  IF(M333 .GT. 1)GO TO 470
       M333=M333+1
C*******END PRESSURE DROP ANALYSIS*********
      CLOG=L1*D(I113)**2*PI/4/A/C
      CALL GAS(TGI,THETA,GCPI,M777)
      CALL GAS1(TGI,GKI,GUI,M777)
      PRF=GCPI*GUI/GKI
      N44=0
      B544=1
      A22=PI*D(I113)**2/4
      A222=(A*C-L1*A22)/144
      A223=(A*B-A23)/144
      A224=B*C/144
      AB=2*A222+2*A223+2*A224
      AB1=AB-2*A223
      AO11=B/144*PI*D(I113)
      AI11=B/144*PI*DI(I113)
      RE1=UMAX*ROI*D(I113)*300/GUI
      IF(RE1 .GT. 1000)GO TO 255
       IF(M666 .EQ. 0)THEN
        C99=1
       ELSE
        C99=C91(N)
       ENDIF
        GO TO 260
 255   IF(M666 .EQ. 0)THEN
        C99=C93(N)
       ELSE
        C99=C92(N)
       ENDIF
C*********BEGIN ITERATIVE PROCESS***********
 260  NG11=0
      N45=0
      TWR=TW+460
      Q1=(TW-TAA)**1.25*(.39*A222/X222**.25+.58*(A224/(B/12)**.25
     *+A223/(B/12)**.25))
      Q2=.1714E-8*E1*AB*(TWR**4-TAR**4)
 270  TGF=(TG1+TW)/2
      CALL GAS(TGF,THETA,GCP1,M777)
      CALL GAS1(TGF,GK1,GU1,M777)
      PR1=GCP1*GU1/GK1
      Q33=.036*(4*W/PI/(1-CLOG)/DH/GU1)**.8*PR1**.33*(DH*12/C)
```

```
     ***.055*GK1/DH*AB1
      TG2=(Q1+Q2)/Q33+TW
      Z111=TG2-TG1
      IF(ABS(Z111) .LT. .01)GO TO 300
       TG3=TG1+Z111/2
      IF(TG3 .LT. TGI)GO TO 280
       GO TO 390
 280  TG1=TG3
      NG11=NG11+1
      IF(NG11 .LT. 20)GO TO 270
       PRINT,'THE RECLAIMER BULK GAS TEMPERATIRE IS NOT CONVERGING WITHI
     *N 0.01 DEGREES.'
       GO TO 999
 300  TGO=2*TG1-TGI
C***********END STEP ONE***************
      NNN2=0
 310  CALL GAS(TTW1,THETA,GCP2,M777)
      CALL GAS1(TTW1,GK2,GU2,M777)
      PRS=GCP2*GU2/GK2
      IF(M666 .NE. 0)GO TO 320
      IF(SRDO .GT. 2.)THEN
       HCTO=C99*.4*RE1**.6*PRF**.61/PRS**.25*GKI*12/D(I113)
      ELSE
       HCTO=C99*.35*(SRDO)**.2*RE1**.6*PRF**.61/PRS**.25*GKI*
     * 12/D(I113)
      ENDIF
      GO TO 330
320   HCTO=C99*.27*RE1**.63*PRF**.61/PRS**.25*GKI*12./D(I113)
330   TAF1=(TTA1+TTW1)/2
      CALL AIR(TAF1,ACP2,AK2,AU2)
      HCTI=12*AK2/DI(I113)*.036*(48*WA/PI/DI(I113)/AU2)**.8*
     *(AU2*ACP2/AK2)**.33*(DI(I113)/B)**.055
      UO1=1/(1/HCTO+AO11*ALOG(D(I113)/DI(I113))/2/PI/B/CKSTL*
     *12+AO11/(HCTI*AI11))
      CALL AIR(TTA1,ACPB,AKB,AUB)
      TTA2=(HCTI*AI11*TTW1+2*WA*ACPB*TAA)/(2*ACPB*WA+HCTI*AI11)
      Z222=TTA2-TTA1
      TTW2=UO1/HCTO*(TTA1-TG1)+TG1
      Z333=TTW2-TTW1
      IF(NNN2 .GT.20)THEN
       PRINT,'THE TUBE BULK AIR AND/OR TUBE WALL TEMPERATURES ARE NOT CO
     *NVERGING WITHIN 0.1 DEGREES.'
       GO TO 999
       ENDIF
       NNN2=NNN2+1
      IF(ABS(Z222) .LT. .1 .AND. ABS(Z333) .LT. .1)GO TO 360
       TTA3=TTA1+Z222/2
      TTW3=TTW1+Z333/2
      IF(TTA3 .LT. TAA)GO TO 340
       TTA1=TTA3
      TTW1=TTW3
      GO TO 310
 340  TTW3=TTW3+1.
      IF(TTW3 .GT. TG1) GO TO 390
       TTW1=TTW3
```

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      GO TO 310
 360  TTAO=2*TTA1-TAA
C*********END STEP TWO AND THREE************
      CALL GAS(TG1,THETA,GCP4,M777)
 370  Q11=Q1/(TW-TAA)**1.25*(TWA-TAA)**1.25
      CNUM=L1*U01*A011*(TG1-TTA1)+Q11-W*GCP4*(TGI-TGO)
      IF(CNUM .GT. 0) GO TO 410
       TWB=(TAR**4-CNUM/E1/AB/.1714E-8)**.25-460
      Z444=TWB-TWA
      IF(ABS(Z444) .LT. .01)GO TO 380
       TWA=TWA+Z444/2
      GO TO 370
 380  Z544=TWA-TW
      IF(ABS(Z544) .LT. NCVRG)GO TO 460
       IF(Z544 .LT. 0)GO TO 420
        IF(N44 .GT. 0)GO TO 400
         IF(Z544 .GT. 400)THEN
          DELTA=(Z544-200)/100.
          TW=TW+DELTA
          GO TO 260
         ENDIF
 390    TW=TW+1.0
      IF(TW .LT. TLIM) GO TO 260
       PRINT,'THE CASING WALL TEMPERATURE IS GREATER THAN ',TLIM,'F.'
       GO TO 999
 400  TW=TW+B544
      GO TO 260
 410  N45=N45+1
      IF(N45 .GT. 20)THEN
       PRINT,' THE CASING WALL TEMPERATURE CANNOT BE CALCULATED BECAUSE
     * A HEAT BALANCE DOES NOT EXIST.'
       GO TO 999
      ENDIF
       TWA=TWA-20.
      IF(TWA .LT. 99.)GO TO 420
      GO TO 370
 420  N44=N44+1
      IF(N44 .GT. 7)THEN
       PRINT,'THE CASING WALL TEMPERATURE IS NOT CONVERGING WITHIN ',
     * NCVRG,'DEGREES.'
       GO TO 999
       ENDIF
       TW=TW-.9*B544
      B544=B544/10
      GO TO 260
C*******END STEP FOUR AND ITERATION**********
 460  QGAIN=W*GCP4*(TGI-TGO)
      QBANK=WA1*ACPB*(TTAO-TAA)
      QBOX=QGAIN-QBANK
      WEIGHT=AB*2.5+L1*B/12*WEIT(I113)+6.
      EQLG=12.*GCP4*W*(TGI-TGO)/U1/PI/DP(I112)/(TG1-TAA)
      TM2=TR*(CU1(I112)-CU2(I112)*(BRE(I111)+EQLG))+TO(ITEM)
      TB=TG1
      TGOO=TGO
      TTWB=TTW1
```

```
      IF(M333 .EQ. 2)GO TO 110
 470   CALL GAS(TM2,THETA,GCPD,M777)
      TCHO=U1*PI*DC(I112)*CH(I111)/24/W/GCPD*(68-TM2)+TM2
      PERC=QGAIN/AI(I114)*.1/(1-FEFF(I11F))
      SAVE=QGAIN*.1/AI(I114)
      IF(M222 .EQ. 0) GO TO 610
       M555=1
 480  IF(M222 .EQ. 0)THEN
       PRINT,'RECLAIMER DESIGN FAILS PRESSURE DROP'
       GO TO 999
      ENDIF
       IF(M444 .GT. 1)GO TO 500
        PRINT 490
 490  FORMAT('1','MASON  FURN   FLUE   TUBE BANK ARRAY      BOX DIMENSIO
     *NS      GAS      ALL TEMPS DEGREES FAHRENHEIT      APPROX DRAFT   TO
     *TAL   %  ')
      PRINT,'CHIM   INPUT  PIPE TUBE   TRANS  LONGIT  16 GA SHEET STEEL
     *  MAX    BOX    BOX    TUBE   BOX    CHIM   TOTAL  W/UNIT  HEAT
     *HEAT'
      PRINT,'HEIGHT *1000  DIAM O.D.   SPCG   SPCG   WIDTH  DEPTH  LENGT
     *H  RE    GAS    GAS    AIR    WALL   GAS    WEIGHT INCHES  GAIN
     *LOSS'
      PRINT,'FEET   BTU/HR INCH INCHES INCHES INCHES INCHES INCHES INCHE
     *S  NO.   INLET  OUTLET EXIT   TEMP   EXIT   LBS    WATER   BTU/HR
     *RECOV'
 500  IF(M555 .EQ. 1)GO TO 530
      PRINT 520,CH(I111),AI(I114),DP(I112),D(I113),ST,SL,A,B,C
 520  FORMAT('0',1X,F3.0,3X,F5.0,1X,F4.0,1X,F6.3,1X,F6.3,1X,
     *F6.3,1X,F6.2,1X,F6.2,1X,F6.2,'  * * * I N S U F F I C I E N T   D
     *R A F T * * * ')
      GO TO 550
 530  PRINT 540,CH(I111),AI(I114),DP(I112),D(I113),ST,SL,A,B,C,REMAX,
     *TGI,TGO,TTAO,TW,TCHO,WEIGHT,TOTALD,QGAIN,PERC
 540  FORMAT('0',1X,F3.0,3X,F5.0,1X,F4.0,1X,F6.3,1X,F6.3,1X,F6.3
     *,1X,F6.2,1X,F6.2,1X,F6.2,2X,F6.0,F6.1,1X,F6.1,F6.1,2X,F6.1,1X,
     *F6.1,F6.1,2X,F7.4,F7.0,1X,F5.1)
CSTOP7  DO LOOP CONTINUATIONS FOR KEY INPUTS
 550  IF(M444 .EQ. 27)THEN
       M444=1
      ELSE
       M444=M444+1
      ENDIF
 560  CONTINUE
 570  CONTINUE
 580  CONTINUE
 590  CONTINUE
 600  CONTINUE
      GO TO 999
 610  IF(M888 .GT. 0)GO TO 650
      PRINT 620
 620  FORMAT('1','  ')
      DO 630 J=1,6
      PRINT,' '
 630  CONTINUE
      PRINT,' G E N E R A L   N O T E S'
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      DO 640 J=1,4
      PRINT,' '
 640  CONTINUE
      PRINT,' THE FURNACE IS A RESIDENTIAL, NATURAL DRAFT TYPE.'
      PRINT,' '
      PRINT,' '
      PRINT,' THE FURNACE ROOM TEMPERATURE IS 69F.'
      PRINT,' '
      PRINT 645,NTO
 645  FORMAT('0',' THE OUTSIDE AMBIENT TEMPERATURE IS ',I3,'F.')
      PRINT,' '
      PRINT,' '
      PRINT,' THE BAROMETRIC PRESSURE AT 1000 FEET ELEVATION IS 28.86 IN
     *CHES OF MERCURY.'
      PRINT,' '
      PRINT,' '
      PRINT,' THE CHIMNEY IS OF MASONRY CONSTRUCTION WITH LINER.'
      PRINT,' '
      PRINT,' '
      PRINT,' THE FLUE PIPE IS SINGLE WALL STEEL WITH VARIOUS ATTACHMENT
     *S.'
      PRINT,' '
      PRINT,' '
      IF(M777 .EQ. 1)THEN
      PRINT,' INFILTRATION THROUGH THE DRAFT HOOD IS ASSUMED NEGLIGIBLE.
     *'
      ELSE
      PRINT,' INFILTRATION THROUGH THE BAROMETRIC DRAFT DIVERTER IS ASSU
     *MED NEGLIGIBLE.'
      ENDIF
      PRINT,' '
      PRINT,' '
      PRINT,' THE DEVICE CASING IS CONSTRUCTED OF 16 GA CARBON STEEL.'
      PRINT,' '
      PRINT,' '
      PRINT,' THE TUBE BANK CONSISTS OF THIN-WALL, HEAT EXCHANGER TYPE S
     *TEEL TUBES.'
      M888=M888+1
 650  PRINT 660
 660  FORMAT('1',' ')
      PRINT,'H E A T   R E C O V E R Y   P E R F O R M A N C E'
      IF(M777 .EQ. 0)THEN
       PRINT 670,ITFO
 670   FORMAT('0','FUEL: NO. 2 FUEL OIL    FURNACE EXIT TEMPERATURE: ',
     * I4,'F')
      ELSE
       PRINT 680,ITFO
 680   FORMAT('0','FUEL: NATURAL GAS       FURNACE EXIT TEMPERATURE: ',
     * I4,'F')
      ENDIF
      PRINT 690,NEFF,FI,FO
 690  FORMAT('0','FURNACE EFF: ',I3,'%    INPUT: ',F8.0,' BTU/HR    OUTP
     *UT',F8.0,' BTU/HR')
      PRINT 700,CH(I111)
 700  FORMAT('0','CHIMNEY HEIGHT..............',F4.0,' FEET')
```

```
      PRINT 710,DC(I112)
 710  FORMAT('0','INSIDE EQUIV. CHIMNEY DIA....',F3.0,' INCHES')
      PRINT 720,DP(I112)
 720  FORMAT('0','FLUE PIPE DIA................',F3.0,' INCHES')
      PRINT 730,BRE(I111)
 730  FORMAT('0','FLUE PIPE LATERAL BREECHING..',F3.0,' FEET')
      PRINT 750,TOTALN
 750  FORMAT('0','SYSTEM UPDRAFT W/O DEVICE....',F7.4,' INCHES WATER')
      PRINT 760,W
 760  FORMAT('0','GAS MASS FLOW RATE...........',F6.1,' POUND-MASS/HR')
      PRINT 770,WAA
 770  FORMAT('0','TUBE AIR VOLUME FLOW RATE....',F5.0,' CFM')
      PRINT 780,A
 780  FORMAT('0','DEVICE CASING WIDTH..........',F6.2,' INCHES',14X,'
     *        *******     ')
      PRINT,'                                                  TUBE
     *        *******     '
      PRINT 790,B
 790  FORMAT(' ','DEVICE CASING DEPTH..........',F6.2,' INCHES',14X,'
     *        *******     ')
CSTOP8  FILL IN TUBE CONFIGURATION FOR SINGLE RECLAIMER DESIGNS.
      PRINT,'                                             CONFIGURATION
     *        *******     '
      PRINT 800,C
 800  FORMAT(' ','DEVICE CASING LENGTH.........',F6.2,' INCHES',14X,'
     *        *******     ')
      PRINT 805,L1
 805  FORMAT(' ',48X,I3,' TUBES   ',
     *'       *******     ')
      PRINT 810,D(I113)
 810  FORMAT(' ','TUBE O.D.....................',F6.3,' INCHES',14X,'
     *        *******     ')
      PRINT,'
     *        *******     '
      PRINT 820,DI(I113)
 820  FORMAT(' ','TUBE I.D.....................',F6.3,' INCHES',14X,'
     *        *******     ')
      PRINT 830,ST
 830  FORMAT('0','TRANSVERSE SPACING...........',F6.3,' INCHES')
      PRINT 840,SL
 840  FORMAT('0','LONGITUDINAL SPACING.........',F6.3,' INCHES')
      PRINT 850,TGI,TG1,TGO
 850  FORMAT('0','GAS INLET TEMP:',F6.1,' F   BULK GAS TEMP:',F6.1,' F
     * GAS EXIT TEMP:',F6.1,' F')
      PRINT 860,TAA,TTA1,TTAO
 860  FORMAT('0','AIR INLET TEMP:',F6.1,' F   BULK AIR TEMP:',F6.1,' F
     * AIR EXIT TEMP:',F6.1,' F')
      PRINT 870,TW,TTW1
 870  FORMAT('0','DEVICE CASING WALL TEMPERATURE..........',F6.1,' F  TU
     *BE WALL TEMP:',F6.1,' F')
      PRINT 880,HCTI
 880  FORMAT('0','TUBE INSIDE HEAT TRANSFER COEFFICIENT...',F6.2,' BTU/H
     *R/SQFT/F')
      PRINT 890,HCTO
 890  FORMAT('0','TUBE OUTSIDE HEAT TRANSFER COEFFICIENT..',F6.2,' BTU/H
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```
     *R/SQFT/F')
      PRINT 900,WEIGHT
 900  FORMAT('0','TOTAL WEIGHT OF DEVICE..................',F5.1,' POUND
     *S')
      PRINT 910,TCHO
 910  FORMAT('0','CHIMNEY EXIT TEMPERATURE WITH DEVICE....',F6.1,' F')
      PRINT 920,REMAX
 920  FORMAT('0','MAXIMUM REYNOLDS NUMBER.................',F7.0)
      PRINT 930,DT
 930  FORMAT('0','THEORETICAL DRAFT WITH DEVICE...........',F7.4,' INCHE
     *S WATER')
      PRINT 940,TOTALD
 940  FORMAT('0','UPDRAFT OF SYSTEM WITH DEVICE...........',F7.4,' INCHE
     *S WATER')
CSTOP9     DO LOOP CONTINUATIONS FOR SINGLE DESIGN ONLY.
      PRINT 950,QBANK,QBOX
 950  FORMAT('0','HEAT GAIN BY TUBE BANK:',F7.0,'BTU/HR    GAIN BY EXPOS
     *ED CASING:',F6.0,'BTU/HR')
      PRINT 960,QGAIN,PERC
 960  FORMAT('0','TOTAL HEAT GAIN BY DEVICE:',F7.0,'BTU/HR     ',F5.1,'%
     * OF HEAT LOSS RECOVERED')
      PRINT 965,SAVE
 965  FORMAT('0','FUEL SAVED:',F5.1,'%')
 970  CONTINUE
 980  CONTINUE
 990  CONTINUE
 999  STOP
      END
       SUBROUTINE GAS(TG,THETA,GCP,M777)
       THETA=(TG+460.)/180.
       IF(M777 .EQ. 1)THEN
        GCP=1.0857E-5*THETA**2+1.6185E-5*THETA**1.5-.005*THETA
     *+.0858*THETA**.5-.1599*THETA**.25-3.4004/THETA**1.5
     *+7.0031/THETA**2-5.2101/THETA**3+.4011
       ELSE
        GCP=1.7929E-5*THETA**2+1.0089E-5*THETA**1.5-6.0461E-3*THETA
     *+8.9865E-2*THETA**.5-.14915*THETA**.25+.37592-3.2884/THETA**1.5
     *+6.8104/THETA**2-5.1177/THETA**3
       ENDIF
       RETURN
       END
       SUBROUTINE GAS1(TG,GK,GU,M777)
       IF(M777 .EQ. 1)THEN
        GK=2.1375E-5*TG+.012465
        GU=4.5429E-5*TG+.040693
       ELSE
        GK=2.1357E-5*TG+.012257
        GU=4.5319E-5*TG+.04158
       ENDIF
       RETURN
       END
       SUBROUTINE AIR(TA,ACP,AK,AU)
       ACP=5.0E-6*TA+.2401
       AK=2.4833E-5*TA+.0132
       AU=6.3167E-5*TA+.0396
```

```
       RETURN
       END
$ENTRY
15.,20.,25.,30.,35.
54,58,60,63,0,90,98,101,106,109,142,160,161,167,169
5,6,7
.625,.75,.875,1.,1.25,1.5,1.75
1.25,1.5,2.,2.5
.9,1.2,1.5,1.8
1.75,2.,2.25,2.5
1.25,1.5,2.,2.5
.555,.65,.775,.9,1.15,1.4,1.62
2,2,3,3,4
4,5,6
.2205,.3738,.4406,.5073,.6408,.7743,1.170
.45,.8902,1,1,1,1,1,1,1
.245,.203,.343,.33
3390,2480,303,98.9
9.84E6,7.58E6,7.17E4,1.48E4
1.32E10,1.04E10,8.8E6,1.92E6
5.99E12,4.82E12,3.8E8,8.62E7
.8,.815,.845
.0333,.0259,.0202
.795,.683,.162,.119
247,111,1810,4980
-335,97.3,-7.42E7,5.07E7
1550,426,-1.65E12,2.51E11
2410,574,-8.72E15,4.63E14
1.25,1.13,1,1,1,1,1,1,1
.5,.55,.6,.65,.7,.75,.8
.83,.87,.91,.93,.95,.96,.97,.98,.98
.69,.80,.86,.9,.89,.94,.95,.96,.97
.62,.75,.83,.89,.92,.94,.95,.96,.97
0,10,20,30,40,50,60
```

APPENDIX D

LIST OF PROGRAM VARIABLES

SYMBOL	DEFINITION	UNITS
A	Reclaimer width	in
A22	Tube cross-sectional area	in^2
A23	Flue pipe cross-sectional area	in^2
A222, A223, A224	Area of reclaimer rectangular surfaces	ft^2
AB	Reclaimer exterior surface area	ft^2
AB1	Reclaimer interior surface area (rectangular duct)	ft^2
ACP, ACP2, ACPB	Specific heat of air at constant pressure	Btu/lbm-^oF
AHH	Reclaimer cross-sectional area based on the hydraulic diameter	ft^2
AI	Furnace input	Btu/h x 10^3
AI11	Tube bank interior surface area	ft^2
AK, AK2, AKB	Dynamic viscosity of air	lbm/h-ft
B	Reclaimer casing depth	in
B544	Reclaimer casing temperature incrementation	^oF
BOXMAX, CMAX, BOXMIN	Reclaimer dimension restrictions	in
BP	Barometric pressure at 1000 ft. elevation	in. mercury
BRE	Flue pipe horizontal breeching	ft
C	Reclaimer casing length	in
C81-C86	System friction coefficients	none
C91, C92, C93, C99	Tube row correction factors for heat transfer analysis	none

APPENDIX D continued

LIST OF PROGRAM VARIABLES

SYMBOL	DEFINITION	UNITS
C97, C98, CEU1-CEU10, CK1, CONST1-CONST10, CZ, CZL, KZZ, SUMZ	Euler number correction variables	none
CH	Chimney height	ft
CKSTL	Thermal conductivity of steel	Btu/h-ft-^{o}F
CLOG	Reclaimer-tube bank volume ratio	none
CMAX	see BOXMAX	
CNUM, Q2	Radiation heat transfer from reclaimer	Btu/h
CO1, CO3	Correction factors for duct area changes	none
CU1, CU2	Mean chimney temperature variables	none
D	Outside tube diameter	in
DC	Chimney inside hydraulic diameter	in
DELTA	Temperature incrementation variable	F
DH	Reclaimer hydraulic diameter	ft
DI	Inside tube diameter	in
DP	Flue pipe diameter	in
DROP0-DROP4, PPPPP	Pressure drop	in. water
DT	Theoretical draft	in. water
E1	Reclaimer surface emissivity	none
EQLG	Equivalent flue pipe breeching	ft
EU2, EUK	Euler number multipliers	none

APPENDIX D continued

LIST OF PROGRAM VARIABLES

SYMBOLS	DEFINITION	UNITS
EUZ	Total Euler number	none
FEFF, NEFF	Furnace steady-state efficiency	none
FI, FO	Furnace input and output, respectively	Btu/h
GCP, GCP1, GCP2, GCP4, GCPC, GCPD, GCPF, GCPI	Specific heat of flue gas at constant pressure	Btu/lbm-^{o}F
GK, GK1, GK2	Thermal conductivity of flue gas	Btu/h-ft-^{o}F
GU, GU1, GU2, GUEU, GUTB	Dynamic viscosity of flue gas	lbm/h-ft
HCTI, HCTO	Interior and exterior tube bank heat-transfer coefficients, respectively	Btu/h-ft^{2}-^{o}F
I11F, I111-I116, ITEM	Array variables for key input data	none
ITFO	see TFI	
KZZ	see C97	
L1	Number of tubes in the tube bank	none
L2	Number of tubes in leading row of bank	none
M222-M888	Design selection and output controls	none
N	Number of tube rows	none
N44, N45, NG11, NNN2, NTFO	Iteration divergence controls	none
NCVRG	Convergence control for reclaimer casing temperature	^{o}F
NEFF	see FEFF	

APPENDIX D continued

LIST OF PROGRAM VARIABLES

SYMBOL	DEFINITION	UNITS
NTO	see TO	
NTW	see TW	
PERC	Per cent of waste heat recovered	none
PI	3.14159	
PPPPP	see DROP1	
PPT	Updraft without tube bank pressure losses	in. water
PR1, PRF, PRS	Prandtl number of flue gas	none
Q1, Q11	Natural convection heat transfer from reclaimer casing	Btu/h
Q2	see CNUM	
QBANK, QBOX, QGAIN	Heat gain from reclaimer	Btu/h
RE1, REMAX	Maximum flow Reynolds numbers	none
RO, ROB, ROO, ROI	Density of flue gas	lbm/ft^3
SAVE	Per cent fuel savings	none
SD	Diagonal tube spacing	in.
SL	Longitudinal tube spacing	in.
STD, STDL	Dimensionless longitudinal tube spacing	none
SRDO	Tube spacing ratio	none
SSD1, SSD11, SST1	Tube gap	in.
ST	Transverse tube spacing	in.
STD, STDL	Dimensionless transverse tube spacing	none

APPENDIX D continued

LIST OF PROGRAM VARIABLES

SYMBOL	DEFINITION	UNITS
SUMZ	see C97	
TA	Air temperature	^oF
TAA, TAR	Tube air inlet temperature	^oF, ^oR
TAF1	Tube interior film temperature	^oF
TB, TG1, TG2, TG3,	Flue gas bulk temperature	^oF
TCHI, TCHO	Chimney flue gas inlet and outlet temperatures respectively	^oF
TFI, TFO, ITFO	Furnace exit temperature	^oF
TFL	Furnace flame temperature	^oF
TFO1	Furnace flue gas mean temperature	^oF
TFOD	Temperature difference	^oF
TG	Flue gas temperature (subroutine)	^oF
TGF	Reclaimer interior film temperature	^oF
TGI	Reclaimer flue gas inlet temperature	^oF
TGO, TGOO	Reclaimer flue gas exit temperature	^oF
THETA	Reference temperature	$^oR/100$
TLIM	Maximum reclaimer casing temperature	^oF
TM1, TM2	Flue gas mean chimney temperature	^oF
TO, NTO	Ambient temperature	^oF
TOTALD	Updraft with reclaimer installed	in. water
TOTALO, TOTALN	Updraft without reclaimer	in. water
TR	Chimney flue gas temperature rise	^oF

APPENDIX D continued

LIST OF PROGRAM VARIABLES

SYMBOL	DEFINITION	UNITS
TTA1, TTA2, TTA3	Tube air bulk temperature	^oF
TTAO	Tube air exit temperature	^oF
TTW1, TTW2, TTW3, TTWB	Tube surface temperature	^oF
TW, TWA, TWB, NTW	Reclaimer surface temperature	^oF
TWR	Reclaimer surface temperature	^oR
U1, UO1	Overall heat-transfer coefficient	$Btu/h\text{-}ft^2\text{-}^oF$
UBINF, UINF, UONF	Tube bank upstream flow velocity	ft/h
UBMAX, UMAX	Maximum tube bank flow velocity	ft/h
VM	Chimney mean velocity	ft/s
VOI, VOO	Flue pipe velocity at reclaimer inlet and outlet respectively	fpm
W	Flue gas mass flow rate	lbm/h
WA	Air mass flow rate per tube	lbm/h
WA1	Total air mass flow rate	lbm/h
WAA	Total air volumetric flow rate	cfm
WEIGHT	Total weight of the reclaimer	lb
WEIT	Weight of tubing per linear foot	lb/ft
X222	Characteristic length	ft
Z111-Z444, Z544	Temperature difference	^oF

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